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SolarDynamics LLC Design Basis Document / Owner's Technical Specification for Nitrate Salt Systems in CSP Projects DOE Grant Number DE-EE0009810	Volume 1 - Specifications for Parabolic Trough Projects	
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SolarDynamics LLC

Design Basis Document / Owner's Technical Specification for Nitrate Salt Systems in CSP Projects

DOE Grant Number DE-EE0009810

Volume 1 - Specifications for Parabolic Trough Projects

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1. Introduction, Report Format, Acknowledgements, and Disclaimer

1.1 Introduction

The work proposed concentrates on a Design Basis Documents / Owner's Technical Specification for parabolic trough and central receiver power plants using nitrate salt as the heat transport fluid and the thermal storage medium. The goals are to 1) distill the successful experience with nitrate salt systems from as many commercial projects as possible, 2) provide technical bases for equipment design/selection that an owner can impose on an EPC contractor, 3) compile this information in one location to provide guidance on salt systems that is as broadly applicable to as many projects as possible, and 4) work toward an industry consensus on a Design Basis Document.

1.2 Project Goal

The principal goal of the project is develop a Design Basis Document for nitrate salt systems in CSP projects that reflects the lessons learned from earlier commercial projects. The document allows an owner to provide design guidance, and to impose a minimum set of requirements, above and beyond those in the normal Codes and Standards, on an EPC contractor in an effort to avoid a repeat of past mistakes. If successful, this would allow salt systems to achieve the same levels of reliability and availability as Therminol systems in commercial parabolic trough plants.

1.3 Report Format

The report consists of 3 volumes:

- Volume 1 - Specifications for Parabolic Trough Projects
- Volume 2 - Specifications for Central Receiver Projects
- Volume 3 - Narrative

Volumes 1 and 2 include discussions of the following topics:

- Plant functional requirements for the salt systems
- Plant operating states, and transitions between states, for the salt systems
- Risk analysis of the principal salt components
- Plant requirements for the salt systems to meet the functional and the operating requirements

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- Type of specification for the principal salt components: functional; or prescriptive
- Current state of the art for salt systems.

Volume 3 discusses a number of cases in which the salt equipment at commercial parabolic trough and central receiver projects has not met the projected levels of reliability and availability. Possible reasons for the sources of the problems are discussed, as is a range of possible alternate designs that could avoid the known problems.

1.4 Acknowledgements

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1.5 Disclaimer

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2. Functional Description

2.1 Introduction

Parabolic trough plants use a field of linear parabolic collectors to redirect, and concentrate, sunlight on a receiver tube located at the focal line of the mirrors. A heat transfer fluid is heated as it circulates through the receiver tubes. Thermal energy in the heat transfer fluid produces steam for use in a conventional Rankine cycle power plant. If so equipped, thermal energy in the fluid can be stored in a storage system for use later in the operating day.

2.2 Collector System

The basic component of the solar field is the solar collector assembly. Each collector assembly is an independently-tracking parabolic trough solar collector made up of parabolic reflectors (mirrors), a metal support structure, the receiver tubes, and a tracking system that includes the drive, sensors, and controls.

The reflectors are made from hot-formed mirrored glass panels. The panels are supported by a truss system that gives the collector assembly its structural integrity.

The receiver tube is constructed of a stainless steel, on which a selective surface ceramic coating is applied to reduce radiation losses. The selective surface has a nominal absorptivity of 0.96 for direct beam radiation, and a design emissivity of approximately 0.095 at 400 °C. The receiver tube is enclosed in an evacuated glass jacket to reduce convection losses. Anti-reflection coatings are applied to the glass envelope. Metal bellows connect the receiver tube with the glass jacket to accommodate the difference in thermal expansion between the two components.

Rotating ball joints accommodate the differential axial and rotational movement between the receiver tubes and the collector structure.

Solar collector designs for commercial projects were developed for use with organic heat transfer fluids; primarily, synthetic oils. Interestingly, for trough projects using inorganic heat transfer fluids, the collector assembly design is essentially unchanged from current designs using synthetic oils. The mirror dimensions, receiver tube wall thickness, receiver tube inside diameter, receiver tube material, and geometric concentration ratios are all suitable for use with inorganic fluids. The principal changes to the collector system for use with inorganic fluids include the following:

- Two-stage flex hose assemblies, rather than rotating ball joints, accommodate the differential movement between the receiver tubes and the collector structure. Flex hoses are required, as suitable sealing materials for ball joints in use with inorganic fluids have yet to be identified

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- Electric impedance heating for the receiver tubes and the flex hose assemblies is added for preheating and freeze protection
- Electric heat tracing on the collector field piping and valves is added for preheating and freeze protection.

2.3 *Heat Transfer Fluids*

2.3.1 Organic Fluids

Working Fluid

For commercial projects today, the heat transport fluid is a eutectic mixture of diphenyl oxide and biphenyl. The common trade names are Therminol VP-1 and Dowtherm A. The oil offers a favorable combination of maximum operating temperature (393 °C), a moderate vapor pressure at the maximum operating temperature (10 bar), and a low freezing point (12 °C). The low freezing point allows the collector field to remain filled throughout the year, with only a small demand for auxiliary energy for freeze protection.

The collector system is connected directly to the steam generation system. Oil circulation is provided by a set of pumps located between the outlet of the steam generator and the inlet to the collector field.

The cold Therminol temperature is on the order of 290 °C. The temperature is defined by the saturation pressure in the evaporator (~ 100 bar), the pinch point in the evaporator (~ 5 °C), and the final feedwater temperature (~ 220 °C). The operating parameters in the steam generator and in the Rankine cycle are selected to provide a Rankine cycle efficiency that is as high as practical, which, in turn, improves the sunlight-to-electric energy conversion efficiency of the project.

Therminol System

A set of process equipment is required to accommodate the thermophysical properties of Therminol. This equipment includes the following items:

- Expansion vessels to accept the change in volume of the Therminol between the minimum circulation temperature (~20 °C) and the maximum working temperature (393 °C)
- Nitrogen supply system to 1) provide an inert cover gas over the Therminol in the expansion vessels, and 2) pressurize the ullage gas to a nominal value of 10 bar to prevent flashing of the Therminol

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- A Therminol cooler, to condense and recycle Therminol vapor released from the expansion vessels
- Activated carbon beds, to capture and adsorb thermal decomposition products of Therminol. The decomposition products are light hydrocarbons, such as benzene. The light hydrocarbons are removed to prevent vapor accumulation in the collector receiver tubes. Once the activated beds are saturated, the activated carbon is shipped offsite for disposal
- A distillation column to separate and remove heavy decomposition products of Therminol. The decomposition process which generates the heavy hydrocarbons proceeds rather slowly, and distillation of the Therminol inventory may not be needed until several years after the start of commercial operation.

2.3.2 Inorganic Fluids

In principle, a wide range of inorganic salts can be used for the heat transfer fluid in a trough project. Candidate anions for the salt mixture include the nitrate ion (NO_3^-) and the nitrite ion (NO_2^-), and candidate cations include sodium, potassium, calcium, magnesium, and lithium.

A review of the thermodynamic properties and the representative costs of the various salt mixtures is presented in Section 10.3. For commercial projects, the only two compounds which offer a commercially viable combination of thermal stability and price are sodium nitrate and potassium nitrate. The typical 60 weight percent / 40 weight percent mixture of sodium nitrate and potassium nitrate, currently in use in commercial solar projects, is the preferred inorganic heat transfer fluid in the collector field.

2.4 Thermal Storage System

2.4.1 Organic Heat Transfer Fluids

Therminol cannot be used directly as a thermal storage medium for two reasons:

- The unit cost of the fluid, in \$/kWht of stored energy, is too high for commercial feasibility
- At the collector field outlet temperature of 393 °C, the vapor pressure of Therminol is a nominal 10 bar. Storing large quantities of Therminol at a pressure of 10 bar would require pressure vessels, and the unit cost of steel for the vessels, in \$/kWht of stored energy, makes the cost of the storage system prohibitive.

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As a result, all of the commercial parabolic trough plants using thermal storage employ an indirect approach. The system consists of a cold salt tank, cold salt pumps, an oil-to-salt heat exchanger, a hot salt tank, and hot salt pumps.

The thermal storage system operates in either a charge cycle or a discharge cycle. During a charge cycle, salt is pumped from the cold tank to the heat exchanger. Therminol from the collector field flows through the heat exchanger in a counterflow arrangement. The high temperature salt at the hot end of the heat exchanger is stored in the hot tank. The low temperature Therminol at the cold end of the heat exchanger returns to the collector field.

During a discharge cycle, salt is pumped from the hot tank to the (same) heat exchanger, but in the reverse direction. Therminol from the steam generator flows through the heat exchanger in a counterflow arrangement. The low temperature salt at the cold end of the heat exchanger is stored in the cold tank. The high temperature Therminol at the hot end of the heat exchanger is supplied to the steam generator.

The indirect design incurs modest penalties in thermodynamic availability due to finite heat exchanger approach temperature during both charge and discharge cycles. The indirect design also requires an oil-to-salt heat exchanger. The heat exchanger is an expensive equipment item, for the following reasons:

- The difference in temperature between the cold Therminol supplied to the collector field (293 °C) and the hot Therminol supplied by the collector field (393 °C) is a modest 100 °C
- To provide a useful difference in temperature between the cold salt tank and the hot salt tank, the approach temperatures at both the cold end and at the hot end of the heat exchanger must both be 'small' (i.e., less than 10 °C) relative to the difference in temperature between the cold Therminol and the hot Therminol.
- The temperature of the Therminol supplied to the steam generator when the plant is operating from storage is lower than the temperature of the Therminol supplied to the steam generator when the plant is operating directly from the collector field. A reduction in the Therminol temperature translates into a reduction in the Rankine cycle efficiency and a reduction in the net plant output. To reduce the performance penalty when the plant is operating from storage, the approach temperature at the hot end of the heat exchanger must be as small as economically practical.
- Commercial plant designs, such as Solana, show the optimum approach temperatures, and the corresponding log mean temperature differences, to be in the range of only 5 to 7 °C. The small temperature differences, in combination with the modest thermal conductivity of nitrate salt, translate into large heat transfer areas and high equipment costs.

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Nonetheless, the cost benefits associated with using a low cost storage medium (nitrate salt), in combination with inexpensive storage tanks operating at atmospheric pressure, more than offset the economic and the efficiency penalties associated with an indirect system.

2.4.2 Inorganic Heat Transfer Fluids

As noted above in Section 2.3.2, the preferred inorganic heat transfer fluid is nitrate salt. Since this salt is the same storage medium currently in use in indirect storage systems, plants using nitrate salt as the heat transfer fluid in the collector field can use a direct storage concept. A direct system eliminates the storage heat exchanger, and avoids the penalties associated with heat exchanger approach temperatures.

2.5 Steam Generation System

2.5.1 Organic Heat Transfer Fluids

The steam generation system uses thermal energy from the high temperature Therminol to produce superheated steam at the conditions required by the turbine-generator and by the auxiliary steam system.

The steam generation system includes the following heat exchangers: superheater; reheater; evaporator; and preheater. Additional components include the steam drum, and water recirculation pumps for the evaporator if the steam generator is a forced recirculation design.

2.5.2 Inorganic Heat Transfer Fluids

The steam generation system for a plant using an inorganic heat transfer fluid is nominally the same as the steam generation system for a plant using an organic heat transfer fluid. However, with an inorganic heat transfer fluid, additional equipment is required to ensure that the temperature of the feedwater supplied to the cold end of the preheater is always high enough to prevent salt from freezing on the outside of the preheater tubes. This additional equipment can take the following forms:

- Recirculation pumps, which draw suction from the steam drum and supply saturated water to a mixing station upstream of the preheater
- A startup feedwater heater, which is supplied with saturated steam from the drum.

2.6 Electric Power Generation System

The electric power generation system converts the energy in the main and the reheat steam to electric power for delivery to the grid. The system consists of the turbine-generator, air cooled condenser,

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condensate system, deaerator, feedwater system, water and steam sampling system, turbine lubrication oil system, and associated pumps and rotating equipment.

The collector field outlet temperature defines, in effect, the operating conditions of the Rankine cycle. For plants with organic heat transfer fluids, the Rankine cycle typically operates with live steam at a pressure of 100 bar and at a temperature of 380 °C, and reheat steam at a temperature of 380 °C. For plants with inorganic heat transfer fluids, representative live steam conditions include a pressure of 125 bar and a temperature of 500 °C, and a reheat steam temperature of 500 °C.

2.7 *Balance of Plant*

The balance of plant equipment supports all other plant functions, which includes, but is not limited to:

- Switch yard / main power distribution system, including main and auxiliary power transformers
- Emergency and uninterruptible power supplies
- Cranes providing access to receiver panels, salt pumps, and heat exchanger tube bundles
- Fire protection and detection systems
- Plant security system
- Compressed air system
- Potable water system
- Cooling water system
- Service water system
- Nitrogen supply system
- Water treatment system
- Deionized water system
- Sanitary waste and industrial waste systems
- Oil/water separator

The balance of plant includes the electric distribution system supplying the motor control centers, grounding, lightning protection, lighting with associated raceway, conduit, and wire.

The balance of plant also includes all site civil work (grading, on-site and off-site drainage, evaporation ponds, and fences), buildings, and the bridge structures over the salt storage tanks.

2.8 *Features of Commercial Projects*

Representative features of commercial projects are briefly discussed in the following sections. These features include site characteristics, labor availability, land, permits, solar radiation, access to transmission, wind, seismic conditions, range of plant sizes, range of storage capacities, water, and soil properties.

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2.8.1 Site Characteristics

The optimum site would have the following characteristics:

- A contiguous section of land, with a shape and an overall area which allows the collector field to be arranged for 1) the best optical performance, and 2) a field header layout which minimizes the parasitic energy demand for pumping. Nonetheless, it can be noted that a commercial project in California was developed without the benefit of a contiguous site, and the annual electric output has been consistent with the projections in the performance model. The collector field is divided between two separate properties, with piping headers connecting the two sub-fields.
- A nominally flat surface, with a minimum of elevation changes, to reduce the amount of grading for drainage
- Acceptable parameters for the following:
 - Haul distance for soil and waste disposal
 - Distance to road and rail transportation connection points
- A location far enough from adjacent mountain ranges that may otherwise 1) block solar radiation early in the morning and late in the afternoon, or 2) channel winds onto the site.

2.8.2 Labor Availability

Most commercial projects are in desert locations to provide annual solar radiation levels that are as high as possible. As might be expected, desert areas often have low population densities, with long distances to large cities.

Commercial projects require a skilled labor force in terms of operators, maintenance personnel, mechanics, welders, instrument technicians, and programmers adept with distributed control systems and programmable logic controllers.

This combination of requirements often presents a problem to commercial projects. The limited population near the project may not have the skills required to properly operate and to maintain the plant. Clearly, incentives can be offered to skilled personnel to relocate to areas near the project, but this will significantly increase the annual staffing cost. Further, there is no guarantee that skilled staff members, once attracted to the project will remain at the project.

It can be noted that this is not a hypothetical problem; it is systemic to the solar industry. In addition, the largest determinants to the annual availability of a project are 1) the technical knowledge of the

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Owner / Project developer, 2) the experience of the engineering company, and 3) the skills and the discipline of the staff to operate the equipment within the limits set by the equipment suppliers ¹.

2.8.3 Land

Securing property for the project, including transmission rights of way, either by purchase or by lease, is highly site specific. The ideal situation is one in which 1) all of the land is under the control of one owner, 2) the site has already been disturbed for a commercial purpose, and 3) there are no competing interests for either the land or the groundwater beneath the site.

2.8.4 Permits

All commercial projects will require land use and construction permits.

As might be expected, the duration and the cost of the permit process will depend on the requirements of the permitting agencies having jurisdiction. The minimum duration is perhaps 1 year, and durations up to 3 years have been experienced.

Items which could postpone, or deny, a project from receiving a permit may include, but is not limited to, 1) visual restrictions due to locations which are close to cities or to airports, 2) limitations on groundwater use, 3) archeological sites or artifacts, 4) endangered species, 5) tribal approvals, and 6) emission limits on projects using organic heat transfer fluids.

Projects using nitrate salt as the thermal storage media, or the heat transfer fluid in the collector field, will generate oxide of nitrogen emissions associated with the thermal decomposition of magnesium nitrate in the salt. However, this is one-time emission during plant commissioning, and there are effective methods for limiting the cumulative release of NO_x.

Projects using nitrate salt as the heat transfer fluid in the collector field can operate at temperatures high enough to thermally decompose the salt. One of the decomposition products is NO_x. However, the NO_x generation rate is on the order of kg per year, and rates this low are not likely to lead to a permitting constraint.

2.8.5 Solar Radiation

To a first order, the levelized cost of energy is inversely proportional to the annual direct normal solar radiation. The minimum practical annual radiation for a commercial project is in the range of 1,700 to 1,800 kWh/m²-year. Many desert sites in the US and in North Africa have annual radiations in the range

¹ Mehos, M., et al (National Renewable Energy Laboratory, Golden, Colorado), Concentrating Solar Power Best Practices Study, Technical Report NREL/TP-5500-75763, June 2020

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of 2,500 to 2,700 kWh/m²-year, while the best sites in the world (Atacama desert in Chile) have annual values on the order of 3,100 kWh/m²-year.

2.8.6 Access to Electric Power Transmission

The site must have access to electric power transmission, both for export to support energy sales and for import to support auxiliary power demands when the plant is not in operation. Feasible distances from the site to major transmission lines range from 1 mile to perhaps 10 miles.

2.8.7 Wind

The collector structures must be designed to survive the highest wind speed expected at the site. A common design wind speed, which is applicable at many commercial sites, is 40 m/sec (90 mph). Nonetheless, collector structures can be designed for essentially any site conditions. For example, projects in Florida must withstand wind speeds expected from hurricanes (60 m/sec (135 mph)).

It can be noted that some commercial projects have suffered damage to collector structures due to brief wind gusts at speeds higher than expected. Two options are available to help avoid this condition: 1) install collectors with a stronger structure near the perimeter of the field, or 2) provide wind fences at the perimeter of the field.

Sites have both a peak wind speed and an average wind speed. The optical accuracy of a collector structure is typically guaranteed for a specific wind speed; a representative value is 7 to 8 m/sec. For wind speeds above the guarantee value, the collector continues to track, but with a reduced optical accuracy and a reduced thermal output. When the wind speed reaches a maximum operating velocity of 11 to 12 m/sec, the collectors are moved from the track position to the stow position. This protects the structures from potentially damaging wind gusts during the somewhat protracted period (2 to 3 minutes) required to stow the field. As such, sites with a high percentage of operating hours with wind speeds below the guarantee value will offer a better solar-to-electric efficiency than sites which operate with a high fraction of wind speeds above the guarantee, but still below the stow, value.

2.8.8 Seismic Conditions

The design of the solar collector assembly is typically governed by wind loads rather than seismic conditions. Unless the site is within a few kilometers of major fault, the design seismic conditions will not influence the choice of the plant location.

Seismic accelerations will produce sloshing movements in the inventory of the salt tanks. To protect the frangible welded connection between the top of the wall and the perimeter of the roof from excessive loads, freeboard allowances are provided, as specified in the API tank design codes.

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2.8.9 Range of Plant Sizes

The operation and maintenance staff requirements in commercial projects are relatively insensitive to the size of the project; i.e., the staff required to operate a 100 MWe project is not much larger than the staff required to operate a 30 MWe project. As such, the larger the project, the lower the unit operation and maintenance cost in \$/kWh.

The majority of commercial projects built to date have a gross plant rating of 50 MWe. However, this was largely an artifact of the project locations (Spain) and the subsidy provisions offered by the Spanish government. Projects outside of Spain have generally been larger, ranging from the 80 MWe SEGS VIII and IX projects in the US, up to the 210 MWe DEWA projects in Dubai.

2.8.10 Range of Storage Capacities

Although many early plants were built without thermal storage, parabolic trough technology can no longer compete on an economic basis with photovoltaic plants for daytime generation. As a result, trough plants developed during the past 2 decades have included storage capacities sufficient to operate the Rankine cycle, at full load, for periods of 6 to 12 hours.

With thermal storage, solar fields are sized to operate the Rankine cycle at full load during the day plus provide additional energy to charge the storage system so that the plants can continue operating after sunset.

2.8.11 Water

Essentially all recent commercial projects, and all future projects will, use dry cooling for heat rejection from the Rankine cycle. Dry cooling reduces water consumption by approximately 90 percent relative to a plant using wet cooling. The reduced water consumption, in turn, simplifies the permitting process and makes available a wider range of potential site locations. As an example, a commercial solar project in the Atacama desert in Chile has no access to municipal or ground water. All of the water used at the site is brought in by truck from the coast. Such an arrangement would be impossible were the plant to have used wet cooling.

For a plant with dry cooling, the principal water demands include 1) distilled water for mirror washing, 2) makeup water to replace blowdown losses from the Rankine cycle, 3) makeup water to replace blowdown losses from the wet surface air cooler of the closed cooling water system, and 4) makeup water to replace steam losses from the condenser ejectors during turbine startup. This are also minor water demands for sanitary systems, building humidification, fire water testing, and closed loop cooling water maintenance.

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2.8.12 Soil Properties

The properties of the soil influence the plant design in the following areas:

- Collector foundations. The foundations for the collector structure can consist of concrete piles, steel piles, screw piles, or concrete footings. The selected approach is based on the strength of the soil, the site design wind speed, the presence, or the absence, of rocks in the soil, and the depth of the ground water
- Thermal storage tank foundations. Essentially all commercial salt tanks use a concrete base mat underneath the floor insulation. This is an economical design, but it requires the weight of the static head of the salt to be less than the allowable bearing load for the soil. A representative soil bearing load is 240 kPa (5,000 lb_f/ft²), which, in turn, limits the maximum salt depth in the tank to a nominal value of 12 m. If the allowable soil bearing load is less than 240 kPa, or if the salt tank is designed with a inventory height greater than 12 m, friction piles can be installed below the concrete base mat to increase the bearing capacity of the soil. However, the addition of friction piles will significantly increase the cost of the foundation
- Water table. The elevation of the water table should be significantly lower than the elevation of the concrete base mat for the storage tanks. A low water table helps to prevents liquefaction of the soil during an earthquake, and limits the vertical heat flux from the tank foundation into the soil
- Percolation. Following a rain, any standing water would ideally percolate into the soil in less than one day. This reduces the extent of puddles and soft soil, which could otherwise make access to the collector field problematic for maintenance vehicles.

2.8.13 Soiling of Mirrors

Soiling refers to the rate at which the mirrors lose their reflective properties. The reduction in reflectivity is due to the deposition of soluble and insoluble particles on the mirror surface. This can have significant influence on both the plant production and the resources required to keep the mirror reflectivity at acceptable levels through periodic cleaning. Factors that affect soiling rates include the following:

- Humidity. Particles are more likely to adhere to the mirror surface if the humidity is high.
- Rainfall. Rain, and particularly snow, are effective methods for cleaning the mirrors.
- Soil properties. Sandy soil is associated with larger particles than clay soils; hence, the transport of particles into the air, and on to the mirror surface, is less likely to occur with sandy soils.
- Dust. The common sources of dust include wind and local vehicle traffic.

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It is advised that the rate of soiling be quantitatively evaluated during the development phase of the project.

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3. Operating States, and Transitions Between States - Organic Heat Transfer Fluids

3.1 Introduction

For the purposes of defining the plant operating states, and the transitions between operating states, a parabolic trough solar plant using an organic heat transfer fluid with nitrate salt thermal storage can be divided into two primary operating sections:

- Energy collection section, consisting of the collector system and the thermal storage system (charging mode)
- Energy conversion section, consisting of the thermal storage system (discharging mode), the steam generation system, and the electric power generation system.

3.2 Operating States for the Energy Collection System

For the energy collection section, there are three operating states: Long Term Hold; Overnight Hold / Standby; and Normal Operation (Refer to Figure 3-1 and to Table 3-1).

- Long Term Hold: The collector field is stowed. The field piping is filled (as there is no mechanism to drain the piping), and the Therminol pumps are off. The oil-to-salt heat exchangers are drained on the salt side, and the salt inventory is divided between the cold tank and the hot tank. The tank standby electric heaters are active to maintain the temperature of the salt inventory above the freezing point
- Overnight Hold / Standby: The collector field is stowed, and the Therminol pumps are circulating oil through the collector field. The oil-to-salt heat exchangers are filled on both the oil- and the salt-sides, but there is no oil flow or salt flow through the heat exchangers. The oil flow is stopped to ensure that potentially cold oil from the collector field does not cause salt to freeze in the heat exchanger. If the temperature of the heat exchanger starts to approach the freezing point of the salt, freeze protection can be provided by 1) activating the electric heat tracing on the heat exchanger, or 2) circulating salt from the cold tank, through the heat exchanger, and back to the cold tank
- Normal Operation: The collector field is tracking the sun, and the Therminol pumps are circulating oil through the collector field. Oil flow rates greater than that required by the steam generator are supplied to the oil-to-salt heat exchanger. The cold salt pump is in operation, supplying salt to the oil-to-salt heat exchanger. The speed of the cold salt pump is adjusted to provide a nominal salt temperature, at the hot end of the heat exchangers, of 385 °C.

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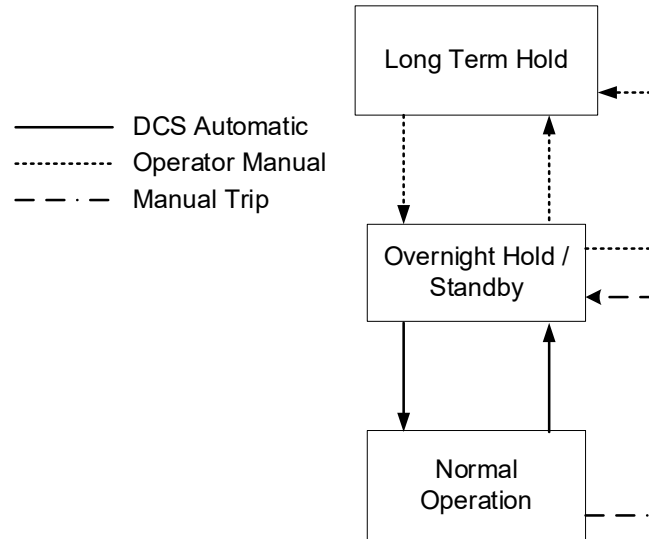


Figure 3-1 States and Transitions for the Energy Collection Section - Organic Fluids

3.3 *Transitions Among Operating States for the Energy Collection System*

The four transitions among states for the energy collection system include the following:

- **Long Term Hold to Overnight Hold / Standby:** The Therminol pumps are started to establish a recirculation flow in the collector field. The oil-to-salt heat exchangers, which are filled on the oil side, are preheated to a nominal temperature of 290 °C by electric heat tracing, oil circulation, or a combination of both. The heat exchangers are filled with cold salt. The cold salt pumps operate in recirculation
- **Overnight Hold / Standby to Normal Operation:** The collector field is moved from the stow position to the track position. The speed of the Therminol pumps is adjusted to provide a collector field outlet temperature of 393 °C. Flows of Therminol and salt are established in the oil-to-salt heat exchangers, and the speed of the cold salt pump is adjusted to provide a nominal salt temperature of 385 °C at the inlet to the hot tank
- **Normal Operation to Overnight Hold / Standby:** The collector field is moved from the track position to the stow position. The Therminol flow is bypassed on the oil side of the oil-to-salt heat exchangers, and the cold salt pump is either operated in recirculation or stopped, depending on the length of the hold period

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Table 3-1 Energy Collection Section States and Equipment Status - Organic Fluids

	Long Term Hold	Overnight Hold / Standby	Normal Operation
Collector Field			
Solar collectors	Stow	Stow	Track
Field piping	Filled	Filled	Filled
Thermal Storage			
Cold tank	Intermediate level	Intermediate level	Intermediate level
Hot tank	Intermediate level	Intermediate level	Intermediate level
Therminol Pumps	Off	Recirculation through collector field	On
Oil-to-Salt Heat Exchangers			
Oil side	Filled	Filled	Filled
Salt side	Empty	Filled	Filled
Salt Pumps			
Cold salt	Off	Off	On, with speed determined by charging duty
Hot salt	Off	Off	Off
Thermal Conditioning			
Tank heaters	Active	Active	Off
Electric heat tracing	Inactive; salt inventory in storage tanks	Active, with set points below cold salt temperature	Active, with set points below normal operating temperatures

- Overnight Hold / Standby to Long Term Hold: The Therminol circulation pumps are stopped, and the oil-to-salt heat exchangers are drained on the salt-side. The tank electric heaters remain active to prevent the salt inventory from freezing.

3.4 Operating States for the Energy Conversion System

For the energy conversion section, there are five operating states: Long Term Hold, Overnight Hold, Auxiliary Steam, Turbine Synchronization, and Normal Operation (Refer to Figure 3-2 and to Table 3-2).

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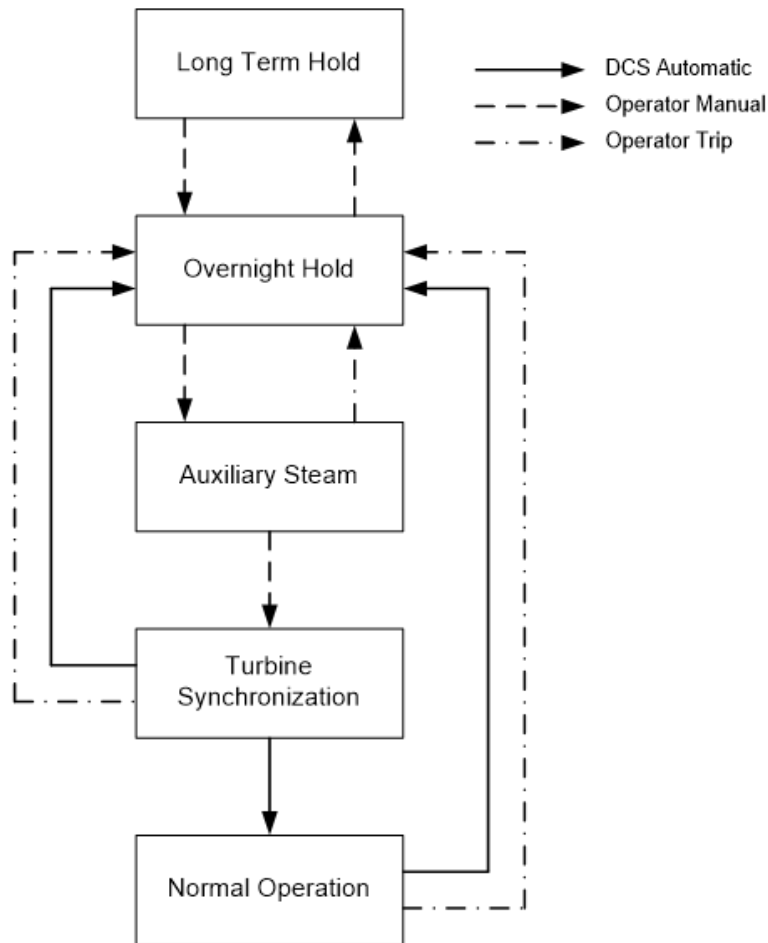


Figure 3-2 States and Transitions for the Energy Conversion Section - Organic Fluids

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Table 3-2 Energy Conversion Section States and Equipment Status - Organic Fluids

	Long Term Hold	Overnight Hold	Auxiliary Steam	Turbine Synchronization	Normal Operation
Thermal Storage					
Cold tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Hot tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Oil-to-Salt Heat Exchangers					
Oil side	Filled	Filled	Filled	Filled	Filled
Salt side	Empty	Filled	Filled	Filled	Filled
Salt Pumps					
Cold salt	Off	Off	Off	Off	Off
Hot salt	Off	Off	Off	On, with speed determined by discharging duty	On, with speed determined by discharging duty
Steam Generator					
Therminol side	Filled	Filled	Filled	Filled	Filled
Water side	Empty	Filled	Filled	Filled	Filled
Water Pumps					
Condensate	Off	Off	On	On	On
Feedwater	Off	Off	On	On	On
Water recirculation ¹	Off	On	On	On	On
Auxiliary Steam	Off	Off	On	Off	Off
Turbine - Generator					
Turbine	Turning Gear	Turning Gear	Turning Gear	Part load	Full load
Condenser	Ambient ²	Ambient ²	Vacuum	Vacuum	Vacuum
Generator	Off	Off	On	Part load	Full load
Thermal Conditioning					
Tank heaters	Active	Active	Active	Off	Off
Electric heat tracing	Inactive; salt inventory in storage tanks	Active, with set points below cold temperature	Active, with set points below cold temperature	Active, with set points below normal operating temperatures	Active, with set points below normal operating temperatures

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Notes:

1. Forced recirculation evaporators
2. If the plant breaks condenser vacuum during hold periods

Each of the operating states in the energy conversion section are briefly described as follows:

- **Long Term Hold:** The oil-to-salt heat exchanger is drained on the salt side. The steam generator is drained on the water side, and the steam turbine is rotated on the turning gear
- **Overnight Hold:** The oil-to-salt heat exchanger is filled on the salt side. The cold salt pump is in operation, and salt supplied to the heat exchanger is recirculated to the cold salt tank. The steam generator is filled on both the Therminol and the water/steam sides; however, the steam production rate is 0 kg/sec. The steam turbine is rotated on the turning gear
- **Auxiliary Steam:** The auxiliary boiler is in service. Sealing steam, after superheating by an electric superheater, is supplied to the turbine shaft seals, and a vacuum is established in the condenser
- **Turbine Synchronization:** Once the Therminol temperature from the collector field reaches a value consistent with the metal temperatures in the steam generator, oil is supplied to the steam generator. The steam generator is started, and steam is dumped to the condenser until the steam conditions match the startup conditions for the turbine. The turbine-generator is synchronized with the grid, and a minimum turbine output is established
- **Normal Operation:** The extraction feedwater heaters are placed in service, and the output of the turbine-generator is increased to the rated value. Oil is supplied to the steam generator from the collector field, from the storage system, or from a combination of both.

3.5 Transitions Among Operating States for the Energy Conversion System

The six transitions among operating states for the energy conversion system are summarized as follows:

- **Long Term Hold to Overnight Hold:** Therminol is circulated through the collector field to match the temperature of the heat collection elements with the temperature of the field headers. During those periods when the temperature of the Therminol is below 300 °C, the heat exchanger is bypassed on the oil side. The temperature of the oil-to-salt heat exchangers is maintained at the cold salt temperature by recirculating salt from the hot end of the heat exchanger back to the cold tank

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- **Overnight Hold to Auxiliary Steam:** The auxiliary steam generator is started, a flow of steam to the turbine shaft seals, after superheating by an electric superheater, is established, and a vacuum is drawn in the condenser
- **Auxiliary Steam to Turbine Synchronization:** The steam generator is placed in service when the temperature of the Therminol from the collector field is consistent with the metal temperatures in the steam generator. Live steam and the cold reheat steam production begin. The steam flows are throttled, and sent to the condenser through the bypass system. As soon as the live steam and the hot reheats steam temperatures satisfy the requirements set by the turbine vendor, a portion of the live steam and hot reheat steam flows are diverted from the bypass system to the turbine. The turbine generator is accelerated to 100 percent speed, synchronized with the grid, and the minimum turbine output is established
- **Turbine Synchronization to Normal Operation:** The extraction feedwater heaters are placed in service. The output of the turbine follows the thermal energy supplied to the steam generator by the flow of Therminol. The Therminol can be supplied by the collector field, by the storage system, or a combination of each
- **Normal Operation to Overnight Hold:** The flow of Therminol is reduced to a value consistent with the minimum turbine output. The turbine is then tripped to preserve the metal temperatures at values as high as practical during the subsequent hold period. Live steam and hot reheat steam is bypassed to the condenser until the steam generator temperatures reach the overnight hold condition, at which time the steam generator is tripped
- **Overnight Hold to Long Term Hold:** The oil-to-salt heat exchangers are drained on the salt-side, and the steam generator heat exchangers are drained on the water-side.

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4. Operating States, and Transitions Between States - Inorganic Heat Transport Fluids

4.1 Introduction

Similar to the discussion above for organic heat transfer fluids, a parabolic trough solar plant using an inorganic heat transfer fluid can be divided into two primary operating sections:

- Energy collection section, consisting of the collector system and the thermal storage system
- Energy conversion section, consisting of the steam generation system and the electric power generation system.

4.2 Operating States for the Energy Collection System

For the energy collection section, there are three operating states: Long Term Hold / Standby, Cloud Standby, and Normal Operation (Refer to Figure 4-1 and to Table 4-1).

- Long Term Hold / Standby: The collector field is stowed. The field piping is filled (as there is no mechanism to drain the piping), and the cold salt pumps are in service. Continuous salt circulation is the only mechanism to prevent salt from freezing in the heat collection elements. The salt inventory is divided between the cold tank and the hot tank. The tank electric heaters are active to 1) compensate for the heat losses from the collector field, and 2) maintain the inventory temperatures above the salt freezing point
- Cloud Standby: The cold salt pumps are in service, and the collector field is tracking the sun. However, the aperture normal radiation is not sufficient to establish the design salt outlet temperature from the collector field. The flow from the field is recirculated back to the cold tank to prevent a decay in the inventory temperature of the hot tank. To prevent the salt in the heat collection elements from overheating at the end of a cloud transient, the flow rate is set to the clear sky value. The clear sky flow rate is the value associated with a collector field outlet temperature equal to the design value, which would occur if the aperture normal radiation was equal to the theoretical clear sky value
- Normal Operation: The collector field is tracking the sun, and salt at the design temperature is delivered to the hot salt tank.

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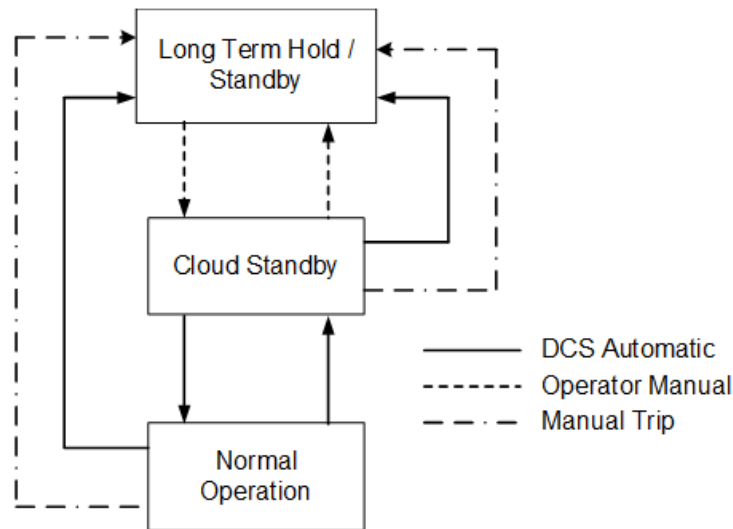


Figure 4-1 Energy Collection Section States and Transitions - Inorganic Fluids

Table 4-1 Energy Collection Section States and Equipment Status - Inorganic Fluids

	Long Term Hold / Standby	Cloud Standby	Normal Operation
Collector Field			
Solar collector assemblies	Stow	Track	Track
Field piping	Filled	Filled	Filled
Thermal Storage			
Cold salt tank	Intermediate level	Intermediate level	Intermediate level
Hot salt tank	Intermediate level	Intermediate level	Intermediate level
Salt Pumps			
Cold salt circulation	On, recirculation flow to cold tank	On; recirculation flow to cold tank	On
Thermal Conditioning			
Immersion heaters	Active	Active	Off
Electric heat tracing	Inactive; salt inventory in storage tanks	Active, with set points below cold salt temperature	Active, with set points below normal operating temperatures

4.3 Transitions Among the Operating States for the Energy Collection System

There are five transitions among the operating states for the energy collection system, as follows:

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- **Long Term Hold / Standby to Cloud Standby:** The aperture normal radiation is not sufficient to establish a collector field outlet temperature equal to the design value. The speed of the cold salt pump is increased to a value which establishes the clear sky flow rate. The collectors are moved from the stow position to the track position, and the flow from the field is recirculated back to the cold tank
- **Cloud Standby to Normal Operation:** The aperture normal radiation is sufficient to establish a collector field outlet temperature equal to the design value, and the speed of the cold salt pumps is adjusted to reach this set point. When the collector field reaches the crossover temperature (approximately 500 °C), the flow is diverted from the cold tank to the hot tank
- **Normal Operation to Cloud Standby:** The aperture normal radiation is no longer sufficient to maintain a collector field outlet temperature equal to the design value. The speed of the cold salt pump is increased to a value which establishes the clear sky flow rate. When the collector field falls below the crossover temperature (approximately 500 °C), the flow is diverted from the hot tank to the cold tank
- **Normal Operation to Long Term Hold / Standby:** The aperture normal radiation is no longer sufficient to maintain a collector field outlet temperature equal to the design value. The collector field is moved to the stow position, and the speed of the cold salt pumps is reduced to the Long Term Hold / Standby condition
- **Cloud Standby to Long Term Hold / Standby:** The collector field is moved to the stow position, and the speed of the cold salt pumps is reduced to the Long Term Hold / Standby condition.

4.4 Operating States for the Energy Conversion Section

The energy conversion section operates in one of the following six states (Refer to Figure 4-2 and to Table 4-2):

- **Long Term Hold:** The steam generator is drained on both the water side and the salt side, the electric heat trace circuits are inactive, and the electric recirculation water heaters are inactive
- **Overnight Hold:** The attemperation pump supplies cold salt to the steam generator to maintain heat exchangers temperatures above 275 °C. The steam turbine is rotated by the turning gear. Depending on the requirements of the project, turbine shaft sealing steam may need to be produced to maintain condenser vacuum

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Table 4-2 Energy Conversion Section States and Equipment Status - Inorganic Fluids

	Long Term Hold	Overnight Hold	Isothermal Steam Production	Superheated Steam Production	Turbine Synchronization	Normal Operation
Steam Generator						
Salt side	Empty	Filled	Filled	Filled	Filled	Filled
Water side	Empty	Filled	Filled	Filled	Filled	Filled
Startup feedwater heater	Empty	Filled	Filled	Filled	Filled	Filled
Thermal Storage						
Cold tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Hot tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Salt Pumps						
Hot salt	Off	Off	On	On	On	On
Attemperation	Off	Pump maintains minimum system temperature	Salt attemperation as required	Salt attemperation as required	Salt attemperation as required	Off
Water Pumps						
Condensate	Off	Off	On	On	On	On
Feedwater	Off	Off	On	On	On	On
Steam generator recirculation	Off	On	On	On	On	On
Auxiliary Steam						
Electric boiler	Off	Off	Initial demand for turbine seals and condenser vacuum	Off	Off	Off
Steam generator	Off	Off	On	On	On	On
Turbine - Generator						
Turbine	Turning gear	Turning gear	Turning gear	Turning gear	Part load	Full load
Condenser	Empty	Nitrogen	Vacuum	Vacuum	Vacuum	Vacuum
Generator	Off	Off	Off	Off	Part load	Full load
Balance-of-Plant	As required	As required	As required	As required	On	On
Thermal Conditioning						
Tank heaters	Energized	Intermittent	Intermittent	Intermittent	Off	Off
Heat tracing	Off; all systems drained	Activated as required for preheating and protection of equipment from salt freezing	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby
Steam generator water heater	Off; water side drained	On, to preheat heat exchangers to 290 °C	Off	Off	Off	Off

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- Isothermal Steam Production:** The auxiliary steam generator is in service, supplying shaft sealing steam to the turbine. A vacuum is drawn in the condenser. Cold salt is supplied to the hot end of the superheater and to the hot end of the reheater. The salt flows from the cold ends of the two steam heat exchangers combine at a hot salt / cold salt mixing station upstream of the hot end of the evaporator. The mixing station is illustrated in the process flow diagram Figure 6-7 of Volume 3 - Narrative. Thermal energy in the hot salt is sufficient to establish a saturated steam flow rate equal to 20 percent of the design steam flow rate. The saturation pressure in the evaporator is set by the turbine bypass system such that the saturation temperature is equal to the cold salt temperature. The saturated steam flow from the evaporator passes through the superheater and then through the reheater. Since the salt temperature in the superheater and in the reheater is equal to the saturation temperature, no heat transfer occurs in either heat exchanger. However, during the period in which the steam flow rate increases from 0 percent to 20 percent (slightly above the vendor's minimum), non-uniform steam distributions in the heat exchangers are not likely to produce damaging stress distributions
- Superheated Steam Production:** The salt mixing station is transferred from the startup point upstream of the evaporator to the normal point upstream of the superheater / reheater. The mixing station is illustrated in the process flow diagram Figure 6-8 of Volume 3 - Narrative. The steam production rates decays slightly to a nominal 16 percent (the vendor's minimum) due to the superheating duties of the superheater and the reheater
- Turbine Synchronization:** A live steam flow rate of (Project specific) kg/hr, with a temperature and pressure of (Project specific) °C and (Project specific) bar, respectively, as specified by the turbine vendor, are established. The turbine-generator is synchronized with the grid. A minimum turbine output of (Project specific) MWe is established to preclude generator trips on reverse power
- Normal Operation:** The extraction feedwater heaters are placed in service. A live steam flow rate of (Project specific) kg/hr, with a temperature and pressure of (Project specific) °C and (Project specific) bar, respectively, are established. For turbine loads less than approximately 90 percent, the turbine operates in fixed pressure control. This ensures that the pinch point temperature in the evaporator is high enough that the salt temperature at the cold end of the preheater remains above the minimum value. For turbine loads greater than 90 percent, the turbine operates in sliding pressure control to maximize the efficiency of the Rankine cycle. Turbine output is set by adjusting the speed of the hot salt pump. The design value for the hot reheat steam temperature is set by adjusting the distribution of the hot salt flow rate between the superheater and the reheater.

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4.5 Transitions Between States for the Energy Conversion Section

The nine transitions between the states for the energy conversion section are as follows:

- **Long Term Hold to Overnight Hold:** The temperatures of the steam generator heat exchangers and the inter-vessel piping are raised to 275 °C by the electric water heaters and the electric heat tracing. The steam generation system attemperation pump is started, and a flow of cold salt is established through the heat exchangers
- **Overnight Hold to Isothermal Steam Production:** The electric auxiliary steam generator is started, supplying sealing steam to the turbine shaft seals. A vacuum is drawn in the condenser. The hot salt pump is started, supplying hot salt to a mixing station upstream of the evaporator. Saturated steam production begins in the evaporator, with a saturation temperature equal to the cold salt temperature. Saturated steam flows through the superheater and the reheater; however, no heat transfer occurs. This allows the steam production rate to increase from 0 percent to a nominal 20 percent without flow maldistributions on the tube sides generating potentially damaging stress distributions. Steam from the steam generator is sent through the turbine bypass system to the condenser
- **Isothermal Steam Production to Superheated Steam Production:** The salt mixing station is transferred from the startup point upstream of the evaporator to the normal point upstream of the superheater / reheater. The flow rates of cold salt and hot salt remain nominally fixed. The metal temperatures of the superheater and the reheater both increase, which results in superheating occurring in both the superheater and in the reheater. Due to the addition of superheating duties, the steam flow rate decays slightly to 16 percent, which is representative of the minimum flow rate set by the heat exchanger vendor
- **Superheated Steam Production to Turbine Synchronization:** When the temperatures of the live steam and the reheat steam meet the startup requirements of the turbine, a portion of the steam in the bypass system is diverted to the turbine, the turbine is accelerated to synchronous speed, the turbine is synchronized with the grid, and output of the turbine is increased to the point where the generator is not at risk of tripping on reverse power. (A note: Different turbine vendors have different mechanisms for starting their turbines.) Steam not required by the turbine flows to the condenser through the bypass system
- **Turbine Synchronization to Normal Operation:** The extraction feedwater heaters are placed into service, the balance of the steam flow in the bypass system is sent to the turbine, the attemperation flow of cold salt is reduced consistent with an the maximum allowable rate of temperature change for the superheater / reheater, and the flow rate of hot salt is increased to the design value

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- **Normal Operation to Isothermal Steam Production:** The speed of the hot salt pump is reduced, which reduces the output of the turbine. When the turbine reaches the minimum allowable load, the turbine is tripped. Tripping the turbine at the full superheat / reheat temperature maintains the turbine metal temperatures as high as possible during the overnight hold period, which reduces the time, the thermal energy, required for the next startup. However, the steam generator is not tripped when the turbine is tripped, and the continuing steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of (only) cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent
- **Isothermal Steam Production to Overnight Hold:** The hot salt / cold salt mixing station is transferred from a point upstream of the superheater / reheater to the startup point upstream of the evaporator. During the transition between mixing stations, the superheating duty decreases relative to the evaporation duty. This results in 1) an increase in the steam flow rate to a nominal value of 20 percent, 2) a decay in the metal temperatures of the superheater / reheater to the cold salt temperature. The rate of change in the metal temperatures can be controlled, in part, by adjusting the saturation temperature in the evaporator. During the transition, the goal is to provide steam flow rates that are high enough to 1) ensure a reasonably uniform flow distribution among the tubes, and 2) reduce the potential for damaging temperature and stress distributions in the superheater / reheater
- **Turbine Synchronization to Isothermal Steam Production:** The transition, although not a normal occurrence, is essentially a generator trip on reverse power. Although the turbine trips, the steam generator is not tripped, and the steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of (only) cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent
- **Overnight Hold to Long Term Hold:** The attemperation pump is stopped, the salt equipment is drained to the main storage tanks, the electric heat trace system is turned off, the water side of the equipment is drained, and nitrogen is supplied to the heat exchangers for corrosion control.

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5. Risk Analysis - Organic Heat Transport Fluids

5.1 Introduction

The goal of the risk analysis is to identify the principal equipment items that have a significant influence on the plant availability and the annual plant performance. The evaluation includes an assessment of probability and consequence to identify the most important items to be included in the technical specification.

As background to, and a starting point for, the risk assessment, an evaluation of the technical, commercial, and operating risks in commercial solar projects was recently conducted by NREL². The SolarPACES CSP project database³ was used to identify the projects that are currently in commercial operation, including 76 parabolic trough plants and 14 central receiver projects. Over the course of the project, the project team held about 50 information gathering sessions, representing nearly two-thirds of the plants operating worldwide.

The principal technical risks associated with parabolic trough projects are summarized in Table 5-1. The projects surveyed include plants both with and without thermal storage. The items associated with the salt equipment are noted in blue.

To help identify which issues are most important, each issue entered in the database was given an impact score and a risk level. The impact score identified the potential impact of the issue to the project in terms of the effects on plant performance, cost, or schedule. The impact score was ranked as 1 (low) to 5 (high). The risk level was an indication of how likely the problem was to happen. A risk level of 1 meant that the problem was rarely experienced or maybe was only associated with a problem at a single plant. A risk level of 5 meant that it was a common problem or could affect many plants. The scores are multiplied to create a priority score. Priority scores can range from 1 to 25 for each issue. The ranking is, of course, subjective, but it is an attempt to give some quantification to the importance of the issues.

The most significant issues were brought up by multiple participants. The number of “occurrences” does not correspond to the number of times some type of incident or issue occurred at a plant, but rather the number of times it was mentioned by the plant personnel. The number of occurrences in which an issue is mentioned indicates how important the issue is to the stakeholders.

² Mehos, Mark, et. al, (National Renewable Energy Laboratory, Golden, Colorado), ‘Concentrating Solar Power Best Practices Study’, Technical Report NREL/TP-5500-75763, June 2020

³ <http://www.nrel.gov/csp/solarpaces/>

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Table 5-1 Principal Risks for Parabolic Trough Projects Identified in the Best Practices Report

System	Component	Issue Type	Occurrence	Weight
Solar field	Receivers	Hydrogen	14	320
Heat transfer fluid	Interconnect	Ball joint vapor	9	205
Power block	Therminol steam generator	System design	11	183
Heat transfer fluid	Ullage	Ullage system design	8	174
Heat transfer fluid	Interconnect	Ball joint stress	9	169
Heat transfer fluid	Therminol pumps	Seal leakage	11	159
Power block	Steam turbine-generator	Turbine start-up	11	139
Solar field	Structure	Wind load design	9	127
Project	O&M	Mirror cleanliness	8	118
Power block	DCS	DCS logic	6	110
Power block	Steam turbine-generator	Turbine reliability	9	99
Heat transfer fluid	Piping	Valve design	6	98
Power block	Therminol steam generator	Exchanger reliability	6	94
Power block	Steam turbine-generator	Turbine blade failure	5	85
Heat transfer fluid	System	Loop flow balance	6	84
Heat transfer fluid	Piping	Valve reliability	4	80
Power block	DCS	DCS design	6	80
Heat transfer fluid	Instrumentation	Flow meter reliability	5	75
Heat transfer fluid	Piping	Piping support design	5	75
Power block	Electrical	Step-up transformer	5	75
Thermal storage	Oil-to-salt heat exchanger	Reliability	5	75
Heat transfer fluid	Piping	Loop isolation valves	6	70
Project	O&M	O&M labor costs	4	70
Thermal storage	Oil-to-salt heat exchanger	Heat exchanger design	4	70
Heat transfer fluid	System	Field flow control	3	65
Power block	Therminol steam generator	Heat exchanger design	5	65
Thermal storage	Salt tanks	Tank design	6	60
Heat transfer fluid	Piping	Piping design	5	57
Heat transfer fluid	Auxiliary heater	Heater design	6	56
Heat transfer fluid	Therminol pumps	VFDs	2	50
Heat transfer fluid	Fluid	HTF degradation	3	43
Power block	Therminol steam generator	Manufacturing QC	2	40
Power block	Therminol steam generator	Control logic	2	40
Heat transfer fluid	Expansion system	System design	5	39
Solar field	Receivers	Receiver Reliability	5	33
Heat transfer fluid	System	HTF Leaks	2	30
Thermal storage	Oil-to-salt heat exchanger	Maintenance	2	30

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System	Component	Issue Type	Occurrence	Weight
Thermal storage	Oil-to-salt heat exchanger	Manufacturing QC	2	30
Thermal storage	Piping	Heat tracing	2	30
Solar field	Civil	Site design	3	27
Heat transfer fluid	Fluid	Safety	1	25
Heat transfer fluid	Fluid	HTF properties	1	25
Heat transfer fluid	Interconnect	Limited suppliers	1	25
Heat transfer fluid	Interconnect	Flex hoses	1	25
Heat transfer fluid	Piping	Insulation quality	1	25
Heat transfer fluid	System	HTF flow control	1	25
Power block	Steam cycle	Water supply	1	25
Solar field	Control system	Control design	1	25
Power block	Steam cycle	VFDs	2	24
Project	O&M	O&M provider quality	2	24
Solar field	System	Design standards	2	24
Solar field	Local controllers	Control system	3	23
Solar field	Mirrors	Mirror breakage	3	23
Power block	Steam cycle	Gland steam system	2	18
Power block	Steam cycle	Valve reliability	2	18
Thermal storage	Piping	Valve reliability	2	18
Thermal storage	Salt pumps	Salt pump design	2	18
Heat transfer fluid	Expansion system	Safety valves	1	15
Heat transfer fluid	Piping	Pump bellows leakage	1	15
Heat transfer fluid	Piping	Welding	1	15
Power block	Auxiliary systems	Heat tracing	1	15
Power block	Auxiliary systems	Cooling - auxiliary	1	15
Power block	Electrical	System design	1	15
Power block	Therminol steam generator	Reliability	1	15
Project	EPC	Experience	1	15
Project	O&M	O&M staff quality	1	15
Project	Structure	Organization interfaces	1	15
Solar field	Elect & I&C	System design	1	15
Solar field	Elect & I&C	Lightning	1	15
Thermal storage	Piping	Control valve leaks	1	15
Solar field	Instr. & LOC	Field communication	2	14
Heat transfer fluid	Piping	HTF vapor - venting	1	9
Heat transfer fluid	Solar field loops	Temperature control	1	9
Power block	Auxiliary system	Instrument air	1	9
Power block	Civil	Turbine foundations	1	9
Power block	Steam cycle	Condenser material	1	9

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System	Component	Issue Type	Occurrence	Weight
Power block	STG	Generator reliability	1	9
Power block	STG	Lube oil system	1	9
Project	EPC	EPC execution	1	9
Project	EPC	Schedule	1	9
Project	O&M	O&M costs	1	9
Project	O&M	Capital improvement	1	9
Project	O&M	Solar field maintenance	1	9
Solar field	Civil	Collector foundations	1	9
Solar field	Elect & I&C	Solar field grounding	1	9
Solar field	Instr. & LOC	Inclinometer	1	9
Solar field	Structure	Solar field corrosion	1	9
Solar field	System	Temperature control	1	9
Thermal storage	Salt pumps	Pump alignment	1	9
Heat transfer fluid	Piping	HTF Leaks	1	5
Power block	Electrical	Generator breaker	1	5
Thermal storage	Salt pumps	Seal leakage	1	5
Power block	Steam turbine-generator	Heating blankets	1	3
Solar field	Drives	Seal leakage	1	3
Solar field	Civil	Water supply	1	1

A review of the table shows the following:

- The salt equipment topics with the highest weight values are the design and the reliability of the oil-to-salt heat exchangers. However, the risks are not at, or near, the top of the list. This is due to the following effects:
 - Only a limited number of the projects use thermal storage
 - The heat exchangers are hydraulically coupled to the outlet of the collector field, and to the outlet of the steam generator. Since the rates of temperature change at the outlet of the field, and at the outlet of the preheater, tend to be modest, the rates of temperature change seen at the oil-to-salt heat exchanger are also modest. In turn, modest rates of change generally result in a limited influence on the low cycle fatigue life of the heat exchanger
- In contrast to the hot salt tanks in central receiver projects, no failures have occurred in either the cold tanks or the hot tanks of parabolic trough projects. This can, in large part, be attributed to 1) the use of carbon steel for both the cold and the hot tanks, and 2) the tanks are partially isolated from daily thermal transients due to the damping effect of the oil-to-salt heat exchangers. A

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survival hypothesis for carbon steel tanks is discussed in detail in Section 4.3 of Volume 3 - Narrative.

- Also, in contrast to the steam generators in central receiver projects, problems with the Therminol steam generators in trough projects are considerably less prevalent. This is likely due to the following effects:
 - Lower rates of temperature change during startup and shut down, as noted above
 - No requirement for the blending of cold Therminol with hot Therminol during the startup and the shutdown of the steam generator
 - A much lower freezing point for Therminol (12 °C) than for nitrate salt (220 °C).

5.2 *Equipment Risks*

For the purposes of the Design Basis study, the risk evaluations are developed from estimates of the daily probability of the event occurring, multiplied by the number of forced outage days to affect repairs or correct the problem. Potential mitigation or maintenance responses are included in the evaluation. The product of probability and outage is then multiplied by a factor of 330 to account for the number of operating days each year to calculate a value for the annual risk.

Equipment risks for the components in the thermal storage system are summarized in Table 5-2 through Table 5-7 below. It can be noted that there are essentially no public data on the reliability and the availability of salt equipment and salt systems in commercial projects. This is due to 1) a limited number of commercial projects, and 2) the projects often treat the data as proprietary for commercial purposes. The figures in the tables were largely developed on an ad hoc basis, relying on incomplete information from a limited set of projects. As additional information becomes available, the values in the tables should be reviewed and revised as needed.

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Table 5-2 Equipment Risks Associated with the Salt Storage Tanks - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt leakage	0.0005	250	41	1
Foundation settlement	0.0001	250	8	2
Loss of foundation cooling	0.005	1	2	3
Defects in the tank insulation ¹	0.1	0	0	4
Defects in the tank insulation ²	0.0005	250	41	5
Loss of inventory level signal	0.1	0	0	6
Loss of nitrogen ullage supply	0.05	0.1	2	7
Loss of electric salt heaters	0.01	0	0	8

Notes:

1. Local gap, leading to a local increase in the heat loss.
2. Local gap, exposing the outside of the tank to water. In combination with salt spills on the outside of the tank, the carbon steel shell develops stress corrosion cracking.

Mitigation or Response:

1. Transfer the salt inventory to the companion storage tank. Operate the steam generator directly from the collector field while repairs are made to the tank which is leaking.
2. Remove the tank floor, lift the tank, remove and replace the foundation, lower the tank, and replace the floor.
3. Move the electric power supply to the redundant fan. The thermal inertia of the foundation will prevent excessive soil temperature during the period required to start the redundant fan.
4. Repair the defects when discovered. This is to avoid thermal gradients in the shell, which have the potential to establish stresses equal to the yield value.
5. The risk is discussed below in Section 5.2.1.
6. Use the redundant level instrument(s) until repairs are made to the failed instrument.
7. Isolate the tank to limit changes in the inventory level. The tank pressure / vacuum relief valves protect the tank until repairs can be made to the nitrogen gas supply.
8. Replace the failed element(s) in the salt heaters. The temperature of the tank will not meaningfully decrease in the time required to repair the heater.

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Table 5-3 Equipment Risks Associated with the Oil-to-Salt Heat Exchangers - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Leakage leads to Therminol vapor carried to storage tank	0.05	0	0	1
Leakage leads to Therminol vapor binding of heat exchanger	0.003	10	10	2
Leakage leads to excessive Therminol vapor accumulation in salt tank	0.003	10	10	3
Loss of salt flow meter signal for process control	0.1	0	0	4
Salt freezing in the heat exchangers	0.001	2	1	5

Mitigation or Response:

1. Condense Therminol vapors vented from the tank until the leak can be repaired during a scheduled outage.
2. Continuously vent the salt-side of the heat exchanger until the leak can be repaired.
3. Condense Therminol vapors vented from the tank until the leak can be repaired.
4. Switch process control from flow meters (feedforward) to thermocouples (feedback). Some loss in heat exchanger fatigue life, or daily energy supplied to the hot tank, can be expected.
5. Thaw the heat exchanger using energy from the collector field at a controlled temperature. Necessarily leads to a loss of electric energy production during the thawing period.

Table 5-4 Equipment Risks Associated with the Salt Pumps - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Loss of cold salt pump	0.0005	2	0.5	1
Loss of hot salt pump	0.0005	2	0.5	1

Mitigation or Response:

1. Replace the pump with the warehouse spare.

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Table 5-5 Equipment Risks Associated with the Salt Valves - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Sticking salt pump discharge isolation valves ¹	0.01	2	7	1
Sticking salt pump discharge isolation valves ²	0.001	2	1	1
Sticking salt pump minimum flow recirculation valves ¹	0.01	2	7	1
Sticking salt pump minimum flow recirculation valves ²	0.001	2	1	1
Sticking heat exchanger vent and drain valves ^{1,3}	0.005	3	5	2
Sticking heat exchanger vent and drain valves ^{2,3}	0.0005	3	1	2
Valve stem leakage past conventional stem packings	0.01	5	17	3
Internal isolation valve leakage ⁴	0.001	5	2	2

Notes:

1. Failure rate with conventional valve stem packings.
2. Failure rate with bellows valve stem seals.
3. In conjunction with Therminol leakage rates which are high enough to adversely affect the flow distributions on the salt side; i.e., vapor binding.
4. Internal leakage prevents accurate control over heat exchanger temperature during transient conditions.

Mitigation or Response:

1. Drain the heat exchangers on the salt side, repair the valves, preheat the heat exchangers using energy from the collector field, and refill the heat exchangers on the salt side.
2. Drain the heat exchangers on the salt side, repair or replace the valves, preheat the heat exchangers using energy from the collector field, and refill the heat exchangers on the salt side.
3. Drain the piping, replace the stem packing, and replace any insulation or heat trace cables that have been damaged by exposure to the salt.

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Table 5-6 Equipment Risks Associated with the Salt Instruments - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Inaccurate, or sporadic loss of, flow meter readings	0.05	0	0	1
Salt freezing in the pressure instruments	0.05	0.5	8	2
Inaccurate, or loss of, temperature readings	0.01	0.5	2	3

Mitigation or Response:

1. Switch to manual operation until output signals from flow meters are corrected.
2. Repair the insulation and/or the heat tracing on the instrument stubs.
3. Replace the thermocouple, or switch to the redundant dual-element connections.

Table 5-7 Equipment Risks Associated with the Heat Tracing - Organic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt freezing in the heat exchangers ¹	0.001	3	1	1
Salt freezing in the pressure instruments	0.05	0.1	2	2
Salt freezing in the heat exchanger vent and drain lines	0.01	2	7	3
Damaged insulation and marginal heat trace capacity result in lines, when frozen, will not thaw ²	0.01	3	10	4

Notes:

1. System level failure of the heat tracing during an extended outage period.
2. Problems primarily associated with the small diameter lines, such as the heat exchanger vent lines, the heat exchanger drain lines, and the instrument stubs.

Mitigation or Response:

4. Thaw the heat exchanger using energy from the collector field at a controlled temperature. Necessarily leads to a loss of electric energy production during a portion of the thawing period.
5. Repair the insulation and/or the heat tracing on the instrument stubs.
6. Repair the insulation and/or the heat tracing on the vent and drain lines.
7. Repair the insulation and/or the heat tracing on the salt lines, followed by a waiting period for thawing.

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A review of the annual product values shows that the major risks have, to a first order, a value greater than 10 and the minor risks have a value less than 10. Clearly, the selected crossover value of 10 is somewhat arbitrary, but a review of the risks for trough projects with organic fluids, trough projects with inorganic fluids (Section 6), and central receiver projects (Section 4 in Volume 2) show a similar demarcation between major risks and minor risks. Interestingly, and reassuringly, the risks identified as major have shown to be the most problematic in commercial projects.

The largest risks to the equipment items are summarized below in Table 5-8. Detailed discussions of each of the 5 principal risks are presented in the sections which follow. It should be pointed out that all of these known issues may be adequately addressed by implementing sound engineering solutions and best practices.

Table 5-8 Summary of the Highest Risks - Organic Fluids

Equipment Item	Risk	Annual Product
Thermal storage tanks	Leakage due to weld failures, low cycle fatigue, or transient stresses	41
Thermal storage tanks	Leakage due to stress corrosion cracking in the cold salt tank	41
Oil-to-salt heat exchangers	Leakage leading to Therminol vapor binding	10
Oil-to-salt heat exchangers	Leakage leading to excessive Therminol vapor accumulation in salt tank	10
Salt valves	Valve steam leakage past conventional stem packings	17
Heat tracing	Damaged insulation and marginal heat trace capacity result in lines, when frozen, cannot thaw	10

5.2.1 Thermal Storage Tanks

There are 34 parabolic trough projects with thermal storage in commercial operation. This represents 78 salt storage tanks in commercial service. (The Solana project uses 12 tanks.)

A significant risk to the availability of a project is a failure of either the cold salt or the hot salt tank. Should either tank fail, the Rankine cycle is restricted to operating only with thermal energy supplied directly from the collector field. In effect, that portion of the collector field which was purchased and installed to charge the storage system cannot be used.

Should a leak develop, the repair period is a lengthy one, due to the following effects:

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- The tank and the foundation must be cooled to essentially ambient temperature prior to personnel entering the tank and analyzing the failure. The cooling period is on the order of 7 to 10 days due to the large thermal mass of the foundation, and due to a requirement to control intra-tank temperature differentials to values low enough ($< 30\text{ }^{\circ}\text{C}$) to prevent high internal stresses
- The source of the tank failure must be identified. This may include removal of tank sections, and laboratory examinations for items such as corrosion, low cycle fatigue fractures, or weld defects. The time required for laboratory examinations is on the order of one to perhaps a few weeks
- Repairs may involve a wide range of possibilities and conditions. Independent, potentially unrelated activities and associated schedules, include:
 - Procurement and delivery of replacement tank sections: At least several weeks
 - Removal of defective welds and rewelding: One to several weeks
 - Lifting the tank and replacing the foundation insulation: Two to several months
 - Post weld heat treatment of the welds in the thick metal sections: At least one week
- Once repairs are complete, the tank must be preheated prior to refilling with salt. The preheat period is on the order of 7 to 10 days.

The overall outage will last at least 3 months, and could extend to periods as long as 8 months.

For a representative commercial project, the monthly revenues are on the order of \$4,000,000. If the plant is not in operation for, say, 8 months, then the cost to repair the tank plus the loss in revenue could be \$35,000,000. Further, if the plant is not supplying the full complement of electric energy to the local utility, then the project may be considered partially in default regarding the power purchase agreement.

Tank Failures

In commercial trough projects using organic heat transfer fluids, the design collector field temperature ($393\text{ }^{\circ}\text{C}$) is low enough that both the cold tank and the hot tank can be fabricated from carbon steel. As such, the tanks will avoid some of the problems seen with stabilized steel hot tanks, such as stress relaxation cracking, in central receiver projects. Nonetheless, the tank designs in trough projects are nominally the same as the tank designs in central receiver projects, as follows:

- All of the tanks are flat bottom designs, with self-supporting dome roofs. The designs are based on API Standard 650, Welded Steel Tanks for Oil Storage, in combination with allowable material stresses from Section II of the ASME Boiler and Pressure Vessel Code
- All of the tanks operate at atmospheric pressure

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- All of the tanks are supported on insulated foundations
- All of the tanks are preheated by the circulation of combustion gases from a fossil-fired air heater.

Yet, the failure rates of cold salt tanks and hot salt tanks in parabolic trough projects, and the failure rates of cold salt tanks in central receiver projects, are at least an order of magnitude lower than the failure rates of hot salt tanks in central receiver projects. The reasons for the difference in the failure rates are not fully known, but the following elements may contribute to an explanation:

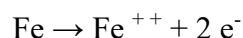
- Residual welding stresses in the floor reach, and often exceed, yield values. The high stresses lead to plastic deformations of the plates, and the floor has, in essence, developed shallow buckles. The completed floor is a large thin sheet, and its continued resistance to buckling due to compression forces associated with radial temperature gradients, or due to friction loads, is generally low. When the tank is preheated, or when the tank is placed into commercial service, the residual welding stresses start to relax. The relaxation process likely occurs slowly, as the inventory temperatures are well below typical post weld heat treatment temperatures. However, the creep resistance of carbon steel is lower than the creep resistance of stainless steel. As such, the relaxation of the deformations in the floor of a carbon steel tank may be more pronounced than the relaxation of the deformations in the floor of a stainless steel tank. This, in turn, may make the floor of a carbon steel tank more resistant to buckling once commercial service begins
- In central receiver projects, the rate of temperature change in the downcomer can reach values as high as 360 °C/min. The hot salt tank will experience some fraction of this rate depending on 1) the depth of the inventory during the transient, 2) the cold tank/hot tank crossover temperature selected by the operator, and 3) the time required to move the downcomer diversion valves from open to close (or from close to open). The cold tank will also be involved in these transients. However, the level of the inventory in the cold tank, at least during morning startup, is usually much higher than the level of the inventory in the hot tank. This has the effect of damping transient effects in the cold tank. Also, the temperature of the salt entering the cold tank is always higher than the temperature of the inventory. As such, buoyancy effects carry the salt entering the tank away from the floor, which reduces the potential for local compressive forces on the floor
- In parabolic trough projects, the temperature difference between the hot side and the cold side is a nominal 100 °C. In contrast, the temperature difference in central receiver projects is on the order of 275 °C. The smaller temperature difference likely imposes smaller transient effects on the hot tank in a trough project than the hot tank in a central receiver project. The smaller transient effects result in lower transient stresses in the floor

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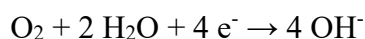
- Also in parabolic trough projects, the oil-to-salt heat exchanger is connected to the hot Therminol headers of the collector field, and the hot salt tank is connected to the salt side of the oil-to-salt heat exchangers. The thermal inertia of the field Therminol headers, plus the approach temperature at the hot end of the heat exchanger, moderate thermal transients seen by the hot tank.

In light of the above, the failure mechanism(s) of cold tanks may be different than the failure mechanisms of the hot salt tanks in central receiver projects. A limited number of failures in cold tanks, in combination with limited information on the source of the failures, implies that failure of several of the salt tanks in a commercial project may be related to stress corrosion cracking.

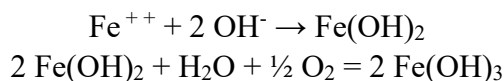
Stress corrosion cracking is a combined mechanical and electro-chemical corrosion process which results in cracking of certain materials ⁴. The mechanism is not simply an overlapping of corrosion and mechanical stresses, but an auto-catalytic, self-accelerating process leading to high metal dissolution rates. The process includes an oxidation reaction:



The oxidation reaction is accompanied by a reduction reaction, typically involving oxygen and water:



Further reactions can then occur, including:



The hydrated oxides can lose water during dry periods, and revert to anhydrous oxides, such as ferrous oxide, FeO, and ferric oxide, Fe₂O₃ (hematite).

Initially, a small pit is formed at the surface, which then develops into a crack due to applied or residual stresses in the material. The crack formation opens up a new active (non-passive) metal surface, which corrodes very easily. This leads to further crack propagation and exposure of new active metal surfaces in the crack. Metal dissolution in the crack advances rapidly until mechanical failure occurs. Brittle failure can occur in normally ductile metals at stress levels which are less than the yield strength. Internal stresses in a material due, for example, to intra-tank temperature gradients can be sufficient to initiate stress corrosion cracking.

⁴ Corrosion in Carbon Steels – IspatGuru

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Stress corrosion cracking of carbon steel is rare compared with stress corrosion cracking of stainless steel. Nonetheless, one potential source of cracking in carbon steel is the exposure of the metal to liquid water in the presence of nitrates. One hypothetical failure mechanism is as follows:

- The outside of the tank is exposed to nitrate salt. The source of the salt could be 1) vapor condensation from ullage gases passing through a vacuum / pressure relief valve, or 2) salt leakage from pumps, valves, instruments, or piping above the tanks
- Water comes into contact with the outside of the tank. The source of the water could be 1) site flooding during a particularly heavy rain, 2) rain water entering gaps in the tank insulation, or 3) spray water from pressure washers to remove residual salt
- As noted above, process conditions (i.e., hydrostatic loads) can produce stresses which are within the ASME allowable values but still high enough to initiate cracking. In addition, should water come into contact with the tank, local heat transfer will produce local temperature gradients, which, in turn, will generate local thermal stresses. Temperature gradients on the order of 45 °C can produce stresses equal to the ASME allowable value, and a gradient of approximately 115 °C can cause the material to yield.

Should the above combination of conditions occur, then a failure due to stress corrosion cracking may be the result.

Recommendations

Low Cycle Fatigue

No failures of cold salt or hot salt tanks in parabolic trough projects, similar to those experienced in the hot salt tanks of central receiver projects, have been reported. However, this is not a demonstration that the failure mechanisms of the hot tanks in central receiver projects are not being duplicated in the cold salt and the hot salt tanks in parabolic trough projects. Stated another way, the salt tanks in trough projects might simply be failing more slowly than the hot salt tanks in central receiver projects. To determine if this is the case, the failure analyses currently being conducted for the hot tanks in central receiver projects should be extended to the tanks in parabolic trough projects. The analyses could lead to one of two results:

- A confirmation that the salt tanks can survive the life of the project.
- An indication that the expected life of the tanks will not be achieved. In this case, recommendations can be developed as to how the tanks should be operated to extend the life of the tanks.

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Stress Corrosion Cracking

In a commercial project, it is unlikely that salt will never come into contact with the outside of the storage tank. If the source of stress corrosion cracking is liquid water in contact with nitrate salt, then it will be the responsibility of the maintenance personnel to perform the following:

- Ensure that no defects develop in the tank insulation, or in the aluminum jacket covering the insulation, that could expose the exterior of the tank to liquid water
- Remove salt from metal surfaces only by abrasive methods or by media blasting.

5.2.2 Oil-to-Salt Heat Exchanger

Operating Limits

There are 34 parabolic trough projects with thermal storage in commercial operation. This represents about 200 oil-to-salt heat exchangers in service.

Almost by definition, the heat exchanger operates through a daily startup cycle. To ensure that the heat exchanger has a fatigue life consistent with the life of the project, the vendor will specify limits on rates of temperature change and thermal shock. Typical values include an allowable rate of temperature change of 10 °C/min and an allowable thermal shock (difference between the temperature of the fluid entering the heat exchanger and the metal temperature at the entrance to the heat exchanger) of 60 °C. Typically, one thermal shock per day is allowed. The vendor will also specify the minimum flow rate required to provide uniform flow distributions on both the shell- and the tube-sides. A representative value is 16 percent of the design flow rate; i.e., a turndown ratio of 6:1.

Process Design

Ideally, the process design would include a series of mechanisms to control the transient conditions in the heat exchanger:

- During hold periods, supply cold salt to the cold end of the heat exchanger. Cold salt leaving the hot end of the heat exchanger is returned to the cold tank
- At the beginning of a charge cycle, blend cold Therminol from the discharge of the Therminol pumps with hot Therminol from the collector field at a Therminol mixing station at the hot end of the heat exchanger. This would control the rate of temperature change in the transition from Hold to Charge.

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- At the end of a discharge cycle, blend cold salt from the cold salt pump with hot salt from the hot salt pump at a salt mixing station at the hot end of the heat exchanger. This would control the rate of temperature change in the transition from Discharge to Hold.

However, most commercial projects adopt a less complex process design, and do not provide a means for cold salt recirculation, a Therminol mixing station, or a salt mixing station. Instead, the process design relies on the following:

- The ability of the heat exchangers to accept some level of thermal shocks each day
- Adjusting the speed of the cold salt pump, or adjusting the speed of the hot salt pump, in a feedback loop to try and match the allowable rates of temperature change.

As might be expected, some projects are more effective at controlling low cycle fatigue damage to the heat exchangers than other projects. Nonetheless, equipment failure rates in some projects are higher than projected in the financial models ⁵.

Consequences of Leaks

Leaks in the heat exchangers can lead to a series of problems:

- The Therminol passes to the salt side of the heat exchanger, flashes to a vapor, and then collects in the upper portion of the shell. The vapor can restrict salt flow in this area, which can lead to 1) a reduction in the duty of the heat exchanger, or 2) an undesirable flow distribution which, in turn, leads to an undesirable temperature distribution and the development of intra-heat exchanger thermal stresses associated with self-restraint
- The Therminol passes to the salt side of the heat exchanger, flashes to a vapor, and is transported to the ullage space in one of the salt tanks. Over the course of a complete charge / discharge cycle, a portion of the ullage gas mixture of nitrogen and Therminol passes to the Therminol treatment system, where a portion of the Therminol is condensed for reuse. However, equilibrium considerations prevent the recovery of all of the Therminol, and some of the Therminol is lost to the environment through the vents in the treatment system. Replenishment of the Therminol represents an operating expense to the project
- When the leakage rate exceeds an acceptable value, the heat exchanger must be repaired. This involves draining the heat exchanger train, cooling the heat exchangers to ambient, identifying and plugging the tubes which are leaking, preheating the heat exchangers using Therminol from the collector field, and refilling with system with salt. The time required to do so is at least

⁵ Aalborg CSP, TEMA SGS/HX U-Tube Type Inspected and Repaired / Replaced by Aalborg Service Department, 2019

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1 week. The expense in a representative commercial project, for both the repairs and the loss in revenue, is on the order of \$1,000,000.

Source of Leakage

If the oil-to-salt heat exchanger is a shell and tube design, as described in Section 11.8.1, then leaks in the heat exchanger can generally be traced to the following sources:

- The friction (i.e., tube expansion) connections between the tube and the tubesheet can relax under daily thermal cycles. Some fraction of the stresses producing relaxation are due to the different thermal response times of the (thin) tubes and the (thick) tubesheet
- The strength welds between the ends of the tube and the face of the tubesheet can act as stress concentration points. Under transient conditions, the end of the welds and the face of the tubesheet will have different thermal response times, which can promote low cycle fatigue damage in the weld region.

Heat exchanger leakage allows the (relatively high pressure) Therminol to leak into the (relatively low pressure) salt side.

If the oil-to-salt heat exchanger is a welded flat plate design, then the leaks are typically due to fatigue cracks developing in the perimeter weld that connect the edges of the plate to form a stack.

In both cases, the problem generally results from a process control that does not accurately match, during transient conditions, the duty and the temperatures on the Therminol side with the duty and the temperatures on the salt side.

Recommendations

The process design should ideally include both a Therminol mixing station and a salt mixing station at the hot end of the heat exchanger. The mixing process is intended to:

- Limit the rate of temperature change to the vendor specification of 10 °C/min
- Limit the number and the magnitude of the daily thermal shocks to the vendor specifications of 1 and 60 °C, respectively
- Provide a uniform heat exchanger temperature of 295 °C during hold periods, which provides a consistent and a repeatable condition for daily startup and in response to lengthy cloud transients.

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The goal of the process design is for the heat exchanger to meet, or exceed, the projected low cycle fatigue life of the equipment.

For shell and tube heat exchangers, the process design can be aided by selecting a fabrication approach that is based on all-welded construction; i.e., header/coil, or internal bore welded.

5.2.3 Salt Valves

Stem Packing Materials

A common commercial stem sealing material is graphite. The material is soft, which allows it to readily conform to the shape of the stem. Also, in the absence of an oxidizing environment, it is thermally stable. However, in salt service, the salt oxidizes the graphite to CO₂, and the packing eventually disappears.

At the Solar Two project, the recommended stem packing consisted of alternating layers of 1) Teflon washers, and 2) Inconel wire braids impregnated with graphite. Over time (weeks to months), the graphite in the braid oxidized, and the stem packings required periodic replacement of the Inconel braids. In general, this combination of packing materials was judged to be marginally acceptable.

In subsequent commercial projects, other stem packing materials have been used, including various fluoropolymers, mica, and vermiculite. In some cases, the packing performed as intended, and in other cases, the packing developed leaks almost immediately. In general, a robust stem packing material, with a longevity measured in years, has yet to be identified.

Stem Leakage

The consequence of a leak, other than creating a spill, is 1) saturation of the insulation on the valve and the adjacent piping, and 2) corrosion, and subsequent, failure of the heat trace cables on the valve bonnet, the valve body, and the adjacent piping. Both consequences can lead to frozen salt in the valve or in the adjacent piping.

Bellows Stem Seals

To this end, the only stem seal which has been shown to be free of leakage is a bellows seal. This, in turn, implies that the preferred valve types are limited to those which can use a bellows seal; i.e., valves with translating stems, such as globe and angle plug valves.

As might be expected, this approach has met with some resistance in commercial projects:

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- A bellows stem seal is much more expensive than a packing stem seal. Budgetary estimates from a study conducted for DOE indicate that adding a bellows stem seal can triple the cost of a valve
- For isolation service, large globe valves are more expensive than triple offset butterfly valves. (As a technical aside, gate valves and ball valves are often used for isolation. However, nitrate salt oxides all metals, including Stellite. If a gate valve or a ball valve in salt service remains in the closed position for even one to two days, corrosion between the plug and the seat can render the valve permanently immovable. In general, the only types of valves suitable for salt service are ones in which the plug moves in a direction perpendicular to the seat; i.e., globe valves and triple offset butterfly valves.)
- The bellows may, under specific sets of circumstances, rupture. These circumstances include moving the valve with frozen salt in the bellows, or exposing the bellows to high transient pressures generated by a leak in a heat exchanger.

Using bellows seals on valves will increase the capital cost of the project. However, the cost penalty can be reduced by developing process designs that minimize the use of valves. One example is the piping arrangement for storage tank recirculation heaters, discussed in the Subsection titled *Electric Salt Heaters in Minimum Flow Recirculation Lines* in Section 6.6.1, Process Design, in Volume 3 - Narrative. The cost penalty can also be offset by an increase in plant availability. Leakage past the valve stem seals exposes the heat trace cables on the valve body and on the adjacent piping to salt. The cables operate at temperatures which are high enough ($> 650\text{ }^{\circ}\text{C}$) to decompose the salt. Several of the salt decomposition products are various oxides, which aggressively corrode the outer metal covering on the heat trace cables. Corrosion lifetimes are often measured in weeks. Once the covering corrodes, the internal heating cables are exposed to moisture, and can fail in a matter of days. Failed cables lead to immovable valves and frozen salt piping, which can necessitate forced outages lasting hours to days. A project economic analysis will often show that an improvement in plant availability and revenue, due to a properly functioning heat trace system, justifies the marginal expense for bellows stem seals.

In one commercial project, the salt control valves in the thermal storage system used bellows stem seals. The valve vendor supplied an engineered insulation enclosure for the valve body and the bellows region. The enclosure included the requisite heat trace cables and control thermocouples. Operating experience from the commercial project shows that the valves, the bellows, and the insulated enclosures are operating at expected, with no stem sealing failures due to frozen salt in the bellows.

5.2.4 Heat Tracing

Heat tracing is an expensive item, and there is an economic incentive to install as little heating capacity as possible. A number of commercial projects have adopted, as a design criterion, a heat trace capacity

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sufficient to provide a maintenance temperature of 275 °C. However, any damage to, or degradation in, the insulation can lead to local freezing and an unscheduled shutdown to thaw the equipment.

Design Criteria for Commercial Projects

A heat trace design criteria based, in part, on maintaining a metal temperature of 275 °C at the minimum ambient temperature is, in effect, an infinite preheat period. As an example, a DN300 (12 in.) line with 125 mm (5 in.) of calcium silicate of insulation requires a heat input of 185 W/m to maintain a temperature of 275 °C at an ambient temperature of 10 °C. However, in a commercial project, the pipe insulation is subject to damage due, for example, to maintenance personnel walking on the pipe. If the insulation is compressed by 25 mm (1 in.), then the local thermal resistance of the insulation is reduced, and the steady state temperature which can be maintained by the heat trace drops to 232 °C. Similarly, the insulation can develop gaps due to daily thermal expansion and contraction cycles. If the local effective thermal conductivity of the insulation increases by 20 percent, then the steady state temperature which can be maintained by the heat tracing falls to 230 °C. In either situation, if stagnant salt is present in the line, then local freezing can be expected. Further, the heat tracing can no longer raise the pipe metal temperature above the freezing point, and the line will remain plugged until repairs are made to the insulation. When the plant is new, the pipe insulation will provide the required thermal resistance, and the heat trace system will be judged to be adequate. However, as the plant ages, the pipe insulation will degrade, and availability problems with frozen salt lines will start to appear.

Recommendations

A more robust design criterion would be based on a preheat period of, for example, 12 hours. Continuing with this example, preheating the pipe to 275 °C in 12 hours requires a heat input of 354 W/m, or nominally double the heat input if the preheat period is infinite. With a heat input of 354 W/m, the steady state temperature of the pipe increases from 275 °C, as above, to a new value of 516 °C. Further, if the insulation is compressed by 25 mm, then the steady state pipe temperatures can, in principle, be raised as high as 435 °C. Similarly, if the effective thermal conductivity of the insulation increases by 20 percent due to gaps in the insulation, then the pipe temperatures can be raised to a theoretical value of 430 °C. It's unlikely that the duty cycles of the heat trace system would be set so high as to produce pipe temperatures as high as 430 °C, but the available heating capacity would be sufficient to both prevent freezing and to thaw a line should freezing occur. As such, the heat trace system will be adequate both when the plant is new and after the plant has accumulated several years of commercial use.

The selection of commercially viable preheat times will only be made by an Owner, or an engineering contractor, who has already been burdened by trying to operate an inadequate heat trace system. Any heat trace capacity above that required to maintain a temperature of 275 °C would be supported by an understanding that the improvement in availability and revenue would more than offset the increase in the capital cost.

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Also, in the interests of improving the reliability of the heat tracing system, it can be noted that the heating elements inside the heat trace cables are prone to failure if the elements are exposed to water vapor. As the cables operate under cyclic thermal conditions, brazing processes that use dissimilar metals are not likely to be as reliable as laser welded designs using autogenous welds.

5.3 *Project Risks*

The discussion above, in Section 5.2, evaluated risks associated with the salt equipment. However, there are also risks at the project level which can influence the availability of the plant. Two such risks are presented in Table 5-9.

Table 5-9 Project Risks - Organic Fluids

	Annual probability, fraction	Annual outage duration, days	Annual product	Mitigation or Response
Process design is not consistent with the performance requirements in the project financial model	0.5	20	10	1
Operating personnel do not have an understanding of the plant that is sufficient to prevent damage to the equipment	0.5	20	10	2

Mitigation or Response:

1. Research the process design in other commercial projects. Copy those features which are successful, and avoid those features which are not effective.
2. Provide sufficient funds to attract, and retain, a mechanical engineer at the site for the first 5 years of commercial service. Provide ad hoc training to the operators in topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

5.3.1 **Process Design**

It can be noted that one the primary goals of the Design Basis Document is to identify the risks of, and specifications for, nitrate salt equipment. Strictly speaking, process design is not the subject of an equipment specification. Nonetheless, some commercial projects have failed to meet their expected levels of availability due to various aspects of the process design. The topic of process design is included in this report because the process design can have as large an influence on the plant availability as a leak in a heat exchanger.

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Process design elements include items such as the following:

- Ensure that the vendor requirements on rates of temperature change, thermal shock, minimum flow rate, and low cycle fatigue can be satisfied. The requirements are generally met through the selection of piping arrangements, pump specifications, valve configurations, valve specifications, control inputs, and control outputs
- Provide sufficient margin on the heat trace capacity to accommodate the inevitable degradation in heating capacity associated with failed cables and damage to the insulation
- Provide an approach to process control that avoids the use of instrument readings that are inaccurate or problematic (pressure, flow) and adopts the use of instrument readings that are accurate and repeatable (temperature, pump speed, valve position)
- Select equipment configurations (materials, fabrication techniques, thermal expansion allowances) that can tolerate some level of operator error
- Provide additional fatigue life for the equipment, above and beyond that required for the life of the project, to accommodate damage to the equipment during commissioning and the first year of operation when the control logic is being developed and refined. Identification of the transient conditions, and the determination of associated thermal stresses, will be difficult to predict. However, even a first-order analysis will often confirm whether a design should last for expected lifetime of the project.

An example of a process design to satisfy the vendor limits on the low cycle fatigue life of the oil-to-salt heat exchangers is discussed above in the Section 5.2.

In general, the process design must provide system and equipment capabilities and capacities that are beyond that required in the ideal plant. The ideal plant is one that is commercially mature, with a proven control system and skilled operators. Stated another way, salt systems and process design have yet to reach full commercial levels. As such, errors in the process design, or mistakes made on the part of the operators, will subject the equipment to damage rates that are not consistent with a 30-year plant life. To compensate, early projects will need to be designed for a life of, say, 50 years to reach the 30-year life projected in the financial models.

5.3.2 Personnel Risks

As with process design, operating personnel are not the subject of an equipment specification. Nonetheless, some commercial projects have failed to meet their expected levels of availability due to mistakes made, and aggressive operating approaches taken, by the operating staff. The topic of

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personnel risks is included in this report, as the actions taken by an operator can have as large an influence on the plant availability as the process design features noted above.

Project Complexity

Parabolic trough plants with thermal storage systems are complex projects, and require an operating staff that has a comprehensive understanding of the equipment and the temperature limitations. Further, parabolic trough projects necessarily operate through daily thermal cycles, and the equipment can incur excessive fatigue damage if it is operated outside of the limits set by the vendor. As the complexity of the project increases, the depth of knowledge required by the operators also increases. Some examples of the complexity of the project include the following:

- The oil-to-salt heat exchangers are directly hydraulically coupled to the collector field. Further, in most projects, there is no mechanism for attenuating the temperature of the Therminol entering the heat exchanger (hot end or cold end), or the temperature of the salt entering the heat exchanger (hot end or cold end). As such, all of the transient conditions in the collector field can be imposed on the oil-to-salt heat exchangers
- The heat exchangers in the storage system are large equipment items. Further, the tube-to-tubesheet connections, which can number up to 1,500 in each of 6 series heat exchangers, typically rely on tube welding and tube rolling to seal the high pressure fluid from the low pressure fluid. During startup and shutdown, the thermal transient experienced by the collector field can result in rates of temperature change at or beyond the vendor limit, leading to high values of local thermal stresses, flexing of the tubesheet, relaxation of the tube-to-tubesheet connections, and internal leakage
- Therminol circulation in the collector field occurs whenever the collector field is not in operation; i.e., during overnight hold, during cloudy weather, or when wind speeds are high enough to stow the collector field. This ensures that the temperature of the Therminol in the field headers is nominally the same as the temperature of the Therminol in the heat collection elements. In this way, thermal shocks to the heat collection elements, which have the potential for permanently damaging the receiver tubes, can be avoided. The temperature of the Therminol can decay to values as low as perhaps 100 °C during non-operating periods. Similarly, during non-operating periods, Therminol circulation is maintained in the power block area. However, with salt present in the heat exchangers, the temperature of the Therminol must be maintained at values of at least 275 °C. As such, there are two circulation loops; one for the collector field, and a second for the thermal storage system. The two loops are separated by isolation valves. However, isolation valves do not always provide perfect isolation, and a situation can arise in which cold Therminol from the collector field loop mixes with warm Therminol in the thermal storage loop. Due to the large piping diameters in commercial projects, cold Therminol can flow along the bottom of the warm Therminol lines due to thermal stratification. The cold Therminol,

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on entering the oil-to-salt heat exchangers, can establish temperature gradients, which in turn can establish stress gradients, which are large enough to permanently damage the heat exchangers

- Heat trace cables can produce outer sheath temperatures in excess of 650 °C. (Design criteria for heat trace systems are discussed in Section 9.8.) On carbon steel equipment, inaccurate control over the duty cycles of cables can lead to graphitization of the steel. Graphitization is the separation of graphite from iron, and the subsequent formation of graphite nodules at the grain boundaries. The resulting structure is brittle, with a low strength and a high susceptibility to failure
- The salt has a nominal melting point of 230 °C. To prevent the salt from freezing, every surface in contact with salt must be electrically heat traced. As might be expected, heat trace systems are expensive, and there is a financial incentive to keep the heat trace capacity as small as possible. Under these conditions, the heat flux provided by the heater cables is on the order of 120 to 130 percent of the steady state heat losses from equipment. As a result, if the thermal insulation becomes moderately damaged, or if small (½ in.) gaps open in the thermal insulation, then the heat trace can no longer maintain the equipment above the salt melting point. The result is frozen salt and plugged equipment. Further, the heat tracing does not have the capacity to overcome heat losses which are higher than design values, and it is not possible to thaw the frozen equipment. To return the equipment to service, the insulation must be repaired. This is followed by a waiting period, which can last from several hours to several days, for the salt to thaw
- Flow instruments in salt service often show differences of 10 to 20 percent between the measured flow and the actual flow. The difference, in some cases, may be due to presence of water in the salt (produced by a leak in the steam generator) and cavitation downstream of the measuring device. Pressure instruments can have better accuracies, but the readings are very sensitive to the quality of the heat trace and the insulation on the instrument standoffs. Flow and pressure measurements are often inputs to the control logic of the distributed control system. But if the flow and pressure readings are inaccurate, then the process control outputs that determine pump speeds and valve positions can result in heat exchanger rates of temperature change that exceed vendor limits by significant margins. If the output signals from the distributed control system lead to oscillations, or lead to other conditions which might damage the equipment, the operators will switch to manual control. The skills of the operator can then have an influence on the fatigue life of the equipment.

Operator Duties

The control logic is initially developed for the plant in an as-new condition. For example, the heat exchangers are in a clean condition, the pump impellers have experienced no wear, the isolation valves

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seal properly, and the heat tracing on the pipe provides a uniform temperature over the length of the zone.

Naturally, these condition do not last throughout the life of the project. To a first order, the control system will respond to items such as heat exchanger fouling or pump wear by adjusting parameters such as valve positions or pump speeds. However, there are other aspects of equipment changes or degradation that the control system cannot recognize, and therefore cannot provide compensation. These changes include items such as isolation valve leakage during hold periods, local degradation of the pipe heat insulation leading to the formation of small frozen salt plugs, drift in the pressure instrument readings, and backflow through check valves. Since the control system cannot respond to these conditions, it is then incumbent on the operator to do the following:

- Recognize the degradation in the system or the equipment
- Determine the source of the degradation
- Develop an approach to compensate for the degradation
- Revise the logic in the distributed control system, or revise the transitions between operating states to include a mixture of automatic and manual steps. One example is the receiver fill procedure discussed in Subsection *Fill* in Section 7.1.3, Daily Operation Sequence, of Volume 2 - Specifications for Central Receiver Projects. Specifically, failures of the backwall thermocouples on the absorber panels give an incomplete picture of the uniformity of the solar preheating prior to filling. As a result, the operators must make a decision each day as to when, or if, to fill and start the receiver.

Staffing Approach

The standard approach to staffing a project is to define 1) the personnel roles, 2) the number of personnel in each role, and 3) the salary budget for each role from the financial model on which the project was developed. Requisitions for each role are prepared, interviews are conducted, staff members are hired, and training is conducted to reflect the specifics of the project. However, the costs for attracting skilled personnel to the projects, many of which are located at remote sites, may be inconsistent with the operation and maintenance costs adopted for the project financial model to meet various investment targets. Also, the quality of the operators often spans a broad spectrum, and the requisitions cannot guarantee that the person filling a role has the interest, or the ability, to absorb the details necessary to operate the equipment in a manner that does not damage the equipment. This is particularly true if the signals from the process instruments are not reliable or accurate, and the operators are then obligated to make a judgement call based on an understanding of the process and the details of the equipment design.

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Recommendations

An alternate, and likely necessary, approach to staffing the project would be as follows:

- Define the skills needed in each role. For example, an operator may need to have the background of a mechanical engineer, with a basic knowledge of heat transfer, stress analysis, and low cycle fatigue
- Determine the salary required to attract, for example, an engineer to a remote location
- Develop a staffing budget based on the required mix of personnel skills.

The alternate approach is essentially guaranteed to result in an annual staffing cost that is higher, and perhaps much higher, than typically seen in commercial projects. However, if the operating staff can prevent even one instance of equipment damage, then the marginal cost to employ a staff with the required skills will be justified.

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6. Risk Analysis - Inorganic Heat Transport Fluids

6.1 Introduction

As background to the risk analysis, and as discussed above in Section 5.1, an evaluation of the technical, commercial, and operating risks in commercial solar projects was recently conducted by NREL⁶.

The principal technical risks associated with central receiver projects are summarized in Table 6-1. Parabolic trough projects using nitrate salt as the working fluid are not the same as central receiver projects using nitrate salt as the working fluid. However, the two types of projects share a number of similarities in terms of design temperatures (520 to 565 °C), type of thermal storage (direct), and type of steam generator (nitrate salt). As such, many of the risks identified for central receiver projects are applicable to trough projects using nitrate salt.

Table 6-1 Principal Risks for Central Receiver Projects Identified in the Best Practices Report

System	Component	Issue Type	Occurrence	Weight
Power block	Salt steam generator	Steam generator reliability	13	245
Thermal storage	Salt tanks	Tank design	10	204
Power block	Salt steam generator	Steam generator design	8	174
Project	O&M	Heliostat cleanliness	9	123
Heliostat field	System	Design standards	8	122
Receiver	Downcomer	Downcomer design	8	110
Receiver	Salt piping	Heat tracing	8	110
Heliostat field	Mirrors/facets	Heliostat optical quality	9	105
Receiver	Tower	Tower construction	6	100
Thermal storage	Salt tanks	QA/QC	4	100
Power block	DCS	DCS logic	5	89
Heliostat field	System	Heliostat qualification	6	84
Receiver	Salt piping	Valve design	7	77
Receiver	Control systems	Aiming strategy	3	75
Thermal storage	Salt tanks	Tank foundation	5	73
Receiver	Control systems	Automation	4	70
Receiver	Downcomer	Piping support design	3	65
Receiver	Outlet vessel	Outlet vessel design	4	60
Receiver	Salt piping	Valve reliability	4	60

⁶ Mehos, Mark, et. al, (National Renewable Energy Laboratory, Golden, Colorado), 'Concentrating Solar Power Best Practices Study', Technical Report NREL/TP-5500-75763, June 2020

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System	Component	Issue Type	Occurrence	Weight
Project	EPC	EPC execution	3	55
Receiver	Control systems	Receiver reliability	3	55
Receiver	System	Heliostat/receiver integration	5	55
Project	Engineering	Technology scale-up	2	50
Receiver	Receiver	Receiver reliability	7	43
Receiver	Control systems	Infrared camera	3	39
Power block	DCS	Automation	2	30
Project	EPC	Schedule	2	30
Receiver	Tower	Elevator	4	30
Receiver	Receiver	Receiver coating	3	27
Heliostat field	Control	Design specifications	1	25
Heliostat field	Mirrors/facets	Heliostat cleanliness	1	25
Project	EPC	Welding	1	25
Receiver	System	Receiver reliability	3	25
Thermal storage	Salt tanks	Salt heater design	3	25
Heliostat field	Control	BCS calibration	2	24
Heliostat field	Drives	Heliostat availability	4	24
Heliostat field	Mirrors/facets	Facet blocking	2	18
Heliostat field	Power/wiring	Electrical system design	2	18
Thermal storage	Salt	Corrosion	2	18
Heliostat field	Control	Heliostat/receiver integration	1	15
Heliostat field	Drives	Drive qualification	1	15
Power block	Steam cycle	Valve reliability	1	15
Project	Structure	EPC experience	1	15
Receiver	Control system	Flux meter	1	15
Receiver	Receiver	Automation	1	15
Thermal storage	Salt	Water emulsion	1	15
Receiver	Downcomer	Downcomer control	3	13
Thermal storage	Hot salt pump	Pump design	3	13
Heliostat field	Mirrors/facets	Heliostat availability	2	12
Power block	Steam turbine-generator	Turbine reliability	2	12
Power block	Steam cycle	Pump reliability	3	11
Power block	Salt steam generator	Pump alignment	2	10
Power block	Steam turbine-generator	Generator reliability	2	10
Receiver	Receiver	Receiver design	2	10
Heliostat field	Civil	Heliostat cleanliness	1	9
Heliostat field	Environmental	Heliostat flux hazard	1	9
Heliostat field	Heliostat structure	Pedestal installation	1	9

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System	Component	Issue Type	Occurrence	Weight
Heliostat field	Power/wiring	Lightning	1	9
Heliostat field	System	Optics versus cost	1	9
Power block	Auxiliary systems	Water supply	1	9
Power block	DCS	Instrument reliability	1	9
Receiver	Cold salt pump	Pump reliability	1	9
Receiver	Salt piping	Piping design	1	9
Thermal storage	Piping	Insulation quality	1	9
Power block	Auxiliary systems	Fire system design	1	5
Thermal storage	Hot salt pump	Pump reliability	1	5
Heliostat field	Civil	Site preparation	1	3
Power block	Auxiliary systems	Hybrid cooling	1	3
Power block	Electrical	Design specifications	1	3
Receiver	Salt piping	Safety valves	1	3
Thermal storage	Salt tanks	Testing standards	1	3

A review of the table shows the following:

- The salt equipment topics with the highest weight values are the design and the reliability of the salt tanks and the steam generators
- In contrast to the salt tanks in parabolic trough projects, several failures have occurred in the hot salt tanks in central receiver plants. This can partially be attributed to 1) the large temperature difference between the cold tank and the hot tank, and 2) high rates of temperature change seen at the hot tank during receiver startup and following a receiver trip
- Also, in contrast to the steam generators in parabolic trough projects, problems with the salt steam generators in central receiver projects are considerably more prevalent. This is likely due to the following effects:
 - A requirement for the accurate blending of cold salt with hot salt during the startup and the shutdown of the steam generator
 - A higher freezing point for nitrate salt (220 °C) than for Therminol (12 °C).

6.2 Equipment Risks

As with the risk analysis for organic fluids, the risk analysis for inorganic fluids is based on an estimate of the daily probability of the event occurring, multiplied by the number of forced outage days to affect repairs or correct the problem. Potential mitigation or maintenance responses are included in the

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evaluation. The product of probability and outage is then multiplied by a factor of 330 to account for the number of operating days each year to calculate a value for the annual risk.

Equipment risks for the thermal storage system, the salt steam generator, the salt pumps, the salt valves, and the heat tracing are summarized in Table 6-2 through Table 6-7 below. It can be noted that there are essentially no public data on the reliability and the availability of salt equipment and salt systems in commercial projects. This is due to 1) a limited number of commercial projects, and 2) the projects often treating the data as proprietary for commercial purposes. The figures in the tables were largely developed on an ad hoc basis, relying on incomplete information from a limited set of projects. As additional information becomes available, the values in the tables should be reviewed and revised as needed.

Table 6-2 Equipment Risks Associated with the Salt Storage Tanks - Inorganic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt leakage	0.001	250	82	1
Foundation settlement	0.0001	250	8	2
Loss of foundation cooling	0.005	1	2	3
Defects in the tank insulation ¹	0.1	0	0	4
Defects in the tank insulation ²	0.0005	250	41	5
Loss of inventory level signal	0.1	0	0	6
Loss of electric salt heaters	0.01	0	0	7

Notes:

1. Local gap, leading to a local increase in the heat loss.
2. Local gap, exposing the outside of the tank to water. In combination with salt spills on the outside of the tank, the carbon steel shell develops stress corrosion cracking.

Mitigation or Response:

1. Transfer the salt inventory to the companion storage tank while repairs are made to the tank which is leaking.
2. Remove the tank floor, lift the tank, remove and replace the foundation, lower the tank, and replace the floor.
3. Move the electric power supply to the redundant fan. The thermal inertia of the foundation will prevent excessive soil temperature during the period required to start the redundant fan.
4. Repair the defects when discovered. This is to avoid thermal gradients in the shell, which have the potential to establish stresses equal to the yield value.
5. The risk for carbon steel tanks is discussed above in Section 5.2.1.
6. Use the redundant level instrument(s) until repairs are made to the failed instrument.
7. Replace the failed element(s) in the salt heaters. The temperature of the tank will not meaningfully decrease in the time required to repair the heater.

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Table 6-3 Equipment Risks Associated with the Salt Steam Generator - Inorganic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Leakage due to failures of the tube-to-tubesheet connections	0.05	15	247	1
Pitting corrosion in evaporators using stainless steel tubes ¹	0.01	15	50	2
Leakage results in excessive water vapor accumulation on the salt side	0.001	15	5	1
Leakage leading to salt exposure to water vapor	0.01	0	0	3
Leakage leading to water vapor transfer to the ullage in the salt tanks	0.01	0	0	4

Note:

1. Inadequate control over water chemistry leads to flow accelerated corrosion in the carbon steel components of the condensate system. Accumulation of deposits in evaporator tubes leads to cathodic damage to passivation layer on tubes.

Mitigation or Response:

1. Drain the heat exchangers on the salt- and the water/steam-sides, repair the tube leaks by plugging, fill on the water side, preheat by means of the electric water heaters, and fill on the salt side.
2. Maintain dissolved oxygen and pH within values required by the water treatment system.
3. Salt/water reactions are largely benign and commercially acceptable.
4. Water vapor is vented from the ullage spaces during the daily storage tank charge/discharge cycles.

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Table 6-4 Equipment Risks Associated with the Salt Pumps - Inorganic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Loss of collector field cold salt pump	0.0005	60	10	1
Simultaneous loss of power to the collector field salt pump and the field heat tracing	0.0001	60	2	2
Loss of steam generator hot salt pump	0.0005	2	1	3
Loss of steam generator attemperation pump	0.0005	2	1	4

Mitigation or Response:

1. Replace the pump with the warehouse spare. The salt in the receiver tubes and in the flex hoses will freeze. Portable impedance heating systems are require to thaw the collector loops in sequence.
2. Repair power supply to cold salt pumps and field heat tracing. Thawing period for large salt piping can be on the order of 30 days, but the overall schedule is determined by impedance heating of the receiver tubes and the flex hose assemblies.
3. Loss of the hot salt pump trips the steam generator. Replace the pump with the warehouse spare. Restarting the steam generator can occur during the pump replacement period.
4. Replace the pump with the warehouse spare. During the repair period, the steam generator can be maintained at the cold salt temperature by means of the electric water heaters and the heat trace cables on the heat exchangers.

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Table 6-5 Equipment Risks Associated with the Salt Valves - Inorganic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Sticking cold salt pump discharge isolation valves ¹	0.001	60	20	1
Sticking cold salt pump discharge isolation valves ²	0.0001	60	2	1
Sticking cold salt pump minimum flow recirculation valves ¹	0.001	60	20	1
Sticking cold salt pump minimum flow recirculation valves ²	0.0001	60	2	1
Sticking hot salt or attenuation pump discharge isolation valves ¹	0.01	2	7	2
Sticking hot salt or attenuation salt pump discharge isolation valves ²	0.001	2	1	2
Sticking hot salt or attenuation pump minimum flow recirculation valves ¹	0.01	2	7	2
Sticking hot salt or attenuation pump minimum flow recirculation valves ²	0.001	2	1	2
Sticking heat exchanger vent and drain valves ^{1,3}	0.005	5	8	3
Sticking heat exchanger vent and drain valves ^{2,3}	0.0005	5	1	3
Valve stem leakage past conventional stem packings ¹	0.01	5	17	4
Internal isolation valve leakage in the steam generation system ⁴	0.001	5	2	2

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Notes:

1. Failure rate with conventional valve stem packings.
2. Failure rate with bellows valve stem seals.
3. In conjunction with water leakage rates which are high enough to adversely affect the flow distributions on the salt side; i.e., vapor binding.
4. Internal leakage prevents accurate control over heat exchanger temperature during transient conditions.

Mitigation or Response:

1. Problem is most likely to occur when switching from Overnight Hold, with 1 cold salt pump in service, to Normal Operation, with 2 or 3 cold salt pumps in service. Worst case condition shown in the table, when valve problems prevent all 3 pumps from operating. The salt in the receiver tubes and in the flex hoses will freeze. Portable impedance heating systems are require to thaw the collector loops in sequence.
2. Drain the heat exchangers on the salt side, repair or replace the valves, and refill the heat exchangers. The steam generator can be maintained at the cold salt temperature by means of the electric water heaters and the heat trace cables on the heat exchangers.
3. If the heat exchangers cannot be drained, then repairs can begin only after the heat exchanger temperatures have decayed below 200 °C. Repair or replace the valves, preheat on the water side, and then fill on the salt side.
4. Drain the piping, replace the stem packing, and replace any insulation or heat trace cables that have been damaged by exposure to the salt.

Table 6-6 Equipment Risks Associated with the Salt Instruments - Inorganic Fluids
(Identical to Table 5-6 Equipment Risks Associated with the Salt Instruments - Organic Fluids)

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Inaccurate, or sporadic loss of, flow meter readings	0.05	0	0	1
Salt freezing in the pressure instruments	0.05	0.5	8	2
Inaccurate, or loss of, temperature readings	0.01	0.5	2	3

Mitigation or Response:

1. Switch to manual operation until output signals from flow meters are corrected.
2. Repair the insulation and/or the heat tracing on the instrument stubs.
3. Replace the thermocouple, or switch to the redundant dual-element connections.

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Table 6-7 Equipment Risks Associated with the Heat Tracing - Inorganic Fluids

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt freezing in the pressure instruments	0.05	0.1	2	1
Salt freezing in the heat exchanger vent and drain lines	0.01	2	7	2
Damaged insulation and marginal heat trace capacity result in lines, when frozen, cannot thaw ¹	0.01	3	10	3

Note:

1. Problems primarily associated with the small diameter lines, such as the heat exchanger vent lines, the heat exchanger drain lines, and the instrument stubs.

Mitigation or Response:

1. Repair the insulation and/or the heat tracing on the instrument stubs.
2. Repair the insulation and/or the heat tracing on the vent and drain lines.
3. Repair the insulation and/or the heat tracing on the salt lines, followed by a waiting period for thawing.

A review of the annual product values shows that the major risks have, to a first order, a value greater than 10 and the minor risks have a value less than 10. The largest risks to the equipment items are summarized below in Table 6-8. Detailed discussions of each of the first 7 of the 9 principal risks (except the second) are presented in the sections which follow. The second, eighth, and ninth principal risks are identical to the second, fourth, and fifth principal risks in the summary Table 5-8 for organic fluids.

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Table 6-8 Summary of the Highest Equipment Risks - Inorganic Fluids

Equipment Item	Risk	Annual Product, Probability * Duration
Salt steam generator	Leakage due to failures of the tube-to-tubesheet connections	247
Thermal storage tanks	Leakage due to weld failures, low cycle fatigue, or transient stresses	82
Salt steam generator	Pitting corrosion in evaporators using stainless steel tubes	50
Thermal storage tanks ¹	Leakage due to stress corrosion cracking in the cold salt tank	41
Salt valves	Sticking cold salt discharge isolation valves	20
Salt valves	Sticking cold salt pump minimum flow recirculation valves	20
Salt valves ¹	Valve steam leakage past conventional stem packings	17
Salt pumps	Loss of collector field cold salt pump	10
Heat tracing ¹	Damaged insulation and marginal heat trace capacity result in lines, when frozen, cannot thaw	10

Note:

1. Same as the risks in the risk summary Table 5-8 for organic fluids

6.2.1 Leaks in the Hot Salt Tank

There are no commercial parabolic trough projects operating with inorganic heat transfer fluids. However, the design collector field outlet temperature for projects using inorganic heat transfer fluids (500 to 540 °C) is the same order of magnitude as the design hot salt temperature for commercial central receiver plants (565 °C). As such, the experience with hot salt tanks in central receiver projects can perhaps be used as a proxy for the projected performance of hot salt tanks in parabolic trough projects using inorganic heat transfer fluids.

Excluding the projects in China, there are 5 hot salt tanks in commercial service: Gemasolar; Crescent Dunes; Cerro Dominador (2 tanks); and Noor III. The largest risk to the availability of a project is the low cycle fatigue life of the hot salt tank. All commercial projects, with the exception of the Cerro Dominador project in Chile, use a single hot tank. Further, all of the salt delivered by the solar field passes through the hot salt tank. Should the hot tank fail, there is no longer a source of thermal energy

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for the steam generator. As a result, the plant cannot generate electric power, and the project revenues drop to zero.

Consequences of a Leak

The consequence of a leak in the hot salt tank in a parabolic trough project using an inorganic heat transfer fluid closely follows the discussion above for the consequences of a leak in the cold tank or the hot tank of a trough project using an organic heat transfer fluid. To briefly reiterate, the repair period associated with a leak is a lengthy one, due to the following effects:

- The tank and the foundation must be cooled to essentially ambient temperature prior to personnel entering the tank and analyzing the failure. The cooling period is on the order of 7 to 10 days due to the large thermal mass of the foundation, and due to a requirement to control intra-tank temperature differentials to values low enough ($< 30\text{ }^{\circ}\text{C}$) to prevent high internal stresses.
- The source of the tank failure must be identified. This may include removal of tank sections, and laboratory examinations for items such as corrosion, low cycle fatigue fractures, or weld defects. The time required for laboratory examinations is on the order of one to perhaps a few weeks.
- Repairs may involve a wide range of possibilities and conditions. Independent, potentially unrelated activities and associated schedules include:
 - Procurement and delivery of replacement tank sections: At least several weeks
 - Removal of defective welds and rewelding: One to several weeks
 - Lifting the tank and replacing the foundation insulation: Two to several months
 - Post weld heat treatment of the welds in the thick metal sections: At least one week
- Once repairs are complete, the tank must be preheated prior to refilling with salt. The preheat period is on the order of 7 to 10 days.

The overall outage will last at least 3 months, and could extend to periods as long as 8 months.

For a representative commercial project, the monthly revenues are on the order of \$4,000,000. If the plant is not in operation for, say, 8 months, then cost to repair the tank plus the loss in revenue could be \$35,000,000. Further, if the plant is not supplying the full complement of electric energy to the local utility, then the project may be considered partially in default regarding the power purchase agreement.

Failure Mechanisms

An hypothesis of the failure mechanism for hot salt tanks is discussed in detail in Section 5.2, *Failure Patterns*, of Volume 3 - Narrative.

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Recommendations

A range of potential solutions to, and alternate tank designs for, the current commercial tank designs is discussed in Section 5.12, *Solutions*, of Volume 3 - Narrative.

6.2.2 Leaks in the Steam Generator

As noted above in Section 6.2.1, there are no commercial parabolic trough projects operating with inorganic heat transfer fluids. However, the design collector field outlet temperature (500 to 540 °C) for projects using inorganic heat transfer fluids is the same order of magnitude as the design hot salt temperature (565 °C) for commercial central receiver plants. As such, the experience with steam generators in central receiver projects can perhaps be used as a proxy for the projected performance of steam generators in parabolic trough projects using inorganic heat transfer fluids.

Leaks have been noted in the salt steam generators in commercial central receiver projects. The source of the leaks can be a variety of short- and long-term damage mechanisms, including fatigue, oxidation, creep, alloy thermodynamic stability, deposition, and corrosion.

There are 7 salt steam generator in commercial service: Gemasolar; Crescent Dunes (2 trains); Cerro Dominador (2 trains); and Noor III (2 trains). However, none of the steam generators which have been in service for an extended period (> 10 years) of time. As such, the source of the leaks are likely associated with short-term damage mechanisms. These include 1) relaxation of the friction connections between the tubes and the tubesheet, allowing the high pressure fluid (water/steam) to leak to the low pressure side (salt), and 2) pitting corrosion of the stainless steel tubes in the evaporator, which results from solids accumulation on the water side.

The relaxation of tube-to-tubesheet connections, discussed in the section *Thermal Cycles* below, generally results from a failure to develop a process control that can safely, and continuously, operate the steam generators at the 1-percent load case. The 1-percent load case is an arbitrary requirement; i.e., it is no more applicable than the 2- or the 5-percent case. But if the process design, at 1 percent load, can protect the equipment against stress distributions that could reduce the low cycle fatigue life, then the process design can comprehensively protect the equipment over the full range of part- and full-load conditions.

The pitting corrosion of the evaporator tubes, discussed in the section *Feedwater Chemistry* below, can result from the following:

- Improper calibration of the instruments in the water treatment system
- Inadequate training of the operators for the water treatment system.

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- The time required to establish the conditions (temperature, pH) necessary to control the levels of dissolved oxygen.

Thermal Cycles

Almost by definition, the steam generator operates through a daily startup cycle. During startups and shutdowns, some portion of the steam flow is directed around the turbine and dumped to the condenser. During startup, the quantity of bypass steam is a function of the turbine metal temperature and the feedwater chemistry. The bypass flows represent an energy loss to the system. To keep the energy losses as low as practical, the steam generator vendor designs the equipment to start, and to shut down, as quickly as practical. However, each heat exchanger requires a large number of tube-to-tubesheet connections, and there are several locations where a step change in the metal thickness occurs. Rapid temperature changes, in combination with friction connections and step changes in thickness, are generally incompatible.

To ensure that the heat exchangers have fatigue lives consistent with the life of the project, the vendor will specify limits on rates of temperature change and thermal. Representative values the following:

- An allowable rate of temperature change of 10 °C/min
- An allowable thermal shock (difference between the temperature of the fluid entering the heat exchanger and the metal temperature at the entrance to the heat exchanger) of 60 °C. Typically, one thermal shock per day is allowed.
- A number of startup cycles consistent with the life of the project, in combination with a safety factor. For example, a project life of 30 years implies a nominal cycle life of 30 years * 1 startup cycle per day = 10,000 cycles. To this is added a safety factor to account for difficulties in exactly predicting the low cycle fatigue damage associated with each startup cycle, and to provide a margin against a process design which does not exactly satisfy the vendor limits on rate of temperature change and thermal shock. A representative safety factor is in the range of 3 to 4.

Problems arise in two areas: 1) the blending of hot salt with cold salt during startup and shutdown; and 2) establishing minimum flow rates.

Hot Salt / Cold Salt Blending During overnight hold, cold salt is circulated through the steam generator. To start the steam generator, the hot salt pump is started, and hot salt is blended with cold salt from the attemperation pump at a point upstream of the superheater / reheater. The relative proportions of hot salt and cold salt are adjusted until the flow of cold salt reaches zero. To shut down the steam generator, the process is essentially reversed.

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The blending process of adding hot salt to cold salt is a consequence of the vendor limits on allowable rates of temperature change and thermal shock.

The vendor also specifies a minimum allowable flow rate on both the shell- and tube-sides. A typical value is 16 percent, which corresponds to a turndown ratio of 6:1. At low flow rates, the pressure drops on both the tube- and the shell-sides approach very low values. Local effects due to friction and buoyancy can then influence the flow distribution, which can produce stratified flow, channel flow, or both. If the flow distribution is not uniform, then the temperature distribution within the heat exchanger is also not uniform. Non-uniform temperature distributions can produce internal stresses which are non-uniform, unpredictable, and oscillating. These stresses may add to, or subtract from, the normal process stresses, which can lead to meaningful reductions in the low cycle fatigue life of the equipment.

The hot salt pumps and the attemperation pumps share a common pressure point at the mixing station upstream of the superheater / reheater. As such, the following parameters all influence one another:

- The discharge pressure of the hot salt pump
- The number of hot salt pumps in operation: one pump; or two
- The position of the minimum flow recirculation valve for the hot salt pump(s)
- The discharge pressure of the attemperation pump
- The position of the minimum flow recirculation valve for the attemperation pump

During the startup of the steam generator, the following characteristics apply:

- The pressure drop in the system is low
- The discharge pressures of the hot salt pump and the attemperation pump are both low
- The flow rate from the hot salt pump to the mixing station is low, which requires the minimum flow recirculation valve to control the (Mixing station flow) + (Recirculation flow)
- Depending on the speed range of the attemperation pump, the minimum flow recirculation valve for the attemperation pump can also be open to control the (Mixing station flow) + (Recirculation flow)

Under these conditions, the system flow stability can be marginal to poor. For example, to increase the flow of hot salt to the mixing station, the minimum flow recirculation valve for the hot pump can be

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closed slightly. Since the pressure drop in the steam generator flow path is different than the pressure drop in the hot salt pump recirculation line, a change in the recirculation valve position results in a change in the discharge pressure of the hot salt pump. This, in turn changes the effective discharge pressure of the attemperation pump, which changes the flow rate from the attemperation pump. In response, the minimum flow recirculation valve for the attemperation pump can change position to provide the required (Flow to the mixing station) + (Recirculation flow). The changes in the discharge conditions for the attemperation pump are then carried over to the hot salt pump, which normally results in a change in the recirculation valve position for the hot salt pump to provide the desired net hot salt flow to the mixing station.

The system can then oscillate between two quasi-stable operating points. The thermal inertia of the metal in the heat exchanger is a very small value compared with the thermal power available in the flow of either the salt or the water/steam. To a first order, the temperature of the metal responds almost instantly to a change in the temperature of the fluid. As such, oscillations in the relative proportions of hot salt and cold salt result in oscillations in the metal temperatures at the hot ends of the superheater / reheater.

The problem is compounded by the valve selection on the salt side. The salt control valves downstream of the superheater and the reheater must have Cv values which are high enough to accommodate the design point flow rates with a reasonable pressure drop (< 1 bar). During startup and shutdown, the rate at which hot salt is added to cold salt must satisfy the vendor limit regarding the allowable rate of temperature change, which is nominally $10\text{ }^{\circ}\text{C}/\text{min}$. Using representative flow - head curves for the attemperation pump, the acceleration rate at which hot salt can be added to the cold salt is $0.0023\text{ m}^3/\text{hr-hr}$ (6 gal/min-min). A flow rate of $0.0023\text{ m}^3/\text{hr}$ is approximately 0.15 percent of the design flow rate from the hot salt pump. Further, early in the startup process, the salt flow rate to the steam generator is perhaps 15 percent of the design flow rate. As such, the pressure drop through the steam generator is on the order of $0.15^2 = 2$ percent of the design pressure drop. To control the flow from the hot salt pump with this level of accuracy, in combination with a low discharge pressure, a backpressure must be imposed on the steam generator train. The backpressure is typically provided by the salt control valves downstream of the superheater and the reheater. However, these valves have reasonably large discharge coefficients, and to provide a meaningful backpressure on the system, the valves are forced to operate in the range of 2 to 4 percent open. Due to the low system pressures, the hot salt pump is already operating at the minimum speed. As such, all of the flow control adjustments during startup must be made by the control valves. As might be expected, the ability to adjust the flow in increments of 0.15 percent per minute is generally poor. The result is oscillations in the flow from the hot salt pump, which results in oscillations in the mixed salt temperature upstream of the superheater and the reheater. The oscillations can result in rates of temperature change which are a factor of 2 to 4 larger than the allowable rates.

Minimum flow A typical vendor limit on the minimum allowable flow rate is 16 percent of the design flow rate.

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If the design flow rate of the cold salt attemperation pump is 15 percent of the design steam generator flow rate, then operating the attemperation pump at 100 percent speed during overnight hold, and early in the startup process, satisfies the minimum flow requirement on the salt side of each of the heat exchangers.

Some projects use a recirculation pump between the steam drum and the cold end of the evaporator, and a separate recirculation pump between the steam drum and the cold end of the preheater. The flow capacities of each pump are greater than 15 percent of the design water flow rates, and operating the pumps early in the startup process ensures that the minimum water flow rates are always maintained in the evaporator and the preheater.

In contrast, steam flow is initiated in the superheater and in the reheater only when the blending of hot salt with cold salt begins. The steam flow rates begin at 0 percent, and increase with an increase in the flow rate of hot salt. As such, during each startup, the steam flow rates in both heat exchangers are necessarily below the minimum allowable flow rates. Due to the very low pressure drops at these flow rates, the flow distribution among the approximately 900 tubes is expected to be poor. Non-uniform flow distributions lead to non-uniform temperature distribution, which, in turn, lead to unpredictable, and potentially damaging, stress distributions.

Potential solutions to the above conditions are discussed in Volume 3 - Narrative. It can be noted that informal discussions with heat exchanger vendors indicate that it is not possible to design a heat exchanger that is sufficiently robust to withstand arbitrarily high rates of temperature change in combination with an arbitrarily large number of thermal cycles.

Feedwater Chemistry

The feedwater chemistry must be controlled to limit solids deposits on the turbine blades, and to limit corrosion rates of the carbon steel equipment in the condensate system. The latter is necessary to control the accumulation of deposits on the heat transfer surfaces of the steam generator. Solids deposits can decrease the heat transfer rates of the heat exchanger, and lead to pitting corrosion in the evaporator if stainless steel tubes are used.

Pitting corrosion is decidedly problematic regarding the availability of the steam generator. Depending on the pH and the dissolved oxygen levels in the condensate, either hematite (Fe_2O_3) or magnetite (Fe_3O_4) will form on the carbon steel surfaces.

Hematite has a much lower solubility in water than magnetite, thus providing significantly better resistance to flow accelerated corrosion. Flow accelerated corrosion is caused by local removal of protective oxide formed on a steel surface by turbulent water flow in single-phase conditions, or by saturated steam flow in two-phase conditions. Loss of the protective oxide leads to rapid decrease in the thickness of the underlying component. To limit single-phase corrosion, an oxidizing feedwater

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chemistry is essential to ensure the formation of hematite, and thereby reduce the dissolution of magnetite. To limit two-phase corrosion, a pH increase across the entire Rankine cycle is required to push alkalinity into the liquid phase.

Oxides produced due to corrosion will travel to the evaporator in the steam generator. At the location in the evaporator tube where the phase change begins, the solids will tend to accumulate. Underneath the solids, the pH and the oxygen levels on the surface of the tube can be different than the pH and the oxygen levels on the surface of the tube away from the deposits. This can produce a galvanic cell, which, in turn, attacks the protective oxide layer on the surface of the tube, leading to pitting corrosion beneath the deposit.

Leaks in the evaporator tubes lead to a series of problems:

- When the leakage rate exceeds an acceptable value, the heat exchanger must be repaired. This involves draining the steam generator train, cooling the heat exchangers to ambient, identifying and plugging the tubes which are leaking, preheating the heat exchangers, and refilling with system with salt. The time required to do so is at least 1 week. The expense to the project, for both the repairs and the loss in revenue, is on the order of \$1,000,000.
- The feedwater passes to the salt side of the heat exchanger, and is eventually lost to the environment through the vents in the storage tanks. This represents an increased demand on the water treatment system.
- During overnight hold, the water level in the drum must be replenished. If the leakage rate exceeds some threshold value, the temperature of the salt at the cold end of the preheater falls below a minimum value. To compensate, additional energy is supplied to the steam generator by 1) operating the electric water heaters normally used for startup, or 2) starting the hot salt pump, and adding a small quantity of hot salt to the flow of cold salt from the attemperation pump. Daily operation of the electric heaters can increase the annual auxiliary electric energy demand by 8 GWhe. Supplying hot salt to the steam generators during overnight hold results in less thermal energy available for electric power production.
- Prior to filling with salt, the steam generator is preheated using the electric water heaters. The combined capacity of the heaters is large (3 MWe), but it is not infinite. If the thermal demand for preheating, in combination with the thermal demand for preheating the makeup water to the drum, exceeds the capacity of the electric heaters, then it is no longer possible to raise the temperature of the heat exchangers to a safe fill temperature of, say, 275 °C. If so, the project may decide to thermally shock the heat exchangers to return the plant to commercial operation.

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- Leakage in the evaporator exposes the salt to steam. The majority of this steam is released from the salt in the cold storage tank. However, small amounts of water can bond with the salt in the form of an emulsion or a hydrate; the actual mechanism is not clear. From the cold tank, the salt/water combination passes through the receiver, and then to the hot tank. Due to the high temperatures in the hot tank, in combination with residences times measured in hours, the water is released from the salt. Many projects use a vent line from the top of the receiver to the top of the hot salt tank. When the receiver is drained each day, the metal temperatures in the panels cool to ambient temperature. If the metal temperatures drop below the dew point, then water vapor can travel from the ullage space in the hot tank, up the vent line, into the receiver panels, and condense on the inside of the tubes. Although the tubes are fabricated from a nickel alloy, the alloy is not immune to intergranular stress corrosion cracking. Tube cracks have been seen in commercial projects, with the source of the cracking traced to a sensitized material, chlorides in the salt, residual tensile stresses, and liquid water.

Dissolved Oxygen Control in the Rankine Cycle

The water treatment system in the Rankine cycle is not an explicit element in the design of the steam generator. However, the corrosion lifetime of the steam generator can be influenced by the feedwater chemistry, including the levels of dissolved oxygen.

Oxygen scavengers are effective only above a certain threshold temperature; typically, 120 °C. To help control dissolved oxygen levels over the full range of startup temperatures, the project can adopt one of the following approaches:

- Provide a vacuum deaerator or a steam sparger in the condenser hot well
- Maintain the condenser vacuum overnight.

The selection, to be made by the project, involves a comparison of capital investment and operating expense.

The keys to preventing the leakage problems noted above is to routinely calibrate the instruments in the water treatment system, and to ensure that the operators are following all of the procedures specified by the supplier of the water treatment system. This is, of course, obvious. But problems persist in commercial projects, and inadequate control over the water chemistry represents a risk to the project.

Recommendations

A detailed discussion of potential changes to the process design to protect the low cycle fatigue life of the steam generator is discussed in Section 6.6, *Changes to the Equipment and the System Design*, of Volume 3 - Narrative.

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Recommendations regarding the selection of tube material for the evaporator, and approaches for the control of water chemistry, are discussed in Section 6.6.4, Material Selection in the Evaporator, of Volume 3 - Narrative.

6.2.3 Salt Pumps

As discussed in Section 12.3.4, the principal means of freeze protection in the field piping and the collector receivers is the continuous circulation of salt. However, at some point in the life of the project, circulation through the collector field could be lost for a period of time which is long enough for the salt to freeze in the receivers, and for the salt to freeze in the field piping. There are no commercial parabolic trough projects which use a nitrate salt as the heat transfer fluid. As such, there are no data on the reliability and the availability of continuous salt circulation.

The basic circulation concept relies on a cold salt pump(s), which draws suction from the cold salt tank and returns the salt to the cold tank. In general, the reliability of salt pumps at commercial projects is excellent. For example, problems with the heat tracing on the receiver pumps at the 10 MWe Solar Two demonstration project required the pumps to operate on a continuous basis, and the pumps did so for an estimated period of some 18,000 hours. Nonetheless, the circulation concept relies on a continuous supply of electric power to the salt pumps, and relies on the continuous operation of the salt pumps. As discussed in Section 2.2.5 of Volume 3 - Narrative, there is a range of conditions which could result in a loss of circulation.

Impedance Heating of Receiver Tubes

As discussed in Section 2.5.1 of Volume 3, Narrative, the time required to recover from frozen salt in a receiver tube is on the order of 12 hours. A 12-hour thaw period is for a receiver in which the vacuum is intact, and the heat losses from the tube are at a minimum. For a tube which has lost vacuum, or has a broken glass envelope, the heat losses from the tube increase to the point where the impedance heating system will not be able to thaw the frozen salt. In this case, a temporary insulation cover is installed over the receiver to increase the thermal resistance to the point where thawing can be completed.

Heat Trace Heating of Field Piping

As discussed in Section 2.5.3 of Volume 3, Narrative, the time required to recover from frozen salt in the field piping ranges from 20 days to 1 month, depending on the condition of the pipe insulation.

Schedule for Thawing the Collector Field

As discussed in Section 10.9.2, it is prohibitively expensive to provide a permanent impedance heating system for the collector field. As such, the project uses a number of portable impedance heating

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systems, that can be moved from one loop to the next as thawing progresses. Representative times to thaw all of the receivers in a commercial plant range from 15 to 30 days.

To ensure that salt circulation in a loop can be established immediately after the salt is melted in the receivers, the field piping must be thawed first. As such, should all of the salt freeze in the collector field, the time to recover should fall in the range of the following possible durations:

- 20 days to thaw the field piping, followed by 15 days to thaw the collector loops, for a total of 35 days
- 30 days to thaw the field piping, followed by 30 days to thaw the collector loops, for a total of 60 days.

Nonetheless, a range of 35 to 60 days should be viewed as something of a lower bound. Should there be problems with the supply of electric power to the field piping heat trace system, the heating process will be interrupted. This will cause the piping to cool, and extend the overall heating period. Similarly, if the availability of the portable impedance heating systems is less than 100 percent, then the overall melting period for the collector loops will be extended.

It is not unreasonable to expect that the first commercial projects will freeze salt in the collector field more often than expected, and the recovery times will be longer than expected. A potential, but not implausible, scenario might be on in which salt freezes in the field once in each of the first 5 years of commercial operation, and the recovery time in each case is 75 days. This would result in an availability in the first 5 years of commercial service of 75 percent, rather than the expected 90 percent. A worst case scenario might be on in which salt freezes twice in the first year of commercial service, and the recovery time in each case is 90 days. This would result in an availability of 50 percent in the first year of service.

Plant availabilities in this range, particularly early in the life of the project, will have a noticeable, and detrimental, effect on the return on investment to the equity providers. In principle, the equity providers can compensate for the availability risks by increasing the return on investment. However, this will increase the levelized cost of energy. At some point, the risk premium for the use of an inorganic heat transfer fluid will offset the benefits of using an inorganic fluid; i.e., a reduction in the unit cost of the storage system, and an increase in the gross Rankine cycle efficiency. The economics of a plant using an inorganic fluid may essentially be on a par with the economics of a plant using Therminol as the working fluid. In this case, the investment community would, in their own interests, continue with the current state of the art in trough technology, and the commercial potential of the advanced working fluid may not be realized.

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6.2.4 Salt Valves

The risks associated with salt valves in projects using inorganic heat transfer fluids are the same as the risks associated with salt valves in projects using organic fluids, as discussed in Section 5.2.3.

6.2.5 Heat Tracing

The risks associated with heat tracing in projects using inorganic heat transfer fluids are the same as the risks associated with heat tracing in projects using organic fluids, as discussed in Section 5.2.4.

6.3 Project Risks

The discussion above, in Section 6.2, evaluated risks associated with the salt equipment. However, there are also risks at the project level which can influence the availability of the plant. Two such risks, which are identical to the project risks listed in Table 5-9, are presented in Table 6-9.

Table 6-9 System Risks for the Salt Equipment - Inorganic Fluids
(Identical to Table 5-9 for Organic Heat Transfer Fluids)

	Annual probability, fraction	Annual outage duration, days	Annual product	Mitigation or Response
Process design is not consistent with the performance requirements in the project financial model	0.5	20	10	1
Operating personnel do not have an understanding of the plant that is sufficient to prevent damage to the equipment	0.5	20	10	2

Mitigation or Response:

1. Research the process design in other commercial projects. Copy those features which are successful, and avoid those features which are not effective.
2. Provide sufficient funds to attract, and retain, a mechanical engineer at the site for the first 5 years of commercial service. Provide ad hoc training to the operators in topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

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6.3.1 Process Design

The process design requirements for a parabolic trough project using inorganic heat transfer fluids are essentially the same as the process design requirements for a trough project using organic heat transfer fluids, as discussed in Section 5.3.1.

6.3.2 Selection of Operating Personnel

Parabolic trough projects using inorganic heat transfer fluids are more complex than projects using organic heat transfer fluids. This, in turn, requires an operating staff that has a comprehensive understanding of the equipment and the temperature limitations. Attracting skilled personnel to the projects, many of which are located at remote sites, often runs counter to a goal of holding operation and maintenance costs to values stated in the financial model.

Parabolic trough projects necessarily operate through daily thermal cycles. Plants using inorganic heat transfer fluids have a number of features which make the design more fragile in cycling conditions than plants using organic heat transfer fluids, as follows:

- The equipment on the hot side is stainless steel. Unlike carbon and ferritic steels, stainless steel is susceptible to a range of corrosion mechanisms, including intergranular stress corrosion cracking, stress relaxation cracking, and pitting corrosion.
- The difference in temperature between the cold side (295 °C) and the hot side (~520 °C) is a meaningful 225 °C. This difference is roughly 2½ times the difference in a parabolic trough project using organic heat transfer fluids. During steam generator startup and shutdown, inaccurate blending of cold salt with hot salt has the potential to thermally shock the equipment well beyond the capabilities of the equipment to withstand the thermal shock.
- The heat exchangers in the steam generator are large equipment items, with thick wall sections to withstand pressure as high as 145 bar. Further, the tube-to-tubesheet connections, which can number up to 3,000 in each of 2 steam generator trains, rely on a combination of friction (tube expansion) and welds (strength welds between the ends of the tubes and the face of the tubesheet) to seal the high pressure fluid from the low pressure fluid. During startup and shutdown, inaccurate blending of cold salt with hot salt can result in rates of temperature change well beyond the vendor limit, leading to high values of local thermal stresses, flexing of the tubesheet, relaxation of the tube-to-tubesheet connections, and internal leakage.
- During overnight shutdown of the Rankine cycle, the condenser is often opened to the atmosphere. The goal is to reduce the auxiliary energy consumption required to maintain condenser vacuum for several hours each day. Exposing the condenser hotwell to the atmosphere increases the dissolved oxygen in the condensate. Prior to daily startup, the

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dissolved oxygen and the pH values must be set to values required by the water treatment system. Any errors in setting the dissolved oxygen and pH levels can lead to long-term problems with flow accelerated corrosion, fouling of the heat transfer surfaces in both the feedwater heaters and the steam generator, and potentially pitting corrosion in the evaporator.

Recommendations

The recommendations regarding plant personnel for trough projects using inorganic heat transfer fluid are the same as the recommendations for personnel as trough projects using organic heat transfer fluids, with one exception. The former type of projects are more complex, and more brittle, than the latter due to use of stainless steels, and due to larger temperature differences between the cold side and the hot side.

The project should provide sufficient funds to attract, and retain, a mechanical engineer at the site for the first 5 years of commercial service. The engineer would perform the following:

- During the period required to refine the logic in the distributed control system, provide guidance to the operators to protect equipment from low cycle fatigue damage during startup, shutdown, and transient conditions
- Provide simulator training to the operators on topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

If retaining a qualified engineer at a remote site proves to be problematic, an alternate approach might involve an engineer or a supervisor at a home office, monitoring the operator screens over an Internet connection. (On the Crescent Dunes project, a controls engineer and a mechanical engineer from SolarReserve remotely monitored the status of the receiver during the 3 year performance guarantee period.)

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7. Operating Requirements - Organic Heat Transport Fluids

7.1 Introduction

Operating requirements for parabolic trough projects, using organic heat transport fluids, include the following topics:

- Selection of design parameters, such as temperatures, flows, allowable fluxes, rates of temperature change, low cycle fatigue lives, and design Codes
- Evaluation of (Capital cost) versus (Capital cost + Operating cost). The evaluation considers items such as material selection, fabrication techniques, and installed redundancy
- Outline of hardware recommendations and specifications.

7.2 Oil-to-Salt Heat Exchangers

7.2.1 Requirements

In essentially all commercial parabolic trough projects, the working fluid in the collector field is a synthetic oil (Therminol). However, it is not economically feasible to use Therminol as the medium in the storage system for the following reasons:

- The cost of the fluid is high; on the order of \$20/gallon
- At the design temperature for the collector field (393 °C), the vapor pressure of Therminol is about 10 bar. Were the fluid to be stored directly, an ASME Section VIII pressure vessel, rather than an API 650-style tank, would be needed. The unit cost of the steel in the vessels (\$/kWht of energy stored) is prohibitively high for a commercial project.

To avoid these constraints, thermal energy from the Therminol is transferred to a relatively inexpensive storage fluid with a low vapor pressure (nitrate salt), which can be placed in an API 650-style tank. This, in turn, requires the addition of an oil-to-salt heat exchanger to the plant design. The requirements for the heat exchangers are fairly stringent. In commercial trough projects, the temperature rise across the collector field is only about 100 °C. Since the storage system requires the transfer of heat from the Therminol to the storage fluid, the approach temperatures in the heat exchangers must be smaller than typical to preserve as much of the limited 100 °C temperature rise as possible. Due to the requirement for small approach temperatures, the heat exchangers tend to be large and expensive.

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7.2.2 Process Arrangement

During a storage charge cycle, thermal energy is transferred from the collector field to the storage system. Due to the approach temperature in the oil-to-salt heat exchanger, this necessarily results in a hot salt temperature that is lower than the collector field outlet temperature.

During a storage discharge cycle, thermal energy is transferred from the storage system to the Rankine cycle. There are two potential process arrangements to accomplish this:

- Salt from the hot salt tank is supplied to the oil-to-salt heat exchanger. Oil from the hot end of the heat exchanger is supplied to a Therminol steam generator
- Salt from the hot salt tank is supplied to a salt steam generator.

Therminol Steam Generator

With a Therminol steam generator, the plant operates in one of three modes:

- Therminol from the collector field, at a temperature of 393 °C, is supplied directly to the steam generator. The Rankine cycle operates at the design values for the live steam pressure (100 bar), live steam temperature (379 °C), reheat steam temperature (379 °C), gross cycle efficiency (0.380), and gross generator output (100 percent)
- When the thermal output from the collector field exceeds the design duty of the steam generator, the additional energy is delivered to the storage system through the oil-to-salt heat exchanger
- When the thermal output from the collector field is less than the design duty of the steam generator, energy is withdrawn from the storage system, through the oil-to-salt heat exchanger, to supplement the energy supplied to the steam generator. Due to the approach temperatures in the oil-to-salt heat exchanger, the temperature of the Therminol temperature at the hot end of the heat exchanger is necessarily lower than the salt temperature in the hot tank. In the limit, the steam generator operates only with energy from the storage system; a representative temperature for the Therminol supplied to the steam generator is 386 °C. The Rankine cycle operates at reduced conditions for the live steam pressure (92 bar), live steam temperature (370 °C), reheat steam temperature (367 °C), gross cycle efficiency (0.367), and gross generator output (90 percent).

Salt Steam Generator

With a salt steam generator, the plant operates in one of two modes:

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- When the collector field is in operation, Therminol from the collector field, at a temperature of 393 °C, is supplied to the oil-to-salt heat exchangers. Salt at the hot end of the heat exchanger, at a temperature of 388 °C, is supplied to the hot salt tank. Salt from the hot tank, in turn, is supplied to the salt steam generator. The Rankine cycle is designed to operate under full load conditions with a steam generator source temperature of 388 °C
- When the collector field is not in operation, salt from the hot tank is supplied to the steam generator. Since the steam generator source temperature remains constant at 388 °C, the Rankine cycle operates at the design rating and efficiency.

With a steam generator source temperature of 388 °C, the efficiency of the Rankine cycle (0.375) is about mid-way between the efficiencies noted above for the Rankine cycle operating directly from the field (0.382) or from the storage system (0.367).

Commercial Selection

All commercial projects to date have selected a storage approach based on a Therminol steam generator rather than a salt steam generator. This has been done for three reasons:

- The storage capacities in commercial projects are generally in the range of 6 hours. On an annual basis, the Rankine cycle can operate for approximately 50 percent more hours directly from the field rather than from the storage system. This, in turn, favors an approach which can exploit the higher Rankine cycle efficiency available with a steam generator source temperature of 393 °C
- With a Therminol steam generator, only that portion of the thermal energy from the field which exceeds the design rating of the steam generator needs to be directed to the storage system. This results in a smaller oil-to-salt heat exchanger than the case with a salt steam generator, in which all of the thermal energy from the collector field needs to be directed to the storage system
- Therminol steam generators have demonstrated much higher annual availabilities than salt steam generators, and reaching the availability targets used in the financial model is perhaps the most important contributor to a successful project.

To avoid the need for separate sets of charging and discharging heat exchangers, the cost of the system is reduced by using one set of heat exchangers for both charging and discharging. This requires that the oil and the salt flow in one direction during charging, and then reverse and flow in the opposite directions during discharging.

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7.2.3 Startup and Shutdown Procedures

The oil-to-salt heat exchangers necessarily undergo a start and a stop cycle each day. Two methods have been adopted for controlling the temperatures in the heat exchanger during hold periods:

- At the end of a discharge cycle, the flows of Therminol and salt are reduced to the minimum, and the heat exchanger is tripped. This largely preserves the longitudinal temperature profile during the hold period
- At the end of a discharge cycle, the flows of Therminol and salt are reduced to the minimum, and the hot salt pump is tripped. This largely erases the longitudinal temperature profile, and brings the entire heat exchanger to a temperature close to the cold salt temperature.

The first method reduces the changes in the metal temperature associated with the next start cycle. However, a charge cycle cannot begin until the temperature of the Therminol from the collector field is within perhaps 10 to 15 °C of the metal temperature at the hot end of the heat exchanger.

The second method results in the largest changes in the metal temperature during the next start cycle. However, a charge cycle can begin as soon as the temperature of the Therminol from the collector field exceeds the salt temperature in the cold tank.

Most commercial project have adopted the second method for two reasons: 1) experience from commercial projects has shown that the longitudinal temperature profile in the heat exchanger does not persist as long as expected, and 2) natural circulation patterns can generate undesirable vertical temperature gradients in the heat exchanger. Nonetheless, stopping the flow of hot salt at the end of a discharge cycle can result in rapid cooling at the hot end of the heat exchanger. The heat exchanger must be designed to accommodate these rates of temperature change without affecting either the low cycle fatigue life or the integrity of the tube-to-tubesheet connections.

7.2.4 Possible Revisions to the Operating Procedures

In terms of reducing the potential for high transient thermal stresses in the heat exchangers, potential revisions to the process control might consist of the steps outlined below.

Morning Startup

- During overnight hold periods, cold salt circulates through the heat exchanger, and salt from the hot end of the heat exchanger is returned to the cold tank. The salt flow rate is equal to, or greater than, the minimum flow rate defined by the vendor

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- During morning startup, when the temperature of the Therminol from the collector field exceeds the temperature of the cold salt, a flow of Therminol is established at the hot end of the heat exchanger. The initial Therminol flow rate is equal to, or greater than, the minimum flow rate defined by the vendor
- As morning startup progresses, the temperature of the Therminol from the field increases. This, in turn, results in an increase in the temperature of the salt at the hot end of the heat exchanger. If the temperature of the salt is less than the temperature of the hot tank, the salt is recirculated back to the cold tank
- When the temperature of the salt at the hot end of the heat exchanger is within 5 °C of the salt temperature in the hot tank, the salt flow is directed to the hot tank, and the recirculation flow to the cold tank is stopped. The heat exchanger is now in normal operation, and the flow of cold salt is adjusted to provide the desired approach temperature at the hot end of the heat exchanger.

Transition from Charge to Discharge

At the end of a charge cycle, the heat exchanger is operating with the following characteristics:

- The design longitudinal temperature profile has been established along the heating length
- The salt flow rate and the Therminol flow rate are at values no lower than the minimum allowable values specified by the vendor.

To make the transition from charge to discharge, the flow directions of the salt and the Therminol must be reversed. This can be accomplished in the following steps:

- The cold salt pump is tripped. Simultaneously, the isolation valve in the Therminol flow to the hot end of the heat exchanger is closed. The cold salt flow and the hot Therminol flow must be stopped at the same time to 1) preserve the longitudinal temperature profile in the heat exchanger, and 2) prevent rapid change in the local metal temperatures at either the cold end or the hot end of the heat exchanger
- The hot salt pump is started. Simultaneously, the isolation valve in the Therminol flow to the cold end of the heat exchanger is opened. The hot salt flow and the cold Therminol flow must be started at the same time to 1) preserve the longitudinal temperature profile in the heat exchanger, and 2) prevent rapid change in the local metal temperatures at either the cold end or the hot end of the heat exchanger

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For the process to minimize the transient thermal stresses in the heat exchanger, the following must occur:

- When the cold salt pump is tripped, the coast down time of the pump should match the closing period for the Therminol isolation valve
- When the hot salt pump is started, the acceleration time for the pump should match the opening period for the Therminol isolation valve
- When the hot salt pump is started, the speed of the pump, and the percent open of the Therminol isolation valve, should be pre-selected values.

Nonetheless, startup and shutdown processes don't always repeat in a predictable manner. Problems can occur with pump vibrations during acceleration, sticking valves, and inaccurate instrument readings. If the pre-selected values for pump speed and valve position don't always preserve the longitudinal temperature profile during the transition from charge to discharge, a more conservative operating approach might be to remove the longitudinal temperature profile, and then reestablish the temperature profile. A conceptual approach, based on the use of variable speed drives for the pumps, is outlined below (If the pumps use constant speed drives, the approach can be modified to use valves for flow control.):

- Remove the longitudinal temperature profile by reducing the flow of Therminol to the hot end of the heat exchanger, and simultaneously increasing the flow of cold salt to the hot end of the heat exchanger. The two flow rates would be set to decrease the metal temperature at the hot end of the heat exchanger by the allowable rate set by the vendor (a nominal 10 °C/min). Relatively low temperature salt leaving the hot end of the heat exchanger would be recirculated back to the cold tank. When the flow of Therminol reaches the minimum value set by the vendor, and with the cold salt pump operating at the design speed, the approach temperature at the hot end of the heat exchanger would be a modest 5 °C or so. At this point, the flow of Therminol can be stopped, and the heat exchanger will assume a uniform temperature equal to the cold salt temperature. The speed of the cold salt pump is reduced, and pump is then tripped
- A flow of cold Therminol is established at the cold end of the heat exchanger. The temperature of the Therminol is nominally the same as the temperature of the cold salt; as such, no heat transfer occurs. The flow rate of Therminol is the same order of magnitude as the design flow rate. Therminol from the hot end of the heat exchanger is recirculated back to the suction side of the Therminol pumps

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- A flow of cold salt is established at the hot end of the heat exchanger by means of a bypass line. Salt leaving the cold end of the heat exchanger flows to the cold tank. The flow rate of cold salt is at least equal to the minimum allowable flow rate specified by the vendor
- The hot salt pump is started, with a speed which develops a head that is somewhat less than the cold salt pressure at the hot end of the heat exchanger. The pump is protected from overheating by the flow through the minimum flow recirculation valve
- The speed of the hot salt pump is increased to the point where the discharge pressure equals the cold salt pressure. The discharge check valve on the pump opens, and hot salt starts to mix with cold salt at the hot end of the heat exchanger
- The speed of the hot salt pump is increased at a rate which results in an increase in the mixed salt temperature at a rate equal to the allowable rate for the heat exchanger. The speed of the cold salt pump remains constant. As such, the flow rate on the salt side is always at least equal to the minimum allowable flow rate
- When the flow rate of hot salt is nominally equal to the flow rate of cold salt, the speed of the cold salt pump is reduced at a rate which increases the mixed salt temperature at the hot end of the heat exchanger at the allowable rate of temperature change specified by the vendor. When the flow rate of cold salt reaches a value of zero, the cold salt pump is tripped
- The flow rate of Therminol is adjusted to provide the desired approach temperature at the cold end of the heat exchanger. At this point, the normal longitudinal temperature profile in the heat exchanger has been reestablished.

During these steps, the vendor limits on the minimum flow rates and the allowable rates of temperature change are satisfied.

Compared with the typical process arrangement on commercial projects, the above steps require the addition of a salt bypass line from the cold salt pump to the hot end of the heat exchanger, and an isolation valve in the bypass line.

Evening Shutdown

Near the end of a discharge cycle, the output of the Rankine cycle is often reduced to a point near the minimum continuous load for the turbine. As the load is reduced, the final feedwater temperature is also reduced. This, in turn, results in a decrease in the temperature of the Therminol at the cold end of the steam generator, and, correspondingly, a decrease in the temperature of the Therminol at the cold end of the oil-to-salt heat exchanger.

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At some point in the process, the temperature of the Therminol at the cold end of the oil-to-salt heat exchanger decreases to a value which results in a salt temperature at the cold end of the heat exchanger that is below the minimum allowable value; perhaps, 285 °C. To compensate, the flow rate of hot salt can be increased to raise the approach temperature at the cold end of the heat exchanger. But a point is eventually reached, in which the temperature of the Therminol at the cold end of the heat exchanger is too low, and the turbine must be tripped. This will result in a trip of the oil-to-salt heat exchanger; i.e., the Therminol isolation valve at the heat exchanger will be closed, and the hot salt pump will be tripped. With the two flows stopped, the existing longitudinal temperature profile will be preserved until the start of the next charge cycle.

The longitudinal temperature profile will depend on a range of conditions at the time of the turbine shutdown, including the hot salt temperature, the load at which the turbine is tripped, the saturation pressure in the steam generator when the turbine is tripped, the number of extraction feedwaters in service, and so on. Since the range of conditions are like to change from day to day, the longitudinal temperature profile is also likely to change day to day. This could lead to a somewhat random set of conditions in which it is not possible to start the oil-to-salt heat exchanger and simultaneously satisfy the vendor requirements on the minimum flow rate and the allowable rate of temperature change.

To avoid this situation, the heat exchanger can be brought to a uniform cold salt temperature through the following steps:

- During part load conditions, the steam turbine normally operates under sliding pressure conditions to improve the expansion efficiency. However, shortly before the turbine is tripped, the operating mode can be switched from sliding pressure to constant pressure at the design value of 100 bar. This has the effect of increasing the temperature of the Therminol at the cold end of the oil-to-salt heat exchanger
- The turbine is tripped at load of perhaps 10 percent. Based on GateCycle models of the Rankine cycle part-load performance in a representative commercial project, the temperature of the Therminol at a load of 10 percent is on the order of 265 °C, as noted in Figure 7-1. A Therminol temperature of 265 °C is about 30 °C below the design value. However, given an approach temperature of perhaps 5 °C at the cold end of the oil-to-salt heat exchanger, the salt temperature at the cold end of the heat exchanger remains in an acceptable range
- After the turbine is tripped, the steam generator remains in operation, sending steam to the condenser through the bypass system. Auxiliary steam is supplied to the deaerator to provide a level of feedwater heating. Therminol, at a temperature of 260 to 265 °C, returns to the oil-to-salt heat exchanger

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- The cold salt pump is started, with a speed which develops a head that is somewhat less than the hot salt pressure at the hot end of the heat exchanger. The pump is protected from overheating by the flow through the minimum flow recirculation valve
- The speed of the cold salt pump is increased to the point where the discharge pressure equals the hot salt pressure. The discharge check valve on the pump opens, and cold salt starts to flow in the salt bypass line around the heat exchanger. The flow of cold salt start to mix with the flow of hot salt at the hot end of the heat exchanger

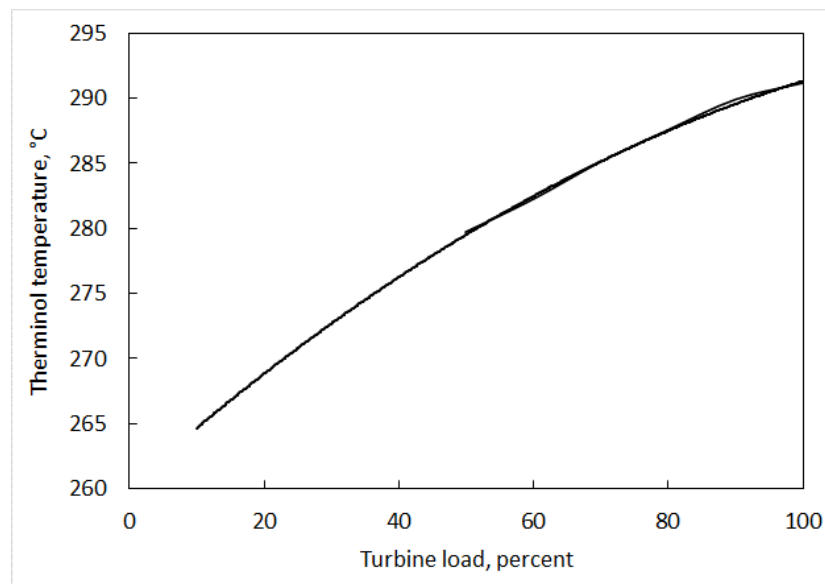


Figure 7-1 Therminol Temperature as a Function of the Turbine Load

- The speed of the cold salt pump is increased at a rate which results in an decrease in the mixed salt temperature at a rate equal to the allowable rate for the heat exchanger. The speed of the hot salt pump remains constant. As such, the flow rate on the salt side is always at least equal to the minimum allowable flow rate
- As the temperature of the mixed salt at the hot end of the heat exchanger decreases, the temperature of the Therminol returning to the steam generator also decreases. However, if the saturation pressure in the evaporator is fixed at 100 bar by the turbine bypass system, then the temperature of the Therminol at the cold end of the steam generator either remains constant or increases slightly; i.e., the Therminol heat transfer line pivots about the pinch point

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- The speed of the cold salt pump is increased to the design value. At some point in the transition, the discharge pressure of the hot salt pump falls below the discharge pressure of the cold salt pump, and the check valve in the discharge line from the hot pump closes. The pump is protected from overheating by the flow through the minimum flow recirculation valve. The hot salt pump is then tripped
- At this point, only cold salt is flowing through the heat exchanger. However, the flow of cold salt is near the design value, while the flow of Therminol is near the minimum allowable value. With a Therminol temperature of approximately 265 °C at the cold end of the heat exchanger, little heat transfer occurs, and the entire exchanger soon approaches a uniform temperature which is slightly less than the temperature of the cold salt
- The flow of Therminol is stopped by closing the isolation valve. The cold salt pump is stopped, the positions of the salt isolation valves are changed, and the cold salt pump is restarted. This establishes a flow of salt to the cold end of the heat exchanger. The speed of the cold salt pump is set to the overnight hold value. Cold salt from the hot end of the heat exchanger returns to the cold tank through the heat exchanger bypass line.

7.3 Thermal Storage System

The thermal storage system includes the cold salt tank, the hot salt tank, and the salt inventory. The mass of the salt inventory, and thereby the volumes of the tanks, is largely defined by the number of hours in which the Rankine cycle is required to operate, at full load, from the storage system.

7.3.1 Performance Requirements

Cold Salt Tank

The cold salt tank performs 3 primary functions:

- Provides a volume sufficient to store the entire salt inventory of the plant
- During a storage charge cycle, supplies salt, at a nominal temperature of 293 °C, to the cold end of the oil-to-salt heat exchanger
- During a storage discharge cycle, receives salt, at a nominal temperature of 293 °C, from the cold end of the oil-to-salt heat exchanger.

The normal operating temperature of the tank is the temperature of the salt at the cold end of the oil-to-salt heat exchanger (295 to 305 °C, depending on the approach temperatures selected for the heat

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exchanger). However, the design temperature of the tank is typically selected to be 350 °C. This is a representative value for the temperature of the salt added to the tank during the salt melting process.

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². The insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

- Limit the heat losses from the floor of the tank to values in the range of 60 to 75 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Hot Salt Tank

The hot salt tank performs 3 primary functions:

- Provides a volume sufficient to store the entire salt inventory of the plant
- During a storage charge cycle, receives salt, at a nominal temperature of 388 °C, from the hot end of the oil-to-salt heat exchangers
- During a storage discharge cycle, supplies salt, at a nominal temperature of 388 °C, to the hot end of the oil-to-salt heat exchangers.

The normal operating temperature of the tank is the temperature of the salt at the hot end of the oil-to-salt heat exchanger (388 °C). However, the design temperature of the tank is often selected to be the high-high trip temperature for the collector field (400 °C).

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². As with the cold tank, the insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

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- Limit the heat losses from the floor of the tank to values in the range of 70 to 85 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Inlet Flow Distribution System

Distribution Ring Header

In essentially all commercial projects, the flow is introduced into the cold tank and the hot tank by a circular or an orthogonal ring header located just above the floor. The ring is centered in the tank, and the diameter of the ring is approximately 50 percent of the diameter of the tank. A series of holes or mixing eductors are located along the circumference of the ring to promote mixing between the incoming flow and the bulk inventory.

The design is simple and inexpensive. However, it is not particularly effective in promoting mixing, for the following reasons:

- To a zeroth order, effective mixing occurs only within perhaps 10 pipe diameters of the distribution header. In a representative commercial project, the diameter of the tank is 44 m, the diameter of the header ring is 21 m, and the diameter of the header pipe is 0.6 m. As such, effective mixing occurs only over perhaps 20 to 30 percent of the surface area of the floor
- During morning startup of the collector field, the temperature at the outlet of the field increases from the overnight circulation value (perhaps 150 °C) to the normal design value (393 °C). At some point in the progression, a flow of Therminol is established on the oil side of the oil-to-salt heat exchanger, and a matching flow of salt is established on the salt side. The temperature at which Therminol is introduced in the heat exchanger is a function of the overnight temperature profile in the heat exchanger, and the thermal shock tolerance of the heat exchanger. Nonetheless, the initial temperature of the salt leaving the hot end of the heat exchanger will be less than the design value (388 °C). This, in turn, introduces relatively cold salt into the hot tank. Since the inventory level in the hot tank is typically near the minimum at the start of a charge cycle, the incoming salt will descend towards the floor due to buoyancy effects. If the depth of the inventory is low, then the temperature of the incoming salt may only increase by a small amount before the flow reaches the floor. This produces two undesirable effects:
 - The inventory will thermally stratify, and the stratification will be difficult to remove; i.e., there is a permanent stagnant layer of salt at the floor due to the minimum clearance required between the floor and the bottom of the pump suction bell
 - If the mixing provided by the distribution ring only occurs locally, then there is a greater possibility of establishing a potentially damaging radial temperature gradient in the floor.

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The topic of radial temperature gradients, and the associated radial and tangential stresses in the floor, is discussed in Section 5.6, *Radial Temperature Distributions in the Floor*, of Volume 3 - Narrative.

It can be noted that the problems with buoyancy in the hot tank are generally not replicated in the cold tank. At the end of a discharge cycle, the oil-to-salt heat exchanger is operated in one of two approaches:

- The flows on both the oil side and the salt side are reduced to the minimum, and the heat exchanger is tripped. The goal is to preserve the longitudinal temperature profile for operation the next day. Since the salt flow is tripped, there is essentially no effect on the inventory temperature in the cold tank.
- Alternately, the temperature of the entire heat exchanger can be reduced to the cold salt temperature by adding a flow of cold oil to the flow of hot oil supplied to the hot end of the heat exchanger. Since the change in the metal temperature at the hot end of the heat exchanger is about 100 °C, the time required to cool the heat exchanger is on the order of $100\text{ °C} / 10\text{ °C/min} = 10\text{ min}$. The effect on the inventory temperature in the cold tank will be low because 1) the salt flow rate is at the maximum turndown ratio for the heat exchanger, and 2) the cold tank will be full, or close to full, at the end of a discharge cycle.

Ring Header Supports

In all commercial projects, there is a mechanism for supporting the distribution header. This typically done with vertical pipe supports, approximately 0.5 to 1 m high, located between the floor and the header.

In some projects, the supports are welded to the floor. However, as noted above, when the temperature of the incoming flow is different than the temperature of the bulk inventory, differential thermal expansion between the ring header and the floor will place a combination of shear and bending loads on the floor. The floor is typically thin (6 to 8 mm), and the ability of the floor to resist bending loads, without deformation, may be limited. To avoid this condition, reinforcing plates can be installed between the bottom of the pipe supports and the top of the floor. However, there is now a step change in the thickness of the floor, and this location can act as a stress concentration when the tank undergoes daily thermal expansion and contraction cycles. The problem can be compounded if the welds for the reinforcement plates overlap with the welds for the floor plates.

Alternately, in other projects, the supports are placed on, but are not welded to, the floor. This eliminates the potential for the supports to impose bending loads on the floor. However, erosion has been observed at the bottom of the pipe supports. This is likely due to movement of the ring header associated with the following:

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- Daily thermal expansion and contraction cycles
- Two-phase flow in the vertical line between the top of the tank and the inlet to the ring header. At essentially all flow rates, the pressure drop in the vertical line will be less than 1 m of head per meter of change in elevation. This will produce a two-phase flow, with the potential for oscillations in the flow rate of liquid entering the ring header and vibrations in the line.

In future commercial projects, alternate approaches to flow distribution should be adopted. Potential candidates include the following:

- Provide multiple distribution ring headers, spanning a larger fraction of the floor surface area
- Introduce salt at locations inside the roof or near the top of the wall. This helps to isolate receiver startup transients from the floor, and eliminates the potential of the sharing of forces between the header supports and the floor. The ring header would be replaced with an array of spray nozzles to avoid problems with trying to support the ring header from the roof or from the top of the wall. A note: This approach to flow distribution in the hot tank was demonstrated at the Solar Two project.
- Clearly, introducing salt at the top of the inventory will produce a vertical stratification in the inventory due to 1) a permanent layer of stagnant salt located below the elevation of the suction bells for the salt pumps, and 2) continuous transfer of heat into the foundation. To prevent or to remove the stratification, some form of bulk mixing will be required. Candidates methods might include the following:
 - Pump recirculation, which supply salt to a grid of injection points near the floor
 - Gas compressors, which draw suction from the ullage space in the tank and supply the ullage gas to a grid of injection points near the floor
 - Tank-scale mixing devices, such as a large vertical plate rotating about a vertical axis in the tank, or a large horizontal perforated plate, which vertically traverses the inventory.

7.3.2 Equipment Design

In commercial projects, salt inventories typically range from 20,000 to 40,000 metric tons. Since the vapor pressure of salt is very low (several Pa), the lowest cost storage tank is a flat bottom design with a self-supporting dome roof. The tank is vented to the atmosphere.

At projects sites representative of commercial projects, the allowable soil bearing load is on the order of 240 kPa (5.000 lb_f/ft²). The lowest cost foundation design is one in which the tank is placed directly on compacted, but parent, soil. Given the density of the salt (1,900 kg/m³), the tallest column of salt which can be supported by the parent soil is about 12.7 m. This, in turn, effectively sets a nominal limit on the

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allowable wall height of 12.5 m. To store the required salt inventory, the tank diameter is selected accordingly.

To a first order, the section thicknesses of the wall courses can be calculated from the hoop stress formula ($\text{Thickness} = \text{Pressure} * \text{Diameter} / (2 * \text{Allowable stress})$) and the allowable stresses listed in Section II of the Code.

For the carbon steel used in both the cold tank and the cold salt tank, a post weld heat treatment is required by the Section VIII of the Code for section thicknesses greater than 38 mm (1.5 in.). Specialty contractors can provide heat treatment services, but the process is not without risks:

- The treatment temperature is high enough (650 °C) that the strength of the material decreases to the point where creep deformations are possible
- The iron in the steel can chemically reduce the carbon dioxide in the air. Carbon is infused into the surface of the metal, which increases the strength, but reduces, the ductility of the material.

If the salt inventory requires a tank wall thickness greater than 38 mm, but the engineering contractor would like to avoid the need for heat treatments, the inventory can be divided into two 50-percent capacity tanks.

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8. Operating Requirements - Inorganic Heat Transport Fluids

8.1 Introduction

Operating requirements for parabolic trough projects, using inorganic heat transport fluids, include the following topics:

- Selection of design parameters, such as temperatures, flows, allowable fluxes, rates of temperature change, low cycle fatigue lives, and design Codes
- Evaluation of (Capital cost) versus (Capital cost + Operating cost). The evaluation considers items such as material selection, fabrication techniques, and installed redundancy
- Outline of hardware recommendations and specifications.

8.2 Collector Field

8.2.1 Performance Requirements

Temperatures

The operating temperatures of interest are the collector field outlet temperature and the collector field inlet temperature.

Collector Field Outlet Temperature

The field outlet temperature should, in general, be as high as possible to provide a Rankine cycle efficiency that is as high as possible. However, the thermal losses from the receivers are a function of the absorber temperature, and at high temperature, radiation losses become the predominate loss.

To help define an upper limit on the field outlet temperature, the following representative design parameters can be adopted ⁷:

- 5.75 m collector aperture width
- 0.75 collector optical efficiency

⁷ Burkholder, F., and Kutcher, C. (National Renewable Energy Laboratory, Golden, Colorado), Heat Loss Testing of Schott's 2008 PTR70 Parabolic Trough Receiver, Technical Report NREL/TP-550-45633, May 2009

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- 0.960 fraction of the receiver not shaded by the bellows
- 0.963 transmittance of the glass envelope
- Incidence angle modifier, given by $\cos(\theta) + 0.000884 * \theta - 0.00005369 * \theta^2$

Under ideal conditions, with the sun perpendicular to the aperture and with a direct normal radiation of $1,000 \text{ W/m}^2$, the unit thermal input to the receiver is $3,987 \text{ W/m}$.

The unit heat loss from the receiver, in W/m , is given by the expression $0.141 * T_{\text{abs}} + 6.48 \times 10^{-9} * T_{\text{abs}}^4$, where T_{abs} is the absorber temperature in $^{\circ}\text{C}$. For those collectors located at the inlet to a loop, the absorber temperature is on the order of 300°C , and the unit heat loss is about 95 W/m . This results in a receiver thermal efficiency of $(3,987 \text{ W/m} - 95 \text{ W/m}) / 3,987 \text{ W/m} = 0.976$. Repeating the calculations for absorber temperatures in the range of 300 to 550°C results in the red line shown in Figure 8-1.

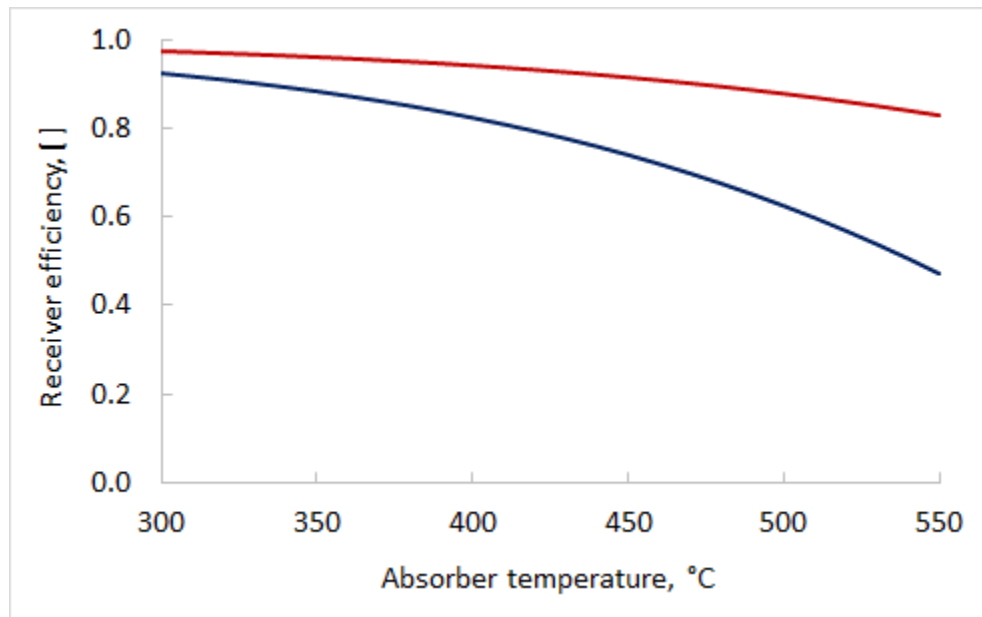


Figure 8-1 Receiver Efficiency as a Function of Absorber Temperature

Efficiency values can also be calculated for less favorable radiation conditions. If the angle of incidence between the sun and the aperture is 45 degrees, and if the direct normal radiation is representative of cloudy conditions (500 W/m^2), then the receiver efficiency decreases to the values shown in the blue line.

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To a first order, a trough project using nitrate salt as the working fluid must have a product of (Receiver efficiency) * (Rankine cycle efficiency) that is at least as high as a commercial trough project using Therminol as the working fluid.

For a trough project using Therminol, the field inlet and outlet temperatures are 293 °C and 393 °C, respectively. With the favorable radiation conditions noted above, the average receiver efficiency is 0.964; with the less radiation favorable conditions, the average decreases to 0.887. A representative value for the gross Rankine cycle efficiency is 0.380. This, in turn, results in a product of 0.366 for the (Receiver efficiency) * (Rankine cycle efficiency) at the favorable radiation conditions, and a product of 0.337 for the less favorable conditions.

Repeating the calculations for a trough project using nitrate salt as the working fluid, under favorable radiation conditions, results in the values shown in Table 8-1.

Table 8-1 Collector Field * Rankine Cycle Efficiencies - Favorable Radiation Conditions

	Therminol	Nitrate Salt - 500 °C	Nitrate Salt - 550 °C
Average receiver efficiency	0.964	0.935	0.922
Gross Rankine cycle efficiency	0.380	0.420	0.425
Product	0.366	0.393	0.392

With collector field outlet temperatures in the range of 500 to 550 °C, selecting nitrate salt as the working fluid offers a nominal 7 percent improvement in overall plant efficiency.

Under cloudy conditions, the results are summarized in Table 8-2.

Table 8-2 Collector Field * Rankine Cycle Efficiencies - Cloudy Conditions

	Therminol	Nitrate Salt - 500 °C	Nitrate Salt - 550 °C
Average receiver efficiency	0.887	0.798	0.755
Gross Rankine cycle efficiency	0.380	0.420	0.425
Product	0.337	0.335	0.321

In contrast, under cloudy conditions, selecting nitrate salt as the working fluid, operating at 500 °C, offers essentially no improvement compared to a commercial trough project, and the project can experience a decrease in performance at temperatures above 500 °C.

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Certainly, this zeroth order analysis does not consider radiation conditions and solar incidence angles over the course of a year, and it does not account for the relative differences in capital costs between projects using Therminol and nitrate salt. But it does indicate that collector field outlet temperatures up to 500 °C can help to reduce the cost of electric energy, but outlet temperatures on the order of 550 °C and above are not likely to offer an economic benefit.

For a first-of-a-kind trough project using nitrate salt as the working fluid, collector field outlet temperatures in the range of 520 to 540 °C are likely to be, or close to, the optimum.

Collector Field Inlet Temperature

The cold salt temperature is defined by the design conditions for the Rankine cycle, as follows:

- The unit work performed by the expansion of steam is increased by increasing the pressure ratio across the turbine. However, the expansion ratio cannot be so high as to result in excessive moisture (more than 8 or 9 percent) in the exhaust of the low pressure turbine. With a hot salt temperature of 540 °C, a representative live and reheat steam temperature is 515 °C. These temperatures, in turn, result in live steam pressures of 120 bar to a maximum of perhaps 130 bar
- The efficiency of the Rankine cycle is improved by increasing the final feedwater temperature. However, raising the final feedwater temperature increases the salt temperature at the cold end of the preheater in the steam generator, which, in turn, reduces the temperature difference between the cold salt tank and the hot salt tank. Optimum final feedwater temperatures are typically in the range of 220 to 240 °C.
- To provide a heat transfer area in the evaporator of the steam generator that can be safely warranted by the vendor, a pinch point of 5 °C is typically selected
- To help prevent water evaporation (steaming) at the hot end of the preheater during low load operation, the preheater is typically designed with a 5 °C approach to saturation.

The requirements above typically result in cold salt temperatures in the range of 295 to 305 °C, depending on the final selection of the live steam pressure and the final feedwater temperature.

Pressures

Commercial receiver are fabricated with tube outside diameters of 70, 80, and 90 mm. For the 70 mm and the 80 mm tubes, the wall thickness is 2.0 mm. For the 90 mm design, the wall thickness is 2.5 mm.

The maximum allowable pressure at the inlet to the field is 38 bar based on 1) the allowable stress for the tube material (Type 321 stainless steel) at the field outlet temperature, and 2) the flow at the outlet of

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the field is blocked; i.e., the fluid pressure at the outlet of the field is equal to the pressure at the inlet of the field.

The maximum pressure, in combination with the desired turndown ratio for the flow rate, effectively defines the allowable length of the loop.

Flow Velocities

The maximum salt velocities, in combination with the length of the loop, are defined by the following:

- The maximum allowable pressure at the inlet to the field.
- The allowable erosion rates of stainless steel. However, the erosion rates of stainless steel tubes are generally not well understood.

In many commercial designs, the peak velocity is limited to about 4 m/sec, as this is taken to be a 'safe' value in terms of the potential for erosion and tube vibration.

It can be noted that the number of hours each year in which the peak velocities are reached are limited to those hours in which the product of the direct normal radiation and the incidence angle modifier is equal to, or greater than, the design value.

Turn Down Ratio

To capture as much of the annual site radiation as possible, the receiver would, under ideal conditions, be able to operate with an infinite turndown ratio. However, if the Reynolds number falls below the minimum value specified by the vendor (5,000), then the internal heat transfer coefficient is no longer sufficient to maintain the temperature difference between the front of the tube (illuminated) and the back of the tube (not illuminated) within allowable values. This, in turn, places the front of the tube in compression, and the back of the tube in tension, at stress levels high to either permanently deform the tube, or to push the tube against the outer envelope with sufficient force to break the glass. These considerations effectively limit the turn down ratio to a maximum value of perhaps 6:1.

Thermal Efficiency

The thermal efficiency of the collector field is an instantaneous function of the following parameters (representative values in parenthesis):

- Direct normal radiation (300 to 1,065 W/m²)

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- Angle of incidence between the direct normal radiation and the collector aperture (0 degrees to 70 degrees)
- Collector optical efficiency (0.75, including the mirror reflectivity and the receiver intercept factor)
- Fraction of the receiver length not shaded by the bellows (0.960)
- Transmittance of the glass envelope (0.963)
- Local wind speeds (0 m/sec to 15 m/sec)
- Local absorber temperature, which influences the absorptivity and the emissivity of the selective surface coating on the receiver tube.

Sunlight-to-thermal efficiencies reach a maximum of about 0.675 under the following conditions:

- 300 °C absorber temperature
- 1,065 W/m² aperture normal radiation
- 0 m/sec wind speed.

Conversely, thermal efficiencies, on winter mornings with cloudy conditions, can reach negative values for the highest temperature receivers near the end of the loop. At these locations, the heat losses exceed the radiation received at the aperture.

Fatigue Life

During the 30-year life of the project, the collector loop operates under the following range of cyclic conditions:

- 35,000 temperature cycles, in which the metal temperature at the cold end of the collector loop starts at, and remains constant at, 295 °C, and the metal temperature at the hot end of the collector loop changes from 295 °C to 540 °C. The 35,000 cycles consist of 15,000 daily startup cycles and 20,000 cloud transient cycles
- 150 trip cycles, in which the metal temperature at the cold end of the collector loop remains constant at 295 °C, and the metal temperature at the hot end of the collector loop changes from

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540 °C to 295 °C. The rate of temperature change at the hot end of the receiver is approximately 100 °C/min.

The receiver is designed to withstand the cumulative creep and fatigue damage associated with these cycles.

8.2.2 Equipment Design

Materials

Commercial receiver tubes are fabricated with Type 321 stainless steel, and are coated with a proprietary Cermet coating to provide a selective surface. The external enclosure is a borosilicate glass, to which is applied an anti-reflection coating.

Redundancy

The length of a collector loop is based on thermodynamic properties of the working fluid, the temperature rise across the loop, and limits on minimum and maximum flow velocities. In commercial projects, the product of the aperture width times the loop length is on the order of 4,000 to 6,000 m².

The total aperture area in a commercial project is a function of the plant rating, the capacity of the storage system, and the site weather characteristics. Representative aperture areas can range from 300,000 m² to more than 1,000,000 m². As such, the number of loops can range from 75 to several hundred.

Since each loop is hydraulically connected in parallel with the balance the loops, the collector field has a high degree of redundancy.

8.3 Thermal Storage System

The thermal storage system includes the cold salt tank, the hot salt tank, and the salt inventory. The mass of the salt inventory, and thereby the volumes of the tanks, is largely defined by the number of hours in which the Rankine cycle is required to operate, at full load, from the storage system.

8.3.1 Performance Requirements

Cold Salt Tank

The cold salt tank performs 5 primary functions:

- Provides a volume sufficient to store the entire salt inventory of the plant

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- Receives salt, at a nominal temperature of 295 °C, from the steam generator
- Receives salt, at temperatures in the range of 295 to 500 °C, from the collector field during daily startup and at the end of a cloud transient
- Provides a source of cold salt to the collector field
- Provides a source of cold salt to the steam generator during startup and shut down.

The normal operating temperature of the tank is the temperature of the salt at the cold end of the steam generator preheater (295 to 305 °C, depending on the Rankine cycle design). However, the design temperature of the tank is typically selected to be 370 °C. This allows the temperature of inventory to increase by as much as 75 °C during extended receiver startup periods typical of cloudy weather.

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². The insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

- Limit the heat losses from the floor of the tank to values in the range of 60 to 75 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Hot Salt Tank

The hot salt tank performs 4 primary functions:

- Provides a volume sufficient to store the entire salt inventory of the plant
- Receives salt, at a nominal temperature of 565 °C, from the receiver
- Receives salt, at temperatures in the range of 450 to 565 °C, from the receiver during receiver startup and following a receiver trip. This requirement is notably quite stringent. The tank inlet distribution system must provide sufficient mixing with the bulk inventory to prevent potentially damaging values of intra-tank temperature differentials. The differentials may only persist for

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short periods (seconds), but it is the instantaneous rates of temperature change that lead to low cycle fatigue damage

- Provides a source of hot salt to the steam generator.

The normal operating temperature of the tank is the temperature of the salt at the hot end of the receiver (540 °C). However, the design temperature of the tank is typically selected to be the high-high trip temperature for the receiver (600 °C).

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². As with the cold tank, the insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

- Limit the heat losses from the floor of the tank to values in the range of 70 to 85 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Inlet Flow Distribution System

Distribution Ring Header

In essentially all commercial projects, the flow is introduced into the cold tank and the hot tank by a circular or an orthogonal ring header located just above the floor. The ring is centered in the tank, and the diameter of the ring is approximately 50 percent of the diameter of the tank. A series of holes or mixing eductors are located along the circumference of the ring to promote mixing between the incoming flow and the bulk inventory.

The design is simple and inexpensive. However, it is not particularly effective in promoting mixing, for the following reasons:

- To a zeroth order, effective mixing occurs only within perhaps 10 pipe diameters of the distribution header. In a representative commercial project, the diameter of the tank is 44 m, the diameter of the header ring is 21 m, and the diameter of the header pipe is 0.6 m. As such, effective mixing occurs only over perhaps 20 to 30 percent of the surface area of the floor

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- During receiver startup, the inventory level in the hot tank is typically near the minimum level. Further, the temperature of the incoming flow is at least 20 °C, and may be as much as 100 °C, lower than the temperature of the bulk inventory. Due to buoyancy effects, the incoming salt will largely descend towards the floor. If the depth of the inventory is low, then the temperature of the incoming salt may only increase by a small amount before the flow reaches the floor. This can result in local, and potentially damaging, thermal stresses due to temperature gradients.

It can be noted that the problems with buoyancy in the hot tank are generally not replicated in the cold tank. During receiver startup, salt, at temperatures in the range of 295 °C to perhaps 510 °C, are introduced into the cold tank. However, this typically has moderate effects on the tank, as follows:

- During morning startup, the inventory level in the cold tank is typically full, or close to full. As such, the thermal inertia of the inventory (mass * specific heat) is high, and the inventory can accept salt at relatively high temperatures without a significant change in the inventory temperature
- Since the initial temperature of the bulk inventory is close to 295 °C, the flow of relatively hot salt rises into the bulk inventory. This has the effect of damping the transient temperature changes imposed on the floor and the wall.

Ring Header Supports

In addition, there needs to be some mechanism for supporting the distribution header. This typically done with vertical pipe supports, approximately 0.5 to 1 m high, located between the floor and the header.

In some projects, the supports are welded to the floor. However, as noted above, when the temperature of the incoming flow is different than the temperature of the bulk inventory, differential thermal expansion between the ring header and the floor will place a combination of shear and bending loads on the floor. The floor is typically thin (6 to 8 mm), and the ability of the floor to resist bending loads, without deformation, may be limited. To avoid this condition, reinforcing plates can be installed between the bottom of the pipe supports and the top of the floor. However, there is now a step change in the thickness of the floor, and this location can act as a stress concentration when the tank undergoes daily thermal expansion and contraction cycles. The problem can be compounded if the welds for the reinforcement plates overlap with the welds for the floor plates.

Alternately, in other projects, the supports are placed on, but are not welded to, the floor. This eliminates the potential for the supports to impose bending loads on the floor. However, erosion has been observed at the bottom of the pipe supports. This is likely due to movement of the ring header associated with the following:

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- Daily thermal expansion and contraction cycles
- Two-phase flow in the vertical line between the top of the tank and the inlet to the ring header. At essentially all flow rates, the pressure drop in the vertical line will be less than 1 m of head per meter of change in elevation. This will produce a two-phase flow, with the potential for oscillations in the flow rate of liquid entering the ring header and vibrations in the line.

In future commercial projects, alternate approaches to flow distribution should be adopted. Potential candidates include the following:

- Provide multiple distribution ring headers, spanning a larger fraction of the floor surface area
- Introduce salt at locations inside the roof or near the top of the wall. This helps to isolate receiver startup transients from the floor, and eliminates the potential of the sharing of forces between the header supports and the floor. The ring header would be replaced with an array of spray nozzles to avoid problems with trying to support the ring header from the roof or from the top of the wall. A note: This approach to flow distribution in the hot tank was demonstrated at the Solar Two project.
- Clearly, introducing salt at the top of the inventory will produce a vertical stratification in the inventory due to 1) a permanent layer of stagnant salt located below the elevation of the suction bells for the salt pumps, and 2) continuous transfer of heat into the foundation. To prevent or to remove the stratification, some form of bulk mixing will be required. Candidates methods might include the following:
 - Pump recirculation, which supply salt to a grid of injection points near the floor
 - Gas compressors, which draw suction from the ullage space in the tank and supply the ullage gas to a grid of injection points near the floor
 - Tank-scale mixing devices, such as a large vertical plate rotating about a vertical axis in the tank, or a large horizontal perforated plate, which vertically traverses the inventory.

8.3.2 Equipment Design

In commercial projects, salt inventories typically range from 20,000 to 40,000 metric tons. Since the vapor pressure of salt is very low (several Pa), the lowest cost storage tank is a flat bottom design with a self-supporting dome roof. The tank is vented to the atmosphere.

At projects sites representative of commercial projects, the allowable soil bearing load is on the order of 240 kPa (5,000 lb_f/ft²). The lowest cost foundation design is one in which the tank is placed directly on compacted, but parent, soil. Given the density of the salt (1,900 kg/m³), the tallest column of salt which can be supported by the parent soil is about 12.7 m. This, in turn, effectively sets a nominal limit on the

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allowable wall height of 12.5 m. To store the required salt inventory, the tank diameter is selected accordingly.

To a first order, the section thicknesses of the wall courses can be calculated from the hoop stress formula (Thickness = Pressure * Diameter / (2 * Allowable stress)) and the allowable stresses listed in Section II of the ASME Boiler and Pressure Vessel Code. For the stainless steel used in the hot salt tank, there are no fabrication restrictions noted in the Code. However, for the carbon steel used in the cold salt tank, a post weld heat treatment is required for section thicknesses greater than 38 mm (1.5 in.). Specialty contractors can provide heat treatment services, but the process is not without risks:

- The treatment temperature is high enough (650 °C) that the strength of the material decreases to the point where creep deformations are possible
- The iron in the steel can chemically reduce the carbon dioxide in the air. Carbon is infused into the surface of the metal, which increases the strength, but reduces, the ductility of the material.

If the salt inventory requires a cold tank wall thickness greater than 38 mm, but the engineering contractor would like to avoid the need for heat treatments, the inventory can be divided into two 50-percent capacity tanks.

8.4 *Steam Generation System*

The steam generator converts the thermal energy in the hot salt to a combination of live steam and reheat steam that is supplied to the Rankine cycle for power generation.

8.4.1 *Process Requirements*

The steam generator receives a flow of feedwater from the Rankine cycle. The feedwater is at a pressure of 130 to 145 bar, and at a temperature of 220 to 245 °C, depending on the cycle design. The nominal flow rate is 1 kg/sec per MWe of electric power production.

The steam generator preheats the feedwater to the saturation temperature, converts the feedwater to saturated steam, and superheats the steam to a nominal temperature of 520 °C.

A portion of the saturated water (~ 1 percent) is removed from the evaporator section for water chemistry control.

After the live steam has expanded in the high pressure steam turbine by a ratio of 4 to 5, the steam is returned to the steam generator for reheating. The steam is reheated to a nominal temperature of 520 °C, and then expanded by a factor of about 200 through the intermediate- and the low-pressure sections of the turbine.

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Daily Startup and Shutdown

The steam generator in a project using an inorganic heat transfer fluid operates in a similar manner to the salt steam generator in a central receiver project (Section 7.4 of Volume 2 - Specifications for Central Receiver Projects). The startup and shutdown process involves the following steps:

- During overnight hold, cold salt from the attemperation pump flows through each of the heat exchangers. The cold salt is returned to the cold tank
- In the first phase of startup, hot salt is mixed with cold salt at a point upstream of the hot end of the evaporator. This establishes a flow of saturated steam in the superheater and in the reheater equal to about 20 minimum of the design flow rate. The flow rate is slightly greater than the minimum allowable flow rate specified by the vendor (16 percent). No heat transfer occurs in either the superheater or the reheater during this phase
- In the second phase of startup, the relative flow rates of hot salt and cold salt remain fixed, and the salt mixing station is moved from a point upstream of the hot end of the evaporator to the normal point upstream of the hot ends of the superheater / reheater. Superheating now occurs in the superheater and in the reheater. Due to the superheating duties, the steam flow rate decreases to 16 percent
- The relative proportions of hot salt and cold salt are adjusted by changing the relative speeds of the hot salt pump and the attemperation pump. When the flow of cold salt has reached a value of zero, the steam generator is in normal operation
- The process is essentially reversed to shut down the steam generator.

Minimum Flow Rate

During the startup and the shutdown process, the flow rates on both the shell-sides and the tube-sides of the heat exchangers must satisfy the minimum allowable flow rates (a nominal 16 percent) specified by the vendor.

The flows on the shell sides of the heat exchanger can satisfy the vendor requirement by setting the sum of the hot salt flow rate and the cold salt flow rate to a value that is always equal to, or greater, than the vendor limit. If the evaporator is a forced recirculation design, then the flows on the tube sides of the preheater and the evaporator can satisfy the vendor limits, as follows:

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- A preheater recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the preheater. The combination of the feedwater flow rate plus the recirculation flow rate always satisfies the vendor requirement
- An evaporator recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the evaporator. The combination of the feedwater flow rate plus the recirculation flow rate can be used to satisfy the vendor requirement
- During startup, there is no mechanism to recirculate steam in the superheater and the reheater. As such, the steam flow rate necessarily passes through the range of 0 percent to the vendor minimum of 16 percent. At the flow rates in this range, the steam is unlikely to be uniformly distributed among the several hundred tubes in each heat exchanger. Non-uniform flow distributions can produce non-uniform temperature distributions, which, in turn, can lead to unpredictable stress distributions. However, if the superheater and the reheater are operated with the same temperatures on the tube side and the shell side, then no heat transfer occurs and potentially damaging stress distributions can be avoided. As noted above, the use of a salt mixing station upstream of the hot end of the evaporator allows the minimum tube-side flow requirement to be met, at which point the mixing station can be safely moved to the normal point upstream of the superheater / reheater.

On a related point, the duty of the steam generator necessarily passes through the range of 0 percent to 16 percent during both startup and shut down. Establishing the minimum allowable tube- and shell-side flow rates, or establishing isothermal operation, throughout the duty range of 0 percent to 16 percent helps to protect the heat exchangers from unpredictable, and potentially damaging, stress distributions.

Maximum Allowable Rate of Temperature Change

The heat exchanger consists of a number of parts which are joined either by welding or by plastic deformation. In addition, the parts generally span a wide range of section thicknesses, ranging from less than 2 mm for the tube walls to perhaps 200 mm for the tubesheet. During transient conditions, the thin metal sections change temperature more quickly than the thick metal sections. The different response times produce different rates of thermal expansion, which lead to local thermal stresses. The transient stresses are often additive to the normal process stresses due to temperature and pressure.

The sum of the normal process stresses and the transient thermal stresses can exceed the allowable stress values listed in Section II of the ASME Boiler and Pressure Vessel Code. As such, the fatigue life of the heat exchanger can be less than infinite. To maintain a low cycle fatigue life consistent the life of the project, the vendor will define a maximum allowable rate of temperature change. Typical values range from 8 to 12 °C/min.

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It can be noted that operating the heat exchangers at rates of temperature change higher than the vendor's limit will produce transient stresses which are above those than consistent with a fatigue life of 30 years. Further, the decrease in the fatigue life associated with an increase in the transient thermal stress is exponential, and even small excursions in the rate of temperature change can significantly reduce the fatigue life of the equipment.

8.4.2 Performance Requirements

The steam generator must meet the following performance requirements:

- The saturated steam at the turbine control valves complies with the steam quality standards set by the turbine vendor
- The heat exchanger designs meet the Tubular Equipment Manufacturers Association requirements for items including flow distribution, tube vibration, and tubesheet temperature gradients, and allowances for spare tubes
- The heat exchangers have a fatigue life consistent with at least 10,000 startup and shutdown cycles (i.e., 30 years * 330 cycles per year).

At some commercial projects, it has proven difficult to meet the vendor's limit on the allowable rate of temperature change startup and shutdown. To provide a margin against potentially inaccurate process temperature control, it may be beneficial to provide a fatigue life of 50,000 to 100,000 cycles for the heat exchangers.

8.4.3 Equipment Design

All commercial steam generator designs are based on a common arrangement, as follows:

- The evaporator uses natural or forced recirculation, in combination with a separate steam drum. A solar project necessarily cycles each day, and the condensate water chemistry in the condenser hotwell or condensate tank typically degrades during the overnight hold period. Reestablishing the water chemistry during morning startup for a drum-type evaporator is less complex and less time consuming than reestablishing the more stringent water chemistry required for a single-pass-to-superheat steam generator
- A preheat heat exchanger, separate from and upstream of the water side of the evaporator, performs two functions:
 - Raises the temperature of the feedwater close to the saturation point, which allows the evaporator to operate under isothermal conditions on the water side

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- Reduces the temperature of the salt leaving the steam generator to a value which is less than the saturation temperature, which effectively reduces the cost of the thermal storage system
- The cold reheat steam temperature (285 to 300 °C) is the same order of magnitude as the saturation temperature in the evaporator (335 to 350 °C). As such, the temperature of the salt at the cold end of the reheater is the same order of magnitude as the temperature of the salt at the cold end of the superheater. As a consequence, the salt flow from the cold end of the reheater can be combined with the salt flow from the cold end of the superheater, and the combined flow can be directed, in series, through the evaporator and the preheater. In this manner, energy can be extracted from the salt flow through the reheater to the same extent as energy can be extracted from the salt flow through the superheater. This, in turn, provides the lowest cold salt temperature of any of the candidate steam generator configurations
- On the salt side of the steam generator, the superheater operates in parallel with the reheater. With this arrangement, the hot reheat steam temperature can be as high as the live steam temperature. This, in turn, maximizes the live steam pressure (up to the point where the moisture content in the turbine exhaust reaches maximum allowable values of 8 to 9 percent). Since the unit work of the steam passing through the turbine is $\int (\text{Specific volume}) * d(\text{Pressure})$, maximizing the live steam pressure maximizes the cycle efficiency.

In general, the shell-side fluid is selected to be salt, and the tube-side fluid is selected to be water/steam. This arrangement is adopted for two reasons:

- Should the salt freeze in a heat exchanger, the thawing process will involve a change in the salt volume of about 4.6 percent. Were the salt to be constrained in a tube or in a vessel, the change in tube or vessel diameter would be on the order of 2.3 percent. This is beyond the yield strain for both carbon and stainless steels, and plastic deformations of this amount can significantly reduce the fatigue life of the equipment. One method to reduce the potential for damage during thawing is to place the salt on the shell side. The thawing process would involve the following steps:
 - Raise the temperature of the salt lines to and from the vessel to at least 275 C. This ensures the salt in the lines is liquid, and any salt melted in the exchanger has an exit path from the vessel
 - Activate the electric heat tracing on the shell. Radial heat transfer into the shell will melt the salt on the inside surface of the vessel, and the melting salt has an exit path from the heat exchanger through the salt lines
 - Simultaneously supply heat to the tube bundle by means of the electric water heaters. In the preheater and in the evaporator, the heat will be supplied by the recirculation of water. In the superheater and in the reheater, the heat will be supplied by a flow of saturated steam from the drum. The temperature on the water side must be maintained

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within a few degrees of the temperature on the outside of the shell. This is to prevent the establishment of radial temperature gradients in the tubesheet that could either damage the tubesheet or flex the tubesheet to the extent that the tube-to-tubesheet connections could relax

- The water/steam operates at much higher pressures than the salt, and thin wall tubes are more adept at withstanding high pressures than thick wall vessels.

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9. Functional / Prescriptive Specifications - Organic Heat Transport Fluids

9.1 Introduction

Plant and equipment specifications fall into two broad categories:

- Functional specifications describe what the system or the equipment needs to do, consistent with the minimum legal requirements of the local jurisdictions. The details of how this is to be accomplished is developed by the engineering contractor. This allows the engineering contractor to define the process and the equipment designs such that the functional requirements can be met at the lowest cost
- Prescriptive specifications, which are developed by the Owner, prescribe to the engineering contractor how the functional requirements of the project are to be met. Prescriptive elements include items such as the process design, Code Section selection, fabrication techniques, materials selection, and component details. This arrangement ensures that the favorable experience from a previous project is repeated, and helps to avoid situations in which an inexperienced contractor repeats mistakes from earlier projects.

It can be noted that the use of prescriptive specifications often requires the following:

- The Owner must have a thorough understanding of the process and the technical features of the plant. For example, the heat trace designs in commercial projects often use 1 dual-element thermocouple in each zone as the input to the temperature set point controller. However, a heat trace design, which reflects the experience from previous projects, might use 2 to 4 thermocouples, distributed along the length of the zone, as inputs to the controller. This arrangement can help to detect defects in the insulation, local areas with temperatures below the freezing point of the salt, local areas which are receiving too much heat input from adjacent zones, and leakage past isolation valves.
- The Owner is prepared to accept what is likely to be an increase in the capital cost as a means of achieving the goals set in the performance and in the financial models. For example, the Owner may require the use of all-welded construction in salt heat exchangers. This fabrication approach is more expensive than the conventional approach; i.e., tube rolling followed by strength welding of the tube to the tubesheet. However, an all-welded design is more tolerant of the transient stresses associated with 1) daily startup and shutdown, and 2) mistakes made by the operators. In turn, the improvement in plant availability more than compensates for the marginal increase in the cost of the heat exchangers due to an all-welded design.

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In the sections below is a list of the Design Codes and Standards, followed by a discussion of recommended prescriptive specifications.

9.2 *Design Codes and Standards*

For the salt components and equipment in a commercial project, reference codes and standards are listed below for the US market. If the plant is constructed outside of the US, an equivalent matrix will need to be developed for the host country.

<u>Designation</u>	<u>Title</u>
API	Standard 650, Welded Steel Tanks for Oil Storage
ASME	B31.1, Power Piping
ASME	Section I, Rules for the Construction of Power Boilers
ASME	Section II, Materials
ASME	Section V, Non Destructive Examination
ASME	Section VIII, Division 1, Rules for the Construction of Pressure Vessels
ASME	Section VIII, Division 2, Alternative Rules for the Construction of Pressure Vessels
ASTM	A105, Specification for Forgings, Carbon Steel, for Piping Components
ASTM	A181, Specification for Forgings, Carbon Steel for General Service
ASTM	A182, Specification for Forged or Rolled Alloy-Steel Pipe Flanges, Forged Fittings, and Valves and Parts for High-Temperature Service
ASTM	A192, Specification for Seamless Carbon Steel Boiler Tubes for High-Pressure Service
ASTM	A193, Specification for Alloy-Steel and Stainless Steel Bolting Materials for High-Temperature Service
ASTM	A194, Specification for Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service
ASTM	A213, Specification for Seamless Ferritic and Austenitic Alloy-Steel Boiler, Superheater, and Heat Exchanger Tubing
ASTM	A216, Specification for Steel Castings, Carbon, Suitable for Fusion Welding, for High-Temperature Service
ASTM	A240, Specification for Heat-Resisting Chromium and Chromium-Nickel Stainless Steel Plate, Sheet, and Strip for Pressure Vessels
ASTM	A312, Specification for Seamless and Welded Austenitic Stainless Steel Pipe
ASTM	A325, Specification for Structural Steel Bolts, Steel, Heat Treated, 120/125 ksi Minimum Tensile Strength
ASTM	A351, Specification for Castings Austenitic Austenitic-Ferritic (Duplex) for Pressure-Containing Parts
ASTM	A36, Specification for Carbon Structural Steel
ASTM	A387, Specification for Pressure Vessel Plates, Alloy Steel, Chromium-Molybdenum
ASTM	A403, Specification for Wrought Austenitic Stainless Steel Piping Fittings

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ASTM	A500, Specification for Cold-Formed Welded and Seamless Carbon Steel Structural Tubing in Rounds and Shapes
ASTM	A506, Specification for Steel, Sheet and Strip, Alloy, Hot-Rolled and Cold-Rolled, Regular Quality and Structural Quality
ASTM	A516, Specification for Pressure Vessel Plates, Carbon Steel, for Moderate- and Lower-Temperature Service
ASTM	A53, Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated Welded and Seamless
ASTM	A556, Specification for Seamless Cold-Drawn Carbon Steel Feedwater Heater Tubes
HEI	Heat Exchange Institute
NEC	National Electric Code
NEMA	National Electrical Manufacturers Association
NFPA	National Electric Code (NEC), National Fire Protection Association (NFPA)
TEMA	Tubular Exchanger Manufacturers Association, 8th Edition TEMA Standards
IBC	International Building Code

9.3 Nitrate Salt Specification

As discussed in Section 2.4.1, thermal storage systems in parabolic trough projects are an indirect design, in which the working fluid in the collector field (Therminol) is different than the working fluid in the storage system (an inorganic salt).

There are a wide range of inorganic salts suitable for use as the storage medium, including the following 6 candidates:

- Binary nitrate salt (solar salt) mixture, which is 60 weight percent sodium nitrate (NaNO_3) and 40 weight percent potassium nitrate (KNO_3)
- Ternary nitrate salt mixture, which, with contributions of 30 weight percent lithium nitrate (LiNO_3), 18 weight percent sodium nitrate (NaNO_3), and 52 weight percent potassium nitrate (KNO_3), offers the lowest melting point.
- Ternary nitrate/nitrite salt mixture, the most common of which is Hitec[®], with contributions of 7 weight percent sodium nitrate (NaNO_3), 40 weight percent sodium nitrite (NaNO_2), and 53 weight percent potassium nitrate (KNO_3)
- A ternary nitrate salt mixture, Hitec XL[®], with contributions of 42 weight percent calcium nitrate ($\text{Ca}(\text{NO}_3)_2$), 15 weight percent sodium nitrate (NaNO_3), and 43 weight percent potassium nitrate (KNO_3)

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- A quaternary nitrate salt mixture, reviewed by Sandia National Laboratories, with contributions of 22 weight percent lithium nitrate (LiNO_3), 17 weight percent calcium nitrate ($\text{Ca}(\text{NO}_3)_2$), 18 weight percent by weight sodium nitrate (NaNO_3), and 43 weight percent potassium nitrate (KNO_3)
- A quinary nitrate/nitrite salt mixture, patented by Sandia National Laboratories, with contributions of 13 weight percent lithium nitrate (LiNO_3), 10 weight percent by weight sodium nitrate (NaNO_3), 12 weight percent potassium nitrate (KNO_3), 29 weight percent sodium nitrite (NaNO_2), and 36 weight percent potassium nitrite (KNO_2).

The ideal salt would have 1) a high upper temperature limit in terms of thermal stability, 2) a low freezing point, and 3) be inexpensive. As one might expect, none of the candidate salts fulfill all of the ideal requirements.

In a parabolic trough project, representative cold tank and hot tank temperatures are 292 °C and 385 °C, respectively. Each of the 6 candidate salts have upper temperature limits of at least 450 °C, and can readily satisfy the thermal stability requirements at the temperatures characteristic of the hot tank.

The cold tank temperature is dictated by the 1) final feedwater temperature of the Rankine cycle (~225 °C), 2) the approach temperature at the cold end of the steam generator economizer (55 to 60 °C), and the approach temperature at the cold end of the oil-to-salt heat exchanger (7 to 8 °C). The ternary nitrate salt, Hitec, Hitec XL, the quaternary nitrate salt, and the quinary nitrate/nitrite salt have freezing points which are 100 to 150 °C lower than the freezing point of the 60/40 salt mixture. However, the lower freezing points cannot be exploited in the storage system because the lowest system temperature (290 °C) is at least 50 °C higher than the candidate salt with the highest freezing point (~235 °C for the 60/40 mixture).

As such, the choice among the 6 candidate salts distills to one of cost, and the 60/40 mixture will always offer the lowest commodity pricing.

9.3.1 Grades

Sodium nitrate is available in refined, technical, and industrial grades, and potassium nitrate is available in refined and technical grades. The refined grades have the lowest impurities and the highest cost, while the industrial grades have the highest impurities and the lowest cost.

The principal impurity of interest are the chlorides, either in the form of chloride ions or the perchlorate. The mechanism for corrosion by chlorine and chloride was suggested by Grabke et al ⁸. Specifically,

⁸ Grabke, H.J., E. Reese, and M. Spiegel, *Effects of chlorides, hydrogen chloride, and sulfur dioxide in the oxidation of steels below deposits*. Corrosion Science, 1995. 37(7): p. 1023-1043

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chlorine forms at the surface of the metal in contact with the salt. Through diffusion, chlorine congregates at the boundary between the inside of the corrosion scale and the parent metal. The chlorine reacts with the metallic elements, forming metal chlorides. The metal chlorides, which have high vapor pressures, evaporate and diffuse back out through the corrosion scale layer⁹. The metal chlorides, in contact with the salt, are oxidized to various metal oxides. This releases the chlorine to repeat the corrosion process. The chlorides act, in effect, as catalysts to accelerate the oxidation rates of the parent metal. In the paper by Grabke, the process was explained specifically in reference to iron based alloys, where various oxides, such as hematite (Fe₂O₃) or magnetite (Fe₃O₄), are formed.

In general, the technical grades of nitrate salt are suitable for parabolic trough applications, offering an acceptable compromise between purity, chloride corrosion rate, and cost.

A typical technical grade specification should include the following items:

- Minimum nitrate concentration of 99 percent by weight
- Maximum total chloride ion concentration from all sources of 0.2 percent by weight.

9.3.2 Total Chloride Concentration

An example of the calculation of the total chloride ion concentration in a mixture is as follows: If the chloride ion concentrations in the sodium nitrate and potassium nitrate are 0.85 and 0.20 weight percent, respectively, then the chloride ion concentration in the mixture is (0.6 weight fraction NaNO₃)*(0.85 percent) + (0.4 weight fraction KNO₃)*(0.20 percent) = 0.59 percent.

For this example, the 0.6 maximum weight percent chloride ion concentration for the nitrate salt mixture is satisfied.

For all of the compounds which contain chlorine, the chloride ion concentrations in weight percent shall be calculated from the weight percent of the compound in the nitrate salt, as follows:

$$\left[\frac{\text{Atomic Weight of Chlorine}}{\text{Molecular Weight of Compound}} \right] (\text{Weight Percent of Compound})$$

For example, a sodium chloride (NaCl) concentration of 0.9 weight percent is converted to a chloride ion concentration of 0.55 weight percent by multiplying the sodium chloride concentration by 0.607, as follows:

⁹ Williams, D.F., *Assessment of Candidate Molten Salt Coolants for the NGNP/NHI Heat-Transfer Loop*, 2006, Oak Ridge National Laboratory

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$$\left[\frac{35.457 \frac{gm}{gm - mole} \text{ for Cl}}{\left(22.997 \frac{gm}{gm - mole} \text{ for Na} + 35.457 \frac{gm}{gm - mole} \text{ for Cl} \right)} \right] (0.9 \text{ Percent})$$

Similarly, potassium perchlorate (KClO₄) concentration in weight percent shall be converted to chloride ion concentration in weight percent by multiplying the potassium perchlorate concentration by 0.256, and sodium perchlorate (NaClO₄) concentration in weight percent shall be converted to chloride ion concentration in weight percent by multiplying the sodium perchlorate concentration by 0.290.

9.3.3 Allowable Contaminants

Maximum contamination from all sources, by weight, will be:

- Nitrite: < 0.02 percent
- Carbonate: < 0.10 percent
- Sulfate: < 0.01 percent
- Hydroxyl alkalinity: < 0.20 percent
- Perchlorate: < 0.085 percent
- Magnesium: < 0.035 percent.

The specification should also state that notification shall be provided for any contaminants not listed above which exceed a concentration of 0.04 percent by weight.

9.3.4 Supply Options

The salt may be supplied in one of two forms:

Option 1: The sodium nitrate is delivered in separate bags from the potassium nitrate. The project is responsible for mixing the two components in the required 60/40 mixture, melting the solid mixture, and loading in salt in the storage system

Option 2: The salt supplier mixes the components, melts the solid mixture, solidifies the liquid mixture in the form of a prill, and delivers the prills to the site. The project is responsible for melting the prills and loading the salt in the storage system.

Option 1 can offer a slight discount in the price of the salt. However, the material handling requirements and costs can be substantial. In addition, the project must have the equipment to treat a one-time emission of oxides of nitrogen. Specifically, all commercial grades of salt contain trace amounts of magnesium. Magnesium is soluble in nitrate salt, appearing as magnesium nitrate. Upon heating, the magnesium nitrate decomposes via the following irreversible reaction:

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The magnesium oxide is insoluble in the salt, and forms a solid precipitate. The NO_x has a very low solubility in the salt, and essentially all of the NO_x appears as a gas in the ullage space above the liquid level. A typical commercial project might require 30,000 metric tons of salt. The magnesium concentration in the technical grade of the salt is a nominal 0.035 percent by weight, which translates to about 10,000 kg (410 kg mole) of magnesium. The decomposition reaction produces 2 moles of NO_x per mole of magnesium, which results in the generation of about 38,000 kg of NO_x. Some form of NO_x capture and treatment system will be required to both meet environmental regulations and to ensure worker safety. The least complex approach is probably a water spray column, in which water reacts with NO_x to form nitric acid, N₂NO₃. The acid is neutralized with a base, and then shipped offsite for disposal.

Under Option 2, the process of melting and mixing the components to form prills will promote the magnesium decomposition reaction. As such, the one-time production and treatment of NO_x will be the responsibility of the salt supplier rather than the responsibility of the project. This division of responsibility will be reflected in the cost of the salt delivered to the project.

As part of the procurement process, the engineering contractor will need to conduct an economic analysis to determine which of the two supply options offers the lowest total cost to the project.

9.3.5 Deviations from a 60/40 Mixture

Under either Option 1 or Option 2, the final composition is likely to vary slightly from the desired 60/40 mixture. In principle, property tables and relationships could be developed for the exact salt mixture at the project. However, the properties are not strongly influenced by mixture fractions, which are often in the range of 58/42 to 62/48 for commercial projects. An example is the melting point, as shown in Figure 9-1.

Other properties, specifically the density, can be estimated by adjusting the 60/40 properties based on molar volumes, as described by Bradshaw¹⁰.

9.4 Nitrate Salt Handling and Melting Specification

Nitrate salt is hygroscopic, and absorbs moisture from the air during transit and when stockpiled for melting.

¹⁰ Bradshaw, R. W., (Sandia National Laboratories, Albuquerque, New Mexico), "Effect of Composition on the Density of Multi-Component Molten Nitrate Salts", Sandia Report SAND2009-8221, December 2009

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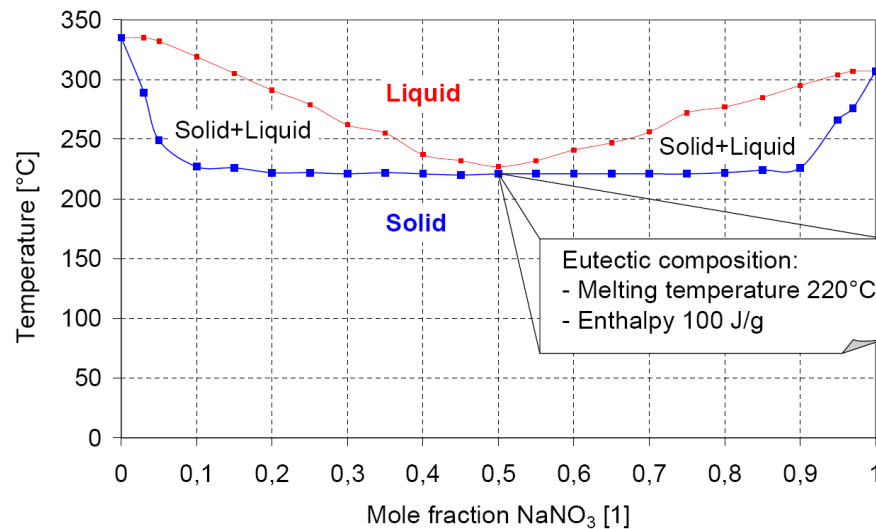


Figure 9-1 Melting Point of Binary Nitrate Salt Mixtures

In trough projects, the design temperature of the hot salt tank is on the order of 395 °C. During the salt melting process, the temperature of the liquid salt is raised to something less than the design temperature of the hot tank; perhaps, 350 °C.

The salt, as delivered, contains 0.01 to 0.05 percent magnesium nitrate, depending on the purity. At temperatures of 550 °C and above, the magnesium nitrate, in a period of several days, decomposes via the following reaction:



The magnesium oxide precipitates as a solid, and the NO₂ and the O₂ leave the salt as gases.

The magnesium nitrate concentration is only a fraction of a percent; however, each mole of magnesium nitrate produces two moles of NO₂. For a plant with a storage mass of 35,000 metric tons, the total NO₂ production will range from 2,000 to 11,000 kg, depending on the purity of the salt.

In a trough project, the salt, after melting, is loaded into the storage tanks. However, the temperature of the salt is perhaps in the range of 275 to 350 °C, depending on the capacity of the tank electric heaters. As such, the magnesium decomposition rate is much slower than would occur at a temperature of 550 °C; certainly on the time span of months. Further, the water that is adsorbed by the salt in transit and in storage does not leave the salt during, and immediately after, the melting process. Anecdotal evidence suggests that the time required to release the water from the salt is measured in weeks. As such, within the liquid inventory of the salt, NO₂ produced from magnesium decomposition comes into

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contact with H₂O released from the salt. The two combine to form nitric acid, H₂NO₃. The nitric acid is released from the salt in the form of a gas. Further, the gas can condense on any surface with temperatures below about 80 to 90 °C. Since nitric acid has a dew point higher than water vapor, any warm surface on which the vapor condenses will essentially be pure acid. Nitric acid samples collected from a commercial project during commissioning showed pH values in the range of 1 to 2.

With the exception of the heat transfer surfaces in the oil-to-salt heat exchanger, essentially everything in the storage system is fabricated from carbon steel. Nitric acid in contact with carbon steel results in high corrosion rates, with the highest rates measured in mm per month.

The specification for salt handling and melting must include the following provisions:

- To prevent damage to the carbon steel equipment which is exposed to the ullage gas in the storage tanks, the equipment must be maintained at a temperature of at least 250 °C for a period of several months after the salt is loaded into the storage tanks. A temperature of 250 °C prevents the condensation of both nitric acid and salt vapors.
- If the equipment cannot be maintained at a temperature of 250 °C, then the equipment must be fabricated from a material which is resistant to nitric acid corrosion, such as stainless steel.

9.5 General Salt System Equipment

9.5.1 Piping

Design Codes

The nitrate salt piping can be designed to the requirements of either ASME B31.1 - Power Piping or ASME B31.3 - Process Piping.

B31.1, compared with B31.3, is more conservative in terms of material specifications, stress intensity factors, stress design factors, and radiographic inspections. In the interests of providing reliable piping systems, at least two commercial central receiver projects have selected B31.1 as the design code for the salt piping in the receiver system. This includes the cold salt pump discharge piping, the riser, the inter-panel piping, the vent lines, the drain lines, and the downcomer.

In a parabolic trough plant using an organic heat transfer fluid, the salt piping is limited to the thermal storage system. The pipe lengths and quantities are relatively small, and the potential cost premium between selecting B31.1 or B31.3 is likely to be modest. However, the design temperatures and the rates of temperature change in the trough storage system are modest compared to central receiver projects. To some extent, these characteristics could influence the decision regarding the design code

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for the piping. At least one commercial parabolic trough project has selected B31.3 for the design of the salt piping in the storage system.

Materials

The cold salt and the hot salt piping is typically ASTM A106 Gr B or Gr C, with a nominal corrosion allowance of 1.6 mm (1/16 in.).

Connections, Sizes, and Schedules

End connections for salt piping will be butt welded joints, except in the following locations:

- Salt pump discharge nozzles. Flanged connections using ring-type joints are permitted.
- Salt tank nozzle connections to the gas space inside the roof.

The minimum system piping diameter shall be DN 100 (4 in.) to reduce the potential for salt freezing due to damaged insulation or due to reduced heat trace capacity.

For both cold salt and hot salt service, the minimum schedule for DN 100 (4 in.) lines is Schedule 40. The goal is to provide sufficient bending stiffness to limit sagging and to prevent sections in the line which may not fully drain.

For lines larger than DN 100, the minimum Schedule is 20. The goal is to provide enough stiffness at the ends of pipe such that normal handling of the pipe does not cause the ends to become oval, which would make lineup and fitment prior to welding problematic.

All pipe will be specified as seamless to provide as much resistance to fatigue damage as possible.

Flanges

Flanges in salt service may be of the following types: ring type joint; or hub type. No other types of flanges, such as raised face flanges, will be used due to the potential for relaxation of the bolted connection and leakage past the gasket due to the excellent wetting characteristics of the salt. Ring type joint gaskets and hub rings will be ASTM A182 Gr F21.

Stud bolts used at temperatures below 400 °C will be specified as ASTM A193 Gr 7.

Fittings

Butt weld fittings will be used in salt service in all pipe sizes.

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Compression fittings, flat face flange, raised face flange, and socket welded connections are not suitable for salt service.

Pipe Supports, Anchors, and Guides

Pipe supports, anchors, and guides are subjected to daily thermal expansion and contraction cycles.

At the Solar Two project, insulated pipe supports, using an external metal clamp with a calcium silicate insert between the pipe and the clamp, had an effective life of perhaps 1 year before the insert worked loose from the clamp. Alternate designs, using a radial fin to positively locate the insert within the clamp, also failed due to fracturing of the insert.

The recommended support, anchor, and guide design uses a metal web, welded directly to the pipe. The web, in turn, is welded to a base plate. In the power block area, the base plate can be welded to structural steel elements, either above or below the salt line. A portion of the web is removed to reduce conduction heat transfer through the web. The basic design is illustrated in Figure 9-2.

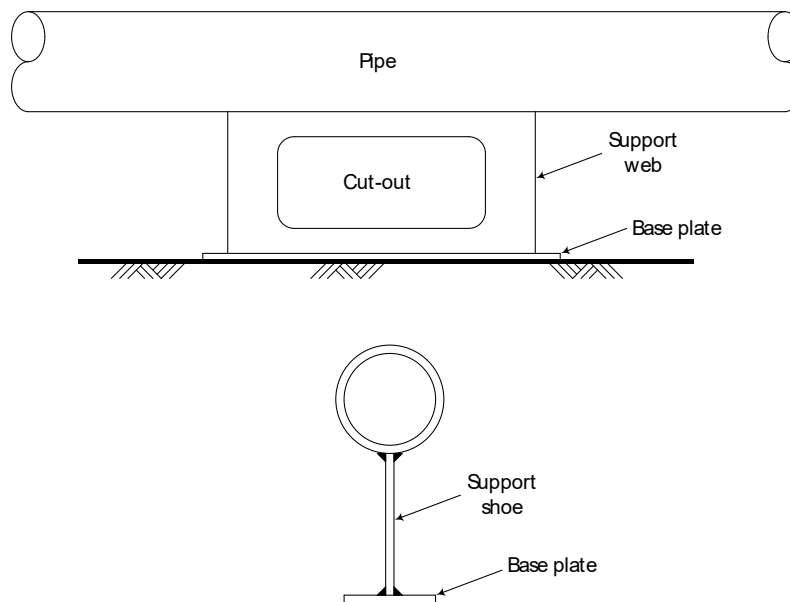


Figure 9-2 Pipe Support, Anchor, and Guide

The heat losses at the support will be greater than the heat losses in the adjacent pipe. As such, additional heat trace may need to be provided at the supports to ensure adequate pipe temperatures.

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For pipe supports handling strictly vertical loads, without the possibility of a horizontal component, conventional spring hangers are recommended.

9.5.2 Control and Isolation Valves

In general, the use of control and isolation valves in salt service should be held to the minimum.

Process Design

In general, valves in salt service are a major contributor to plant forced outages. One of the principal goals of the process design is to develop an equipment arrangement that uses as few salt valves as possible.

One example is the piping arrangement for the external electric salt heaters for the salt storage tanks. At one commercial project, two 100-percent salt heaters can receive salt from any of the 3 hot salt pumps in the hot tank, or salt from either of the 2 cold salt attemperation pumps in the cold tank. However, a total of 9 salt valves are required to either direct the flow as needed or to drain the equipment. In contrast, an alternate design would locate a separate electric heater in the minimum flow recirculation line for each of the salt pumps. Each heater would be located downstream of the minimum flow recirculation valve, and each heater would be located at an elevation above the salt tanks. The minimum flow valve would keep the electric heater flooded during operation, and the piping would be arranged such that the piping and the heater drain automatically when the pump is not in operation. Granted, a typical project might require 2 (or 3) hot salt pumps and 2 attemperation pumps, and as such, would require 4 (or 5) electric salt heaters. Further, the 4 (or 5) heaters could be more expensive than two 100-percent capacity heaters. However, the improvement in plant availability due to the elimination of 9 salt valves will more than offset, by far, the marginal cost of the smaller electric heaters. Plus the use of salt heaters on each of the pumps provides essentially the same redundancy as the two 100-percent heaters in the example commercial project.

Extended Bonnets

Valves for service at temperatures above 350 °C will be specified with extended bonnets to ensure that the packing temperature does not exceed 200 °C in the absence of supplemental heat from the bonnet heat trace system. A heat trace zone, dedicated to the bonnet region, maintains the packing temperature in the desired range of 275 °C to 300 °C.

Valve Types

A salt corrosion layer develops on the plugs and the seats of all valves. For gate and ball valves, the two sealing surfaces slide across one another. When the corrosion layers develop, particularly with the valves in the closed position, the corrosion layers can lead to binding and erratic valve motion. As such,

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gate and ball valves are not suitable for salt service. For globe and offset butterfly valves, the plugs move perpendicular to the seats, and the development of the corrosion layers are less likely to interfere with the motion of the valve.

Globe and offset butterfly valve seats and disks will have Stellite® faces. Nonetheless, hardened surfaces will corrode in salt service. Over time, the valve will fail to provide leak-tight service, and the equipment downstream of the valve must be able to operate safely with a small, but continuous, flow of salt. For example, vent valves for heat exchangers should be located at the top of the vent line to allow leakage past the valve to drain to a storage tank.

Stem Seals

For globe valves, the preferred stem seal, by a considerable margin, is a metal bellows. The bellows, although expensive, provide a hermetic stem seal. Hermetic seals are important, as leakage past the valve stem will expose the heat trace cables on the valve body and the adjacent piping to salt. The cables operate at temperatures which are high enough ($> 650\text{ }^{\circ}\text{C}$) to decompose the salt. Several of the decomposition products are various oxides, which aggressively corrode the outer metal covering on the heat trace cables. Corrosion lifetimes are often measured in weeks. Once the covering corrodes, the internal heating cables are exposed to moisture, and can fail in a matter of days. Failed cables lead to immovable valves and frozen salt piping, which can necessitate forced outages, lasting hours to days. A project economic analysis will often show that the marginal improvement in plant availability and revenue, due to a properly functioning heat trace system, justifies the marginal expense for bellows stem seals.

The bellows should have a minimum fatigue life of 10,000 cycles, and should be replaceable with the valve in place. The bellows region will require both heat tracing and insulation. If available, a pre-engineered bellows enclosure should be procured from the valve supplier.

The primary bellows stem seal should be supplemented with a backup conventional stem packing, as described below.

For offset butterfly valves, bellows stem seals are not an option, and a conventional stem packing must be used. However, carbon and graphite materials used in standard bonnet packings are only marginally acceptable in salt service. The salt is an oxidizing material, and reacts with the carbon and graphite to form CO_2 . To date, the valve stem packing which offers the longest, but nonetheless marginal, service life consists of alternating layers of the following:

- Wire-reinforced graphite braid packing over a fiberglass core: Style 1200-PBI from Garlock Engineering, or Style 387I from John Crane, Inc.
- Fiberglass-filled Teflon® washers.

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The allowable temperature range for the materials is 260 °C to 315 °C. The lower temperature limit is to prevent salt from freezing; the upper limit is to prevent the Teflon from decomposing. Teflon, as it decomposes, produces fluorine gas, which aggressively attacks the valve stem material. Note that the stem packing relies on graphite, and will require periodic replacement, regardless of the valve temperature.

Bonnet Gaskets

For split-body globe valves, the bonnet gaskets should be either metallic ring type (carbon core with a soldered silver sheath), or welded spiral wound.

9.5.3 Check Valves

A conventional check valve can be provided in the discharge line of a salt pump for the following purposes:

- Should a pump trip, the check valves prevents a reverse flow through the pump, and protects against an overspeed condition in reverse rotation
- During steam generator startup, hot salt is blended with cold salt by operating the hot salt pump in parallel with the cold salt attemperation pump. At the start of the blending process, the pressure at the discharge of the hot salt pump is slightly lower than the pressure at the discharge of the attemperation pump. The speed of the hot salt pump is then increased until the discharge pressure of the hot salt pump is a few Pa higher than the discharge pressure of the attemperation pump. A check valve at the discharge of the hot salt pump 1) prevents a reverse flow though the hot salt pump when the discharge pressure of the hot salt pump is less than the discharge pressure of the attemperation pump, and 2) helps to accurately control the relative flow rates of hot salt and cold salt when the hot salt pressure is only slightly greater than the cold salt pressure.

As with all salt valves, corrosion layers will develop on the flap, the seat, and the pivot of the check valve. If the valve remains in the closed position for an extended period of time (weeks to months), then the corrosion layers on the internal components can merge, which would leave the check valve immovable.

As with globe and triple offset butterfly valves, the flap in the check valve moves perpendicular to the seat, which reduces the potential for the corrosion layers on the flap and the seat to merge. Nonetheless, the only effective means of preventing problems with corrosion is to periodically open and close the valve.

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9.6 *Oil to Salt Heat Exchanger*

9.6.1 Equipment Design

The oil-to-salt heat exchanger necessarily undergoes a daily startup and shut down cycle. Further, the higher the allowable transient heating and cooling rate ($^{\circ}\text{C}/\text{min}$) and the higher the allowable thermal shock value, ($^{\circ}\text{C}$), the more thermal energy can be stored each day for electric power production.

The heat exchanger concept with the highest tolerance for cyclic thermal stresses is perhaps an all-welded design. Examples include the header/coil approach, and the internal bore welded design, each of which are discussed in Section 6.6.3, Fabrication Techniques, of Volume 3 - Narrative.

9.6.2 Process Design

To ensure that the heat exchanger reaches its projected fatigue life, the process design must ensure that the transient heating and cooling rates, and the thermal shocks, are maintained within the limits set by the vendor. To this end, the process design should include the following set of features:

- During Hold periods, cold salt is supplied to the cold end of the heat exchanger. The flow of Therminol through the heat exchanger is suspended to reduce the potential for cold Therminol from the collector field freezing salt in the heat exchanger. As such, no heat transfer occurs, and the salt leaving the hot end of the heat exchanger is returned to the cold tank through a recirculation line.
- Included in the design is a Therminol mixing at the hot end of the heat exchanger. At the beginning of a charge cycle, cold Therminol from the discharge of the Therminol pumps is mixed with hot Therminol from the collector field at the mixing station. The blending process controls the rate of temperature increase in the heat exchanger during the transition from Hold to Charge, and reduces the magnitude of any thermal shocks.
- Also included in the design is a salt mixing station at the hot end of the heat exchanger. At the end of a Discharge period, cold salt from the cold salt pump is mixed with hot salt from the hot salt pump at the mixing station. The blending process controls the rate of temperature decrease in the heat exchanger during the transition from Discharge to Hold.
- During the Hold period, the flow rate of salt is set to a value which is equal to, or greater than, the minimum flow rate specified by the vendor. During normal operation, the heat exchanger is always operated at a combination of salt flow rate and Therminol flow rate that satisfies the vendor flow rate requirement. As a consequence, the minimum flow rate criterion on the salt

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side is automatically satisfied both during the Hold period, and during the transition from Discharge to Hold.

- Included in the design is a bypass line on the Therminol side of the heat exchanger. During heat exchanger startup, the bypass line would operate as follows:
 - For collector field outlet temperatures less than the cold salt temperature, the isolation valve in the bypass line is open, and the flow of Therminol bypasses the heat exchanger
 - When the collector field outlet temperature reaches the cold salt temperature, the isolation valve is closed, and a flow of Therminol is established in the heat exchanger. The flow rate is large enough to satisfy the minimum flow rate specified by the vendor. Since the temperature of the salt is equal to the temperature of the Therminol, no heat transfer occurs. As a result, during the period in which the flow rate of Therminol increases from an initial value of zero to a final value equal to the minimum flow rate, the heat exchanger is operating under isothermal conditions. As such, the potential for developing damaging transient thermal stresses is very low.

9.7 Thermal Storage Tank Weld Examinations

The floors in the thermal storage tanks are constructed from an array of rectangular plates. The nominal plate dimensions are 2.5 m by 12.5 m. Representative plate thicknesses are in the range of 7 to 10 mm, and two weld passes are typically required.

The allowable tolerances on the dimensions of the plates are presented in ASTM A568/A568M-19a, Standard Specification for Steel, Sheet, Carbon, Structural. Since all of the plates are not exactly the same size and shape, the dimensions of the gaps between adjacent plates will vary. To provide consistent inter-plate welds, backing strips are welded to the backs of alternating plates prior to arranging the plates on the foundation.

The storage tanks are subjected to daily thermal cycles, which results in cyclic loads on the floor generated by friction and radial temperature gradients. For the tanks in parabolic trough projects, it's not clear if a low cycle fatigue analysis is required. However, if such an analysis is justified, then the fatigue properties of the weld are subject to a weld fatigue strength reduction factor. The topic is discussed in Section 5.13.2, Fatigue Strength Reduction Factors, of Volume 3 - Narrative.

To improve the calculated fatigue life of the floor, the reduction factors should be as low as practical. The smallest reduction factors (1.0) are achieved by 1) machining the weld surface, 2) confirming the weld quality through the use of ultrasonic tests (radiographic examinations are not practical on the floor), and 3) examining the weld surface through a combination of visual, magnetic particle, or dye penetrant tests.

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The largest reduction factors (4.0) occur on the back sides of welds that cannot be observed or are otherwise non-definable. This characteristic applies to the bottom of the root pass in contact with the top of the backing strip.

The Owner must coordinate with the tank fabricator to reach an agreement on the welding procedures and the weld examinations to ensure a consistent treatment among the cost of the tank floor, the calculated fatigue life, and the expected fatigue life in commercial service.

9.8 *Heat Tracing*

9.8.1 System Description

The electric heat trace system thermally conditions equipment prior to initiating salt flow, and provides freeze protection on all salt equipment.

Although not specifically discussed in the Design Basis Document, electric heat trace systems may also be used for the overnight temperature control of large steam valves and the steam turbine casing.

The heat trace system is integrated system with the master control system as part of the distributed control system. The heat trace can be directly linked with other process control functions such that thermal conditioning is fully automated.

9.8.2 Scope of Supply

The electric heat trace equipment will be purchased as a system that encompasses the zone definitions, heat transfer calculations, cable arrangements, sensor locations, cable fabrication, installation, and acceptance testing. Zone definitions, and locations of the temperature sensors, will be developed in cooperation with the project engineering contractor. The scope of supply will encompass all materials and hardware, including the heat trace cables, splice kits, power supply junction boxes, temperature sensors, sensor junction boxes, and all installation hardware.

Fabrication and assembly of each heat trace cable will be performed by the supplier in the field. This is to ensure the cables are cut to match the as-built configuration of the equipment. Normally, fabrication of the cables at the vendor's shop is preferred, as the quality of the hot-to-cold junctions can be more closely controlled. However, the project schedule does not allow the preparation of factory cables after the salt equipment has been installed. The vendor will be required to supply cables, manufactured in the field, to have the same quality and reliability as cables manufactured at the vendor's shop.

The heat trace system design and supply should include the solid state circuit contactors. However, the electric power distribution, installation of the solid state contractors, and connections of the temperature sensors to the instrument junction boxes should be provided by the site electrical contractor.

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9.8.3 Electric Heat Tracing System Design Basis

Zone Definitions

The boundaries of the heat trace zones will be defined by the equipment geometries, the system operating modes, and liquid locations within the equipment. Also, zone temperature set points will vary by state and transition. As such, the definition of the zones will need to be determined by the project engineering contractor, and not by the heat trace vendor. The heat trace vendor will collaborate with the engineering contractor to set limits, as needed, on practical zone dimensions. As an example, transitions between operating modes can result in certain pipe segments changing from flowing salt to stagnant salt. The zone boundaries, and the location of the temperature sensors, must be selected such that the full lengths of the stagnant zones, which change with the mode, are always protected. In addition, the temperature sensors must monitor the temperatures only in the stagnant zones, not in the flowing zones.

In general, zone lengths for piping should be as long as possible. Plant availability is improved by 1) reducing in the number of cables, contactors, controllers, and communication lines; 2) reducing the number of alarms monitored by the operators; and 3) reducing the number of potential operating modes which are not used correctly. The division of a piping circuit into multiple zones tends to decrease the plant availability, as the loss of one zone is equivalent to the loss of the complete circuit. The potential availability penalty associated with lengthy outages to remove and replaced failed cables in long heat trace zones is offset by increasing the number of installed spare cables.

Any significant change in the mass per unit length in a zone, such a valve in a pipe, requires a separate zone.

Any significant change in heat loss per unit length in a zone, such as a valve bonnet, requires a separate zone.

Any pipe section or instrument location, in which a zone of stagnant salt is intentionally formed for temperature control, requires a separate zone. Examples are illustrated in Figure 9-7 Pressure Transmitter in a Vertical Process Line, and in Figure 9-8 Pressure Transmitter in a Horizontal Process Line.

Thermal losses at pipe and equipment supports will be much higher than in the adjacent pipe or equipment. The geometry of the supports will need to be analyzed by the engineering contractor to determine if separate zones are required at the supports.

If different operating modes subject a pipe or a vessel to different static fluid heights, then the boundary between the zones must end above the liquid level. For example, recirculation from a salt pump at a demonstration solar project produced a static height of about 9 m in a riser. If the heat trace zone in the lower section of the riser had ended below an elevation of 9 m, then the stagnant salt above the top of

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the zone would have frozen in a matter of a few hours. Further, if the zone spans a region which is not exposed to salt at the top, but is flooded below, then a heat transfer analysis must be conducted to confirm that the temperature in the upper portion of the zone does not exceed safe levels.

Zone Thermal Capacities

The thermal capacity in each zone is determined by the following factors:

- An economic analysis, comparing the capital cost of the heat trace capacity with the system startup time. One example is a salt line, which is normally drained at the end of an operating day. Prior to startup the following day, the temperature of the piping must be increased to the salt fill temperature. If the line has a high heat trace capacity, then the preheat time will be short, the preheat energy will be low, but the capital cost will be high. (The reverse also applies.) As such, there is an optimum heat trace capacity that minimizes the sum of (the equivalent capital cost of the preheat energy) and (the capital cost of the heat tracing).
- An Owner's assessment of the degree to which the heat trace capacity must accommodate an expected degradation in the cable heating capacity or an increase in the heat losses through the equipment insulation
- An Owner's assessment of the reliability of the heat trace cables, and the requirement for spare installed cables.

The nominal set of design parameters include the following:

- Empty salt pipes, empty valve bodies, and valve bonnets can be preheated from ambient temperature to 275 °C in 12 hours
- Empty salt heat exchangers can be preheated from ambient temperature to 275 °C in 36 hours. Preheating energy will be provided by a combination of electric heat trace cables on the shells and the channels, and an electric recirculation water heater.

The calculations will be performed using the design minimum ambient temperature and a concurrent wind speed of 14 m/sec.

Allowances of 5 percent will be added to the zone capacities for simple geometries (pipe sections) and 30 percent for complex geometries (valve bodies, valve bonnets, pipe supports, guides, and anchors).

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Control and Set Point Temperatures

The minimum system set point temperatures, for both the cold and the hot salt equipment, is a nominal 275 °C.

The controller temperature dead band is selected to limit contactor cycling; i.e., -10 °C to +10 °C.

Component Redundancy

The required number of heat trace cables to satisfy the preheat requirements will be calculated as noted above. The number of installed cables will be equal to the number of required cables, plus the following allowances for spare cables:

- 100 percent for cables on the piping from the heat exchangers. This segment of piping operates under daily thermal cycles, and the failure rates of the heat trace cables are expected to be 'moderate to high'
- 50 percent for cables on 1) the salt piping to the heat exchangers, 2) the nitrogen vent lines to the salt tanks, and 3) the vent and drain lines for the heat exchangers. This equipment operates under essentially steady state temperatures, and the failure rates of the heat trace cables are expected to be 'low'.

The spare cables will accommodate failures of the original cables without the need to remove the insulation. The spare cables will be labeled as such, but not connected to the electric power supply.

To the extent possible, large valves will have spare heat trace cables installed. For small valves, and for the bonnets on both large and small valves, the installation of spare cables is not practical. As such, spare cables, fabricated to the dimensions of the original cables, will be assembled and kept in the warehouse for future maintenance.

For each temperature sensor installed on a pipe, valve body, valve bonnet, or instrument, an installed spare shall be provided. Dual element thermocouples satisfy the redundancy requirement.

Component Requirements

The recommended cable type is mineral insulated, with dual conductor heating elements, a welded sheath, and magnesium oxide insulation. A welded sheath is preferable to a brazed joint, as the latter can be subject to cracking due to differential thermal expansion between different metals. The cables can be supplied with either an Inconel or a 300-series stainless steel sheath. The latter is less expensive than the former, and offers a similar resistance to corrosion should the cables be exposed to salt following a leak.

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Two cable diameters are recommended: $\frac{5}{16}$ -inch (nominal 8 mm) for 600 V service for long piping zones; and $\frac{3}{16}$ -inch (nominal 5 mm) for 120 V service on valves and line devices. The conductor resistance will vary, depending upon the length of the cable. Cable power density should be limited to 160 W/m to reduce the potential for corrosion of the sheath in the event of a salt leak.

The cables are secured to the pipe using stainless steel tie wires or straps. A typical installation, showing the end of a zone, is illustrated in Figure 9-3. A stainless steel foil layer is installed over the cables, followed by a flexible blanket insulation approximately 25 mm thick. The foil prevents the flexible blanket from potentially lodging underneath the cables, which would cause the cables to overheat. A rigid block insulation is installed over the blanket. The blanket conforms to the uneven surfaces produced by the cables, the ties, and the corrugated tubes at the end of the zone. This, in turn, reduces the potential for convection heat flows under the rigid insulation.

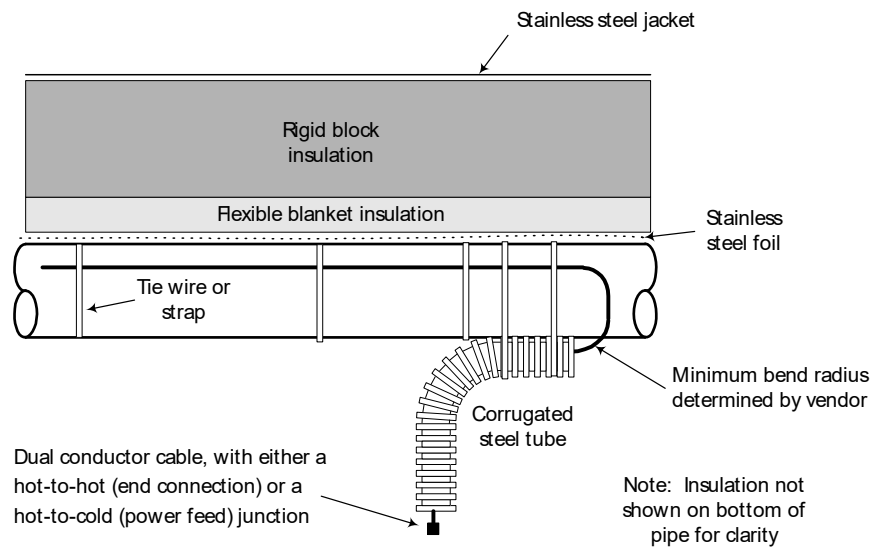


Figure 9-3 Heat Trace Arrangement on Pipe at End of Zone

For zone temperature measurement, either thermocouples or resistance temperature detectors are acceptable. Whichever type is selected, they will be standard across the entire facility.

Valve Zones

The heat trace for a valve will be independent of the heat trace for the adjacent pipe, for three reasons:

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- The unit mass, in kg/m, of a valve body is much higher than the unit mass of the adjacent pipe. As such, if a time is specified to preheat the piping system to 275 °C, the heat input per meter for a valve body must be much higher than the heat input per meter for the pipe
- The type of insulation on a valve body is often different from the type of insulation on the adjacent pipe. As such, under steady state conditions, the heat loss per meter of valve body will be different than the heat loss per meter of the pipe
- The unit heat losses, in W/m², from the valve bonnet are much higher than the unit heat losses from the valve body. Also, the temperature of the bonnet must be maintained within a narrow range of 275 °C to 300 °C if a stem packing is used. The lower value ensures the salt will not freeze in the bonnet. The upper value ensures that Teflon or other organic materials in the stem packing, if used, do not decompose.

A typical cable installation on a valve is illustrated in Figure 9-4. The power rating for cable on the valve body is selected based on the required system preheat time. If the valve body is large enough, redundant cables can be installed.

The power rating for the cable on the valve bonnet is selected to maintain metal temperatures in the range noted above, under the worst combination of wind speed and ambient temperature. In general, limited space is available on valve bonnets, and only one cable can be provided.

Since the heat trace circuit for the valve body operates independently of the heat trace circuit for the valve bonnet, separate thermocouples are required for the two zones.

Check valves do not have extended bonnets. If the insulation thickness on the check valve is the same as the insulation thickness on the adjacent pipe, then the unit heat loss for the valve should be essentially the same as the unit heat loss for the pipe. Consequently, treating the pipe and a check valve as a contiguous zone is an acceptable design practice.

Temperature Sensor Installation

Sensor elements are to be welded or banded to the pipe or equipment, and then covered with a stainless steel tent to reduce convection heat losses from the sensor.

The number of sensors in a zone will depend on the geometry of the zone, as follows:

- For zones of limited dimensions, such as instrument stubs, one dual-element thermocouple is adequate

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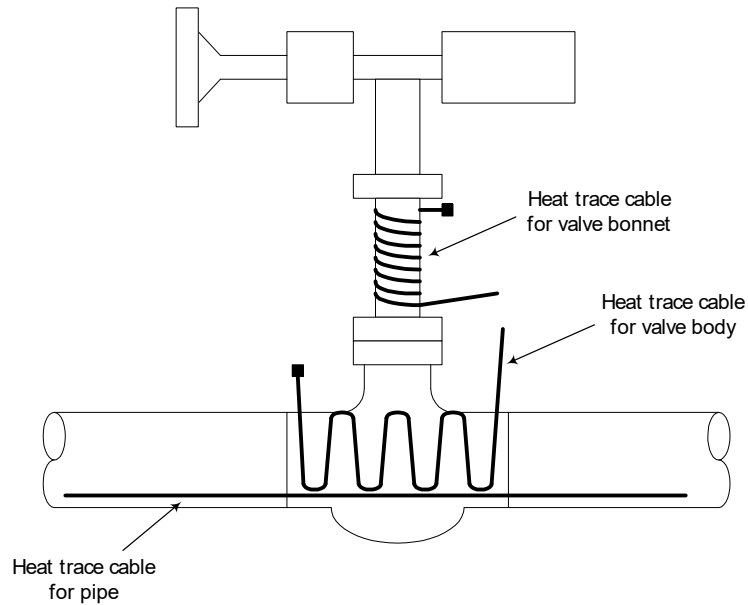


Figure 9-4 Typical Heat Trace Cable Arrangement on Valves

- For long pipes, operating in cyclic service, as many as 6 to 8 dual-element thermocouples may be needed to understand the temperature distribution throughout the zone. The heat trace circuits would be controlled based on a low select feature
- For pipes operating with salt flowing in some modes, but switching to static salt in other modes, a minimum of 2, and as many as 6, dual-element thermocouples may be needed to maintain set point temperatures during all of the modes.

9.8.4 Equipment Heat Tracing

Heat tracing requirements unique to major equipment items are described below.

Pressure Safety Valves

All pressure relief valves in salt service will have electric heat tracing. However, the springs are subject to relaxation, even those fabricated from high temperature materials such as Inconel. As such, the heat trace cable locations and the valve insulation must be designed to 1) maintain a valve body temperature of at least 275 °C, and 2) maintain the spring temperature as low as practical. In general, separate zones will be provided for the valve body, the inlet piping, and the outlet piping.

The discharge line from the valve must have heat tracing over the entire length of the pipe.

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A weep hole at the low point in the valve discharge line should be provided to indicate that 1) the valve is leaking, and 2) the valve should be removed for lapping of the seat and the plug during the next scheduled plant outage. Leakage from the discharge line can be collected in metal containers, allowed to freeze, and returned to the storage tanks through the manway.

Vortex Shedding Flow Meters

Vortex shedding flow meters reside in fittings, which are generally the same diameter as the pipe. As such, the meter and the adjacent pipe can be treated as a contiguous zone. Nonetheless, the meter has a small boss which houses the vibration sensor, and the boss extends through the pipe insulation. The boss is not insulated due to temperature limits on the piezoelectric sensor. Therefore, the boss acts like a fin and cools the top of the fitting.

At the one solar project, a loop in the shape of an “S” with a length of 30 cm, was added to each cable on the 150 mm flow meters to compensate for the convection losses from the boss. With a unit rating of 130 Watts per meter, and 2 active cables, the loops increased the heat input to the meter by 80 Watts over that which would have been provided by the cables on the adjoining pipe.

Tank Vents

Each of the storage tanks has a vent in the roof, which is open to the atmosphere. To prevent the condensation of salt vapors in the vent, the entire length of the line is heat traced and insulated.

Further, the heat losses from the vent are relatively high because 1) the end of the vent is exposed to the atmosphere, and 2) ambient air enters the vent whenever the level in the tank is decreasing. As such, the heat trace circuit on the vent line is treated as a separate zone.

Tank Vacuum / Pressure Relief Valves

The vacuum / pressure relief valve are attached to the roof tank, and the valves are insulated. However, the combination of convection and radiation heat transfer from the liquid inventory to the valves is not always sufficient to maintain the temperature of the valves above the salt freezing point. This is particularly the case if the insulation on the valve has been damaged or degraded. As such, the valves are heat traced, and each valve is treated as a separate zone.

In general, inventory levels in the storage tanks decrease only slowly. As such, the mass flow rate of air entering the tank, either through the vent or through the vacuum relief valve, is a relatively low value. Although the air entering the tank is at ambient temperature, the cooling effect on the vent and the relief valve is modest, and the likelihood that salt vapors from the ullage space will freeze on, and eventually plug, the vent or the relief valve is vanishingly small.

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Tank Level Gauges

Air supply lines to bubbler level gauges will have electric heat tracing. The purpose is to prevent any residual moisture, which might be present in the instrument air, from freezing and plugging the lines on cold days. However, the heat tracing equipment on the air lines is different from the heat tracing equipment on the salt lines. The air lines use self-regulating heat trace cables, in which the electric current varies inversely, and passively, with the cable temperature. Representative maintenance temperatures range from 65 to 75 °C.

Capillary Lines for Diaphragm Pressure Transmitters

The capillary fluid is the eutectic NaK mixture, which has a freezing point is about -12 °C. As such, no electric heat trace is required for the capillary line.

Salt Pumps

The salt pumps are supported on a platform above the storage tanks, and draw suction from near the bottom of the tanks. An elevation view of the general arrangement is shown in Figure 9-5. The gap between the top of the tank and the bottom of the support structure is spanned with a metal bellows, as shown by the arrow in Figure 9-6. Electric heat trace will be installed on the bellows.

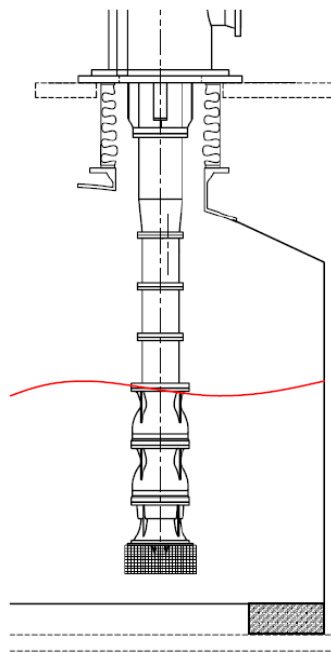


Figure 9-5 Elevation View of Salt Pump

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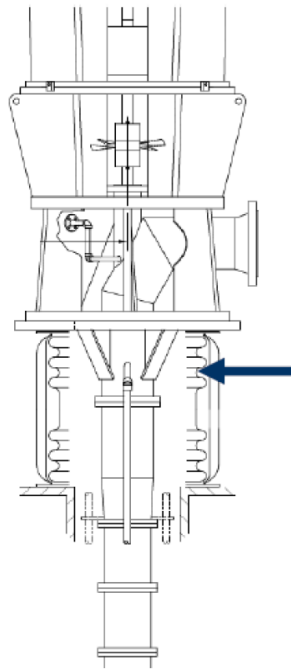


Figure 9-6 Detail of Salt Pump Discharge Head and Bellows

Depending on the pump geometry, it may also be possible to install heat tracing in the region above the bellows and adjacent to the discharge flange. If not, this portion of the pump must be thoroughly insulated to prevent salt from freezing on the shaft, and, depending on the location of the bearings, lock the rotor to the housing.

Essentially all salt pumps use a stuffing box near the top of the pump shaft. The stuffing box reduces the vertical migration of salt up the shaft due to surface tension effects. The stuffing box is provided with either service or instrument air to 1) develop a slight positive pressure near the top of the pump, and 2) cool the packing in the box. The air lines to the box must have electric heat tracing to prevent salt vapors from condensing in the lines. (When the pump is idle, the air flow to the stuffing box is stopped to limit cooling at the top of the shaft.)

9.9 Salt Pumps

The pumps are a vertical turbine design, with the bearings lubricated by the salt. The pumps are located on structures above the salt tanks, and draw suction from a point near the bottom of the tank.

Commercial tank designs have volumes on the order of 17,000 m³. The mass of steel in the tank, and to a first order the price, reaches a minimum value with a height-to-diameter ratios of 1. As such, the optimum tank would have a height and a diameter of 28 m. However, API 650-style tanks are supported

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by the parent soil beneath the foundation base mat. A representative allowable soil bearing load at many project locations is 240 kPa (5,000 lb_f/ft²), which effectively limits the allowable salt inventory depth to about 12 m. As such, overall pump shaft lengths of 14 to 16 m are expected.

Representative fabrication materials, for both the cold salt and the hot salt pumps, include the following:

- Column - Carbon steel
- Shaft - Alloy steel
- Bowls - Carbon steel
- Impellers - Stainless steel
- Shaft bearings - Cast iron.

An insulated bellows, illustrated in Figure 9-5, 1) bridges the space between the top of the pump and the bottom of the support structure, and 2) allows for differential movement between the tank and the support structure.

The pump discharge connection may be either a ring type joint or a hub type flange. The ring gasket or hub ring shall be Type 321H or 347H stainless steel.

The pumps use variable speed motor drives. The pump will be capable of operating from minimum speed through rated speed with no restrictions on speed.

Installed Redundancy

In the thermal storage system, one 100-percent pump is provided for each process requirement. No installed redundancy is provided because 1) the reliability of salt pumps is excellent, and 2) the isolation and control valves associated with redundant pumps in parallel often exhibit a mediocre-to-poor reliability.

To quantify the effect, each pump requires 3 valves: a discharge isolation; a discharge control; and a minimum flow recirculation. As discussed in Section 5.2.3, the mean time between failures for salt valves is on the order of 4,000 hours, with a mean time to repair of 12 hours. Each valve must be in service for the pump to operate, which translates to an equipment-in-series availability of 0.991. In contrast, a salt pump has a mean time between failure of 10,000 hours, and a mean time to repair of 48 hours. This translates to an availability of 0.999.

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9.10 Salt Instruments

9.10.1 Pressure Instruments

The most common pressure instrument is a diaphragm unit in combination with a remote transducer. A capillary tube, filled with NaK, connects the diaphragm with the transducer. NaK is the preferred capillary fluid. It has a very low vapor pressure, which reduces the potential for changes in the diaphragm temperature to influence the pressure readings. The fluid also has a low freezing point (-12°C), which eliminates the need to heat trace the capillary tube. Nonetheless, temperature compensation is still required for accurate readings.

The diaphragm should be located in a position, in combination with a dedicated heat trace zone and thermal insulation, which maintains a constant temperature of 275°C during all operating modes.

An example of an installation in a vertical process line is shown in Figure 9-7. The dimensions of the external loop are selected such that a diaphragm temperature of 275°C is maintained, even if the temperatures in the process line reaches 385°C . During normal operation, the isolation valve is closed. If it is necessary to drain the system, the valve is opened.

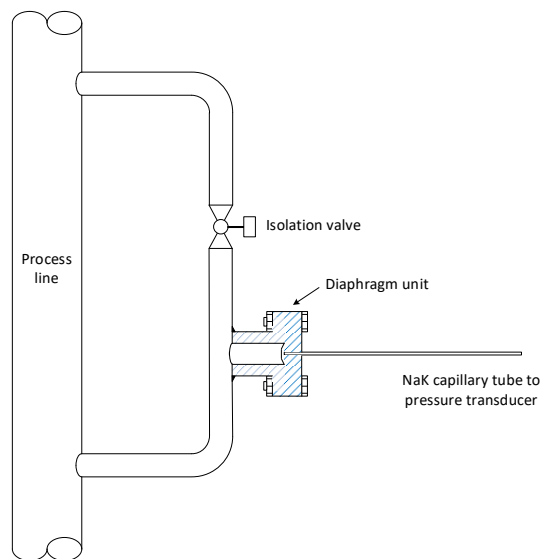


Figure 9-7 Pressure Transmitter in a Vertical Process Line

An example of an installation in a horizontal process line is shown in Figure 9-8. The dimensions of the standoff line are selected such that a diaphragm temperature of 275°C is maintained, even if the temperatures in the process line reaches 395°C . During normal operation, the isolation valve is open.

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If it is necessary to perform maintenance on the diaphragm unit, the process line is drained and the isolation valve is closed.

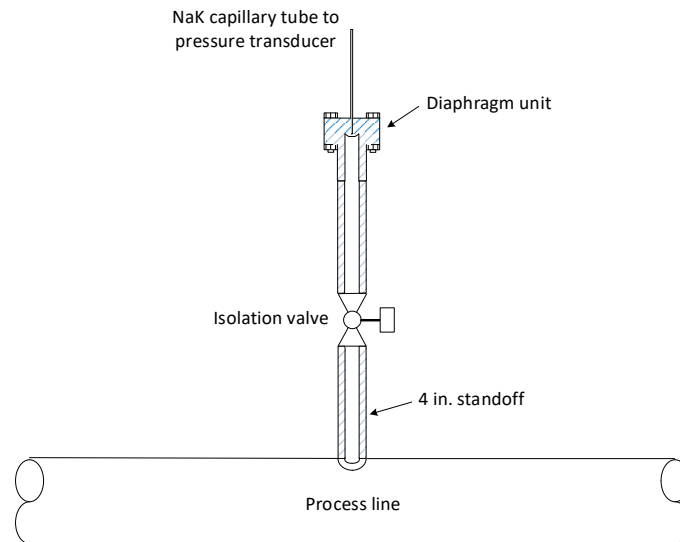


Figure 9-8 Pressure Transmitter in a Horizontal Process Line

In both the vertical and the horizontal installations, compressed air, rather than salt, is likely to be in contact with the diaphragm. Rapid changes in the pressure in the salt are likely to be damped by the trapped air. However, for process control, the installation geometries have proven satisfactory.

9.10.2 Flow Instruments

Candidate Designs

Vortex shedding flow meters have shown good reliability and availability in salt service. Meters are available in sizes up to 400 mm (16 in.), with a maximum process temperature of 450 °C.

Ultrasonic flow meters are suitable for use at all salt temperatures. For pipe sizes of 75 mm and smaller, the refraction angles may need to be increased to achieve the desired accuracy.

In principle, the pressure transmitters described above in Section 9.10.1 can be used in combination with a differential pressure transducer to measure flow based on differential pressure. The differential pressure can be developed using conventional flow devices, including venturis, flow nozzles, and orifice plates. A potential design of a flow venturi in combination with diaphragm pressure transmitters is shown in Figure 9-9.

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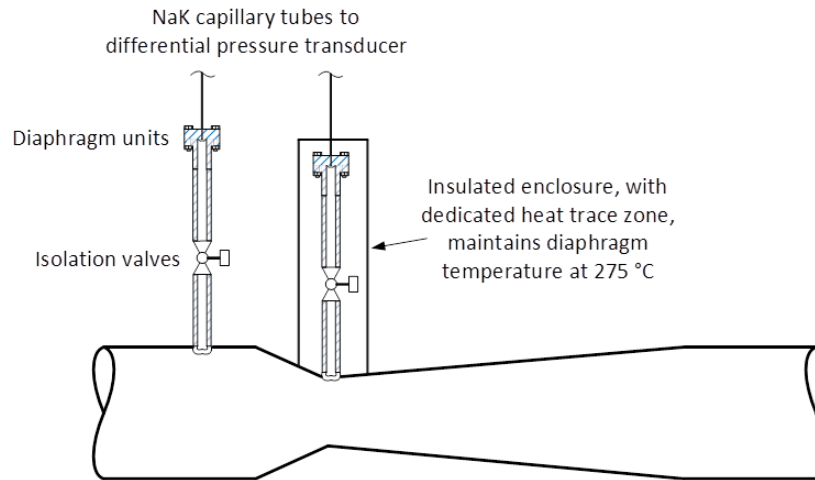


Figure 9-9 Flow Venturi with Diaphragm Pressure Transmitters

Venturis, flow nozzles, and orifice plates are available from a number of commercial vendors, and can operate over the full range of temperatures and flow rates expected in a commercial project. However, it is not known if a vendor has combined a conventional flow device with diaphragm units and NaK capillary tubes to develop a differential pressure flow device for use in salt service. In principle, the reliability and the accuracy of such a device should be suitable for commercial use, and could offer an alternate to ultrasonic flow meters in high temperature ($> 450\text{ }^{\circ}\text{C}$) service.

Accuracy and Reliability

In general, the reliability and the accuracy of flow instruments, particularly ultrasonic units, tends to be in the range of 'fair' to 'good'. For process control, in which accurate flow meters readings are important for equipment protection, redundant instruments can be placed in series, and a 1-out-of-2 voting logic employed. Clearly, this will increase the cost of the project to provide the necessary straight pipe lengths upstream and downstream of the meters. However, the marginal cost of the piping must be evaluated in terms of the reduced risk to the equipment.

9.10.3 Temperature Instruments

Industry standard thermowells, with thermocouples or resistance temperature detectors, will be used for fluid temperature measurements. External thermocouples, pushed against the outside of the pipe or equipment by a spring, are not acceptable. The small, but inevitable, natural convection air flows around the outside of the thermocouple housing lead to inaccurate measurements.

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Temperature measurements in various locations will not be amenable to the use of thermowells. Examples include the shells of heat exchangers and the outside of a pipe for heat trace control. In these cases, the thermocouple will be welded to the pipe or equipment by means of a welding tab. The thermocouple is covered with a stainless steel tent, which is then tack-welded to the equipment.

Electric heat trace temperature sensors, either thermocouple or resistance temperature detector, are acceptable, but must be standardized across the entire project.

For temperature measurements taken directly from the tank inventories, immersion thermocouples may be used.

Thermocouple extension wire used with salt components must use high temperature ceramic fiber insulation.

9.10.4 Level Instruments

Bubbler level gauges may be used in both cold and hot tanks and vessels, operating at essentially any pressure.

Radar level detectors are suitable for both cold salt service and hot salt service in tanks operating at atmospheric pressure.

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10. Functional / Prescriptive Specifications - Inorganic Heat Transport Fluids

10.1 Introduction

Plant and equipment specifications fall into two broad categories:

- Functional specifications describe what the system or the equipment needs to do, consistent with the minimum legal requirements of the local jurisdictions. The details of how this is to be accomplished is developed by the engineering contractor. This allows the engineering contractor to define the process and the equipment designs such that the functional requirements can be met at the lowest cost
- Prescriptive specifications, which are developed by the Owner, prescribe to the engineering contractor how the functional requirements of the project are to be met. Prescriptive elements include items such as the process design, Code Section selection, fabrication techniques, materials selection, and component details. This arrangement ensures that the favorable experience from a previous project is repeated, and helps to avoid situations in which an inexperienced contractor repeats mistakes from earlier projects.

It can be noted that the use of prescriptive specifications often requires the following:

- The Owner must have a thorough understanding of the process and the technical features of the plant. For example, the heat trace designs in commercial projects often use 1 dual-element thermocouple in each zone as the input to the temperature set point controller. However, a heat trace design, which reflects the experience from previous projects, might use 2 to 4 thermocouples, distributed along the length of the zone, as inputs to the controller. This arrangement can help to detect defects in the insulation, local areas with temperatures below the freezing point of the salt, local areas which are receiving too much heat input from adjacent zones, and leakage past isolation valves.
- The Owner is prepared to accept what is likely to be an increase in the capital cost as a means of achieving the goals set in the performance and in the financial models. For example, the Owner may require the use of all-welded construction in salt heat exchangers. This fabrication approach is more expensive than the conventional approach; i.e., tube rolling followed by strength welding of the tube to the tubesheet. However, an all-welded design is more tolerant of the transient stresses associated with 1) daily startup and shutdown, and 2) mistakes made by the operators. In turn, the improvement in plant availability more than compensates for the marginal increase in the cost of the heat exchangers due to an all-welded design.

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In the sections below is a list of the Design Codes and Standards, followed by a discussion of recommended prescriptive specifications.

10.2 *Design Codes and Standards*

For salt equipment and components, reference codes and standards are listed below for the US market. If the plant is constructed outside of the US, an equivalent matrix will need to be developed for the host country.

<u>Designation</u>	<u>Title</u>
API	Standard 650, Welded Steel Tanks for Oil Storage
ASME	B31.1, Power Piping
ASME	Section I, Rules for the Construction of Power Boilers
ASME	Section II, Materials
ASME	Section III, Division 1, Subsection NH, Class 1 Components in Elevated Temperature Service
ASME	Section V, Non Destructive Examination
ASME	Section VIII, Division 1, Rules for the Construction of Pressure Vessels
ASME	Section VIII, Division 2, Alternative Rules for the Construction of Pressure Vessels
ASTM	A105, Specification for Forgings, Carbon Steel, for Piping Components
ASTM	A181, Specification for Forgings, Carbon Steel for General Service
ASTM	A182, Specification for Forged or Rolled Alloy-Steel Pipe Flanges, Forged Fittings, and Valves and Parts for High-Temperature Service
ASTM	A192, Specification for Seamless Carbon Steel Boiler Tubes for High-Pressure Service
ASTM	A193, Specification for Alloy-Steel and Stainless Steel Bolting Materials for High-Temperature Service
ASTM	A194, Specification for Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service
ASTM	A213, Specification for Seamless Ferritic and Austenitic Alloy-Steel Boiler, Superheater, and Heat Exchanger Tubing
ASTM	A216, Specification for Steel Castings, Carbon, Suitable for Fusion Welding, for High-Temperature Service
ASTM	A240, Specification for Heat-Resisting Chromium and Chromium-Nickel Stainless Steel Plate, Sheet, and Strip for Pressure Vessels
ASTM	A249, Specification for Welded Austenitic Steel Boiler, Superheater, Heat Exchanger, and Condenser Tubes
ASTM	A312, Specification for Seamless and Welded Austenitic Stainless Steel Pipe
ASTM	A325, Specification for Structural Steel Bolts, Steel, Heat Treated, 120/125 ksi Minimum Tensile Strength

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ASTM	A351, Specification for Castings Austenitic Austenitic-Ferritic (Duplex) for Pressure-Containing Parts
ASTM	A36, Specification for Carbon Structural Steel
ASTM	A387, Specification for Pressure Vessel Plates, Alloy Steel, Chromium-Molybdenum
ASTM	A403, Specification for Wrought Austenitic Stainless Steel Piping Fittings
ASTM	A500, Specification for Cold-Formed Welded and Seamless Carbon Steel Structural Tubing in Rounds and Shapes
ASTM	A506, Specification for Steel, Sheet and Strip, Alloy, Hot-Rolled and Cold-Rolled, Regular Quality and Structural Quality
ASTM	A516, Specification for Pressure Vessel Plates, Carbon Steel, for Moderate- and Lower-Temperature Service
ASTM	A53, Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated Welded and Seamless
ASTM	A556, Specification for Seamless Cold-Drawn Carbon Steel Feedwater Heater Tubes
HEI	Heat Exchanger Institute
NEC	National Electric Code
NEMA	National Electrical Manufacturers Association
NFPA	National Electric Code (NEC), National Fire Protection Association (NFPA)
TEMA	Tubular Exchanger Manufacturers Association, 8th Edition TEMA Standards
IBC	International Building Code

10.3 Nitrate Salt Specification

As discussed in Section 9.3, there is a range of inorganic salt options for use as the working fluid in the collector field, the storage system, and the steam generator system. The options include the following:

- Binary nitrate salt (solar salt) mixture, which is 60 weight percent sodium nitrate (NaNO_3) and 40 weight percent potassium nitrate (KNO_3)
- Ternary nitrate/nitrite salt mixture, the most common of which is Hitec[®], with contributions of 7 weight percent sodium nitrate (NaNO_3), 40 weight percent sodium nitrite (NaNO_2), and 53 weight percent potassium nitrate (KNO_3)
- A ternary nitrate salt mixture, Hitec XL[®], with contributions of 42 weight percent calcium nitrate ($\text{Ca}(\text{NO}_3)_2$), 15 weight percent sodium nitrate (NaNO_3), and 43 weight percent potassium nitrate (KNO_3)
- A ternary nitrate salt mixture, which, with contributions of 30 weight percent lithium nitrate (LiNO_3), 18 weight percent sodium nitrate (NaNO_3), and 52 weight percent potassium nitrate (KNO_3), offers the lowest melting point among the ternary nitrate mixtures (~120 °C)

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- A quaternary nitrate salt mixture, reviewed by Sandia National Laboratories, with contributions of 22 weight percent lithium nitrate (LiNO_3), 17 weight percent calcium nitrate ($\text{Ca}(\text{NO}_3)_2$), 18 weight percent sodium nitrate (NaNO_3), and 43 weight percent potassium nitrate (KNO_3)
- A quinary nitrate/nitrite salt mixture, patented by Sandia National Laboratories, with contributions of 13 weight percent lithium nitrate (LiNO_3), 10 weight percent sodium nitrate (NaNO_3), 12 weight percent potassium nitrate (KNO_3), 29 weight percent sodium nitrite (NaNO_2), and 36 weight percent potassium nitrite (KNO_2).

The ideal salt would have 1) a low freezing point, 2) a high upper temperature limit in terms of thermal stability, and 3) be inexpensive. As one might expect, none of the candidate salts fulfill all of the ideal requirements.

The collector field in a commercial project will require several kilometers of piping, both in the collector receiver tubes and in the field headers. All of the piping must be protected from freezing, either in the form of impedance heating on the receiver tubes or electric resistance heating on the field piping. Electric heating, in general, is expensive, and selecting a working fluid with a low freezing temperature reduces the duty, and the cost, of the heating system. Of the candidate salts, the quaternary nitrate salt mixture had the lowest freezing point ($\sim 100^\circ\text{C}$), followed by the quinary nitrate/nitrite mixture (105°C), followed by the ternary nitrate mixture (120°C), Hitec XL (133°C), Hitec (142°C), and the 60/40 mixture (235°C).

With an organic working fluid in the collector field, the maximum temperature is limited to a nominal 392°C based on the thermal stability of the oil. With an inorganic working fluid, the maximum temperature of the working fluid must be greater than 392°C to offset the complexity and the cost of the electric heating system. The benefits of the higher temperature are 1) an increase in the efficiency of the Rankine cycle, 2) a reduction in the unit cost of the storage system, in $\$/\text{kWh}$, due to an increase in the difference between the cold tank and the hot tank temperatures, and 3) a reduction in the cost of the storage system due to a switch from an indirect system to a direct system; i.e., eliminating the oil-to-salt heat exchangers. Previous studies have shown that the collector field outlet temperature should be at least 450°C to offset the cost of the electric heating systems¹¹.

Binary Nitrate Salt

Of the candidate salts, the binary nitrate salt mixture offers the highest thermal stability, with an upper temperature limit of about 600°C . It is also the least expensive mixture, as it does not require lithium or

¹¹ Kearney, D., et. al., (Kearney & Associates, Vashon, Washington), 'Engineering Evaluation of a Molten Salt HTF in a Parabolic Trough Solar Field', National Renewable Energy Laboratory Contract NAA-1-30441-04, January 2001

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nitrite contributions. However, it is the most problematic in terms of freeze protection, as it has the highest melting point.

Hitec

If the collector field outlet temperature is 450 °C, then a representative film temperature in the collector receiver tubes is on the order of 465 to 470 °C. At these temperatures, the nitrite anion component of Hitec undergoes the following decomposition reaction:



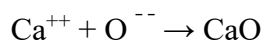
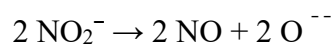
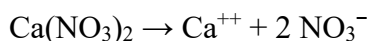
The decomposition reaction is more problematic in Hitec than in a 60/40 salt mixture because the nitrite concentration in Hitec (40 weight percent) is at least an order of magnitude greater than the equilibrium concentration of nitrite in a 60/40 mixture. This applies, even under conditions in which binary salt temperature in central receiver projects is a nominal 565 °C.

The nitrite decomposition reaction is largely irreversible, as the NO produced is vented from the storage tanks as part of the daily charge/discharge cycle. As with the calcium nitrate decomposition (discussed below), NO emissions can represent a health hazard. Also, the oxide species (O^{--}) are chemically aggressive in terms of corrosion in carbon, ferritic, and austenitic steels. Further, the decomposition and the loss of the nitrite ions effectively increases the concentration of the more stable nitrate ions. This, in turn, raises the freezing point of the mixture, and removes some of the motivation to select Hitec as the working fluid.

In general, the use of Hitec at a maximum working temperature of 450 °C is marginally acceptable; at working temperatures above 450 C, Hitec is likely unsuitable.

Hitec XL

At the working temperatures in the receiver tubes, the calcium nitrate in Hitec XL thermally decomposes via the following reactions:



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The calcium oxide (CaO) is largely insoluble and forms a precipitate. The NO is released from the salt as a gas, and can represent a health hazard to the plant personnel. The decomposition reaction results in an inventory mass loss on the order of 1+ percent per year, and the cost to replenish the inventory can represent a significant operating expense to the project. In addition, calcium nitrate is available commercially as the tetrahydrate. The high water content of the hydrate results in a lengthy and a cumbersome dehydration and melting process. In general, the operating problems with Hitec XL are not offset by the benefits of a relatively low melting point.

On a related point, the relative stability of salt mixtures containing lithium nitrate (LiNO₃) (discussed in the section below) is greater than mixtures containing calcium nitrate (Ca(NO₃)₂), as indicated by the maximum temperature at which decomposition products became appreciable. The trend in stability of ternary mixtures followed the correlation for single salts established by Stern ¹². This correlation is based on a parameter of the cation species consisting of the square root of the covalent metallic radius, *r*, divided by the effective nuclear charge, *Z*. Stability increases as the cation parameter increases, which implies that electrostatic effects between the cations and the polyatomic nitrate anions are important. The values of this parameter for the alkali and alkaline earth metals of interest are given in Table 10-1.

Table 10-1 Cation Parameters for Calcium, Lithium, Sodium, and Potassium

Cation	$r^{0.5} / Z$
Ca ⁺⁺	0.38
Li ⁺	0.45
Na ⁺	0.53
K ⁺	0.62

The stability of ternary mixtures appears to be determined primarily by the least stable constituent. The types of cations in the salt mixtures had relatively little effect on the nitrate / nitrite / oxygen equilibrium reaction (i.e., NO₃⁻ ↔ NO₂⁻ + ½ O₂) over the range of compositions investigated. In contrast, decomposition to form corrosive oxide ions was strongly influenced by the types of cations present. This suggests that cations may interact more strongly with nitrite or oxide ions than with nitrate ions.

Ternary Nitrate Salt

The ternary nitrate salt composition should offer an improved thermal stability, compared to Hitec and to Hitec XL, for the following reasons:

- There are no nitrite contributions to the mixtures

¹² Stern, K. H., Journal of Physical Chemistry, Reference Data 1 (1972), 747

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- As noted in Table 10-1, lithium has a higher cation parameter than calcium.

Nonetheless, as summarized in Table 10-2, ternary mixtures containing LiNO₃ can produce significant concentrations of oxide ions ¹³. As a point of reference, the 60/40 binary salt reportedly contains less than 1 * 10⁻² molal total oxide ions under similar conditions of temperature and atmosphere.

Table 10-2 Oxide Concentrations in Ternary Nitrate Salt Mixtures as a Function of Composition

LiNO ₃ :NaNO ₃ :KNO ₃ , weight percent	Melting point, °C	Operating temperature, °C	NO ₂ ⁻ concentration, 10 ⁻² molal	Oxide concentration, 10 ⁻² molal
12 : 18 : 70	200	500	8.5	1.1
		550	20.6	1.2
20 : 28 : 52	160	500	11.1	1.2
		550	30.0	2.9
27 : 33 : 40	150	500	12.6	7.6
		550	27.8	11.0
30 : 18 : 52 ¹	120	500	12.4	11
		550	27.0	20

Note 1: Eutectic mixture

It should be noted that the above data were developed with an ullage gas composition of 100 percent O₂. This was done to prevent the formation of lithium carbonate from the CO₂ normally present in atmospheric air. However, if O₂, rather than air, is used as the ullage gas, this forces the following equation to the left:



This, in turn, reduces the concentration of nitrite ions, which, in turn reduces the formation of oxide reactions via the following reaction:



¹³ Nissen, D. A., and Meeker, D. E., Inorganic Chemistry 22, (1983) 716

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In central receiver projects, the long term nitrite concentration is in the range of 2 to 3 weight percent ¹⁴, and the associated oxide concentrations (which, to some degree, are unknown) result in acceptable corrosion rates for carbon, ferritic, and austenitic steels. A nitrite concentration of 2.5 percent by weight is equivalent to a molal concentration of 5.2×10^{-2} . As indicated in Table 10-2, to achieve comparable corrosion rates between a ternary nitrate salt (using an ullage cover gas of O₂) and the 60/40 binary nitrate salt (using an ullage cover gas of air), the lithium contribution in the ternary salt must be held to values less than 20 percent. This in turn, results in a melting point in the range of 180 to 200 °C.

In addition, the ternary nitrate salt is more expensive than the 60/40 binary nitrate salt because the unit cost of lithium nitrate is approximately 9 times the price of both the sodium nitrate and the potassium nitrate. The cost of the 12 : 18 : 70 ternary salt mixture translates to an increase in the unit cost of storage (\$/kWht) on the order of 50 to 60 percent.

Quaternary Salt

The quaternary salt does not contain nitrite salts, and therefore should avoid the thermal stability limits associated with Hitec. However, the mixture does contain calcium nitrate; as such, the quaternary salt is likely to be constrained to the same upper temperatures limit as Hitec XL. Further, the mixture contains 22 percent by weight lithium nitrate, which translates to an increase in the unit cost of the storage system, relative to the 60/40 mixture, of at least 200 percent.

Quinary Salt

The quinary salt has a nitrite contribution of 65 percent by weight. This value is larger than the nitrite contribution in Hitec (40 percent); as such, the thermal stability limit of the quinary salt is likely to be lower than the thermal stability limit of Hitec. In addition, the unit cost of the quinary mixture will be greater than the unit cost of the 60/40 binary nitrate salt because 1) the quinary mixture contains 13 percent by weight lithium nitrate, and 2) the unit cost of sodium nitrite and potassium nitrite is approximately 50 percent higher than the unit cost of sodium nitrate and potassium nitrate. The cost factors translate to an increase in the unit price of the storage system, relative to the 60/40 mixture, of approximately 150 percent.

Salt Selection

As noted above, the minimum collector field outlet temperature for a trough project using an inorganic working fluid is on the order of 450 °C. This temperature limit essentially eliminates Hitec, Hitec XL, the quaternary nitrate mixture, and the quinary nitrate/nitrite mixture from consideration in a commercial project.

¹⁴ Pacheco, J. E. (Sandia National Laboratories, Albuquerque, New Mexico), 'Final Test and Evaluation from the Solar Two Project', Sandia Report SAND2002-0120, January 2002

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The remaining candidate salts include the following:

- The 60/40 binary nitrate salt mixture, with a low cost and a high thermal stability, but with a high freezing point of 235 °C
- The ternary nitrate mixture, with a freezing point of 180 to 200 °C (depending on the composition), a favorable thermal stability limit of about 540 °C, but with a unit cost of storage that is 50 to 60 percent higher than the unit cost of storage with the 60/40 binary salt.

To select between the two salts, a detailed economic analysis of the combined cost of the thermal storage system and the freeze protection system would be needed. To a zeroth order, the analysis is likely to favor the 60/40 binary mixture, for the following reasons:

- The cost for freeze protection in a plant with a fluid melting point of ~190 °C is not significantly lower than the cost for freeze protection in a plant with a fluid melting point of 235 °C
- There is a considerable data base, totaling several hundred plant years, on the use of the binary salt in commercial service
- The 60/40 binary salt is immune from the swings in the commodity prices of lithium carbonate seen in the past few years.

For the 60/40 binary salt mixture, details on the salt characteristics are presented in the following sections:

- Grades: Section 9.3.1
- Total Chloride Concentration: Section 9.3.2
- Allowable Contaminants: Section 9.3.3
- Supply Options: Section 9.3.4
- Deviations from a 60/40 Mixture: Section 9.3.5.

10.4 Nitrate Salt Handling and Melting Specification

An outline of the salt handling and melting procedures for the 60/40 binary mixture are presented in Section 9.4.

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10.5 General Salt System Equipment

Section 9.5 includes a discussion of salt piping, flanges, fittings, pipe supports, pipe anchors, pipe guides, and valves. However, the discussion is limited to low temperature ($< 390\text{ }^{\circ}\text{C}$), carbon steel salt piping in the thermal storage system.

A description of the high temperature ($500+^{\circ}\text{C}$), stainless steel piping for trough projects using inorganic heat transfer fluids is discussed below.

10.5.1 Piping

Design Codes

The nitrate salt piping can be designed to the requirements of either ASME B31.1 - Power Piping or ASME B31.3 - Process Piping, subject to approval from the local jurisdictional agencies

Collector Field Piping

The salt piping in commercial central receiver projects, which is typically designed to B31.1, has experienced very few, if any, failures associated with material specification, low cycle fatigue, or weld cracking. To some degree, this would argue for the selection of B31.1 for the design of the collector field piping in a trough project using an inorganic heat transfer fluid for the following reasons:

- The design temperatures of the hot salt piping in a trough project using an inorganic heat transfer fluid (510 to $540\text{ }^{\circ}\text{C}$) are similar to the design temperature of the hot salt piping in a central receiver project ($565\text{ }^{\circ}\text{C}$)
- If a project is to meet the availability values typically adopted for project financing, then the field piping in a trough project must have an availability comparable to the availability of the salt piping in a central receiver project; i.e., at least 99.5 percent.

However, in a parabolic trough project using an inorganic heat transfer fluid, the total length of the salt piping in the collector field is comparable to the total length of the Therminol piping in the collector field of a parabolic trough plant using an organic heat transfer fluid; i.e., considerably more than the salt piping in a central receiver project. As such, selecting B31.3 rather than B31.1 for the field piping should offer a lower project cost, due to the following considerations:

- The weld examinations are less rigorous

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- The stress intensity factors and the stress design factors permit fewer expansion loops, or expansion loops with smaller dimensions (assuming that the allowable nozzle loads on equipment connected to the salt piping can be satisfied).

Nonetheless, there is no operating experience with large commercial trough projects using inorganic heat transfer fluids. To some degree, it remains an open topic as to whether B31.3 is an acceptable design code for the field piping.

Steam Generation System and Rankine Cycle

B31.1 is the required design specification for boiler external piping; i.e., the piping upstream of, or downstream from, a steam generator to the first flanged or welded connection. As a practical matter, the following piping is also often designed to B31.1:

- The cold salt and the hot salt piping to, within, and from the steam generator
- The water/steam piping segments in and around the steam generator and the turbine-generator, which operate at combinations of high temperature and high pressure.

Materials

The cold salt piping is typically ASTM A106 Gr B or Gr C, with a nominal 30-year corrosion allowance of 1.6 mm (1/16 in.). The hot salt piping will be a 300-series stainless steel, with a nominal corrosion allowance of 0.7 mm (1/32 in.). The candidate materials include both L-grade materials (Type 304L and Type 316L), and H-grade materials (Type 304H, 316H, and 347H). Recent commercial projects have generally selected Type 347H, but this may not always be the preferred choice. A discussion of the advantages, and the disadvantages, of each option is presented in Section 8 of Volume 3 - Narrative.

Stainless Steel Material Specification

All stainless steel materials are to be supplied in a solution annealed condition.

Stabilization Heat Treatment of Type 347H Piping

The pipe shall conform to ASME SA-312, Grade TP347H.

After solution annealing of the pipe, an additional stabilization heat treatment is to be performed per ASME SA-312 Supplementary Requirement S6. The stabilization process occurs at a temperature which is lower than the temperature which dissolves niobium carbide, but it is higher than the temperature which dissolves chromium carbides, such as Cr₂₃C₆. The material is held at the stabilization

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condition for a period which is long enough to promote the formation of the desired niobium carbides at the expense of the undesired chromium carbides.

The stabilization steps are as follows:

- Raise the temperature to the stabilization temperature at a minimum rate of 280 °C/hour
- Hold at the stabilization temperature of 885 °C ± 14 °C for a minimum period of 2 hours
- Remove the pipe section from the heat source and allow the assembly to air cool to ambient conditions.

The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Stabilization Heat Treatment of Type 347H Wrought Fittings

All wrought stainless steel pipe fittings shall conform to ASME SA-403, WP347H, Class S (Seamless) per ASME SA-403.

After solution anneal of the fittings, an additional stabilization heat treatment shall be performed per ASME SA-403 Supplementary Requirement S2, except it shall be done in a temperature range of 845 to 870 °C and the fittings cooled in air.

The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Stabilization Heat Treatment of Type 347H Forged Flanges and Fittings

All forged stainless steel pipe flanges and fittings shall conform to ASME SA-182, F347H.

After solution anneal of the fittings, an additional stabilization heat treatment shall be performed per ASME SA-182 Supplementary Requirement S10, except it shall be done in a temperature range of 845 to 870 °C and the fittings cooled in air.

The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Post Weld Heat Treatment of Type 347H Pipe and Fittings

It is acceptable for the stabilized piping and fittings to receive an additional stabilization treatment during the post weld heat treatment of spool assemblies in the shop, and during the post weld heat treatment of the field weld that attach to these pipes and fittings.

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10.5.2 Flanges

Flanges in salt service may be of the following types: ring type joint; or hub type. No other types of flanges, such as raised face flanges, will be used due to the potential for relaxation of the bolted connection and leakage past the gasket due to the excellent wetting characteristics of the salt. Ring type joint gaskets and hub rings will be ASTM A182 Gr F21 for low temperature service, and ASTM A240 Gr 304 for high temperature service.

Stud bolts used at temperatures below 400 °C will be specified as ASTM A193 Gr 7. For temperatures above 400 °C, or pressure containing components with external heat tracing, stud bolts will be specified as ASTM A193 Gr B8R, with ASTM A194 Gr 8R heavy hex nuts.

10.6 Heat Tracing

Section 9.8 presents a discussion of scope of supply, zone definitions, zone thermal capacities, control and set point temperatures, component redundancy, component requirements, valve zones, temperature senso installation, and installation details for pressure safety valves, vortex shedding flow meters, tank level gauges, capillary lines for diaphragm pressure transmitters, and salt pumps.

Cable Redundancy

The required number of heat trace cables to satisfy the preheat requirements will be calculated as described in Section 9.8. The number of installed cables will be equal to the number of required cables, plus the following allowances for spare cables:

- 100 percent for the cables 1) on the hot salt piping from the collector field, and 2) the hot salt piping to the steam generator. These segments of piping operate under daily thermal cycles, and the failure rates of the heat trace cables are expected to be 'moderate to high'
- 50 percent for cables on 1) the cold salt piping to the collector field, 2) the cold salt piping from the steam generator, and 3) the vent and drain lines for the steam generator heat exchangers. These segments of piping operate under essentially steady state temperatures, and the failure rates of the heat trace cables are expected to be 'low'.

The spare cables will accommodate failures of the original cables without the need to remove the insulation. The spare cables will be labeled as such, but not connected to the electric power supply.

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10.7 Salt Pumps

The basic features of salt pumps are discussed in Section 9.9. However, the hot salt temperatures in projects using nitrate salt as the collector field working fluid are approximately 125 °C higher than the hot salt temperatures in projects using Therminol as the collector field working fluid. This requires a change in the fabrication materials for the hot salt pump, as follows:

- Column - Stainless steel
- Shaft - Stainless or nickel alloy steel
- Bowls - Stainless steel
- Impellers - Stainless steel
- Shaft bearings - Cobalt chromium alloy; i.e., Stellite.

10.8 Salt Instruments

Pressure instruments, flow instruments, temperature instruments, and level instruments are discussed in Section 9.10.

10.9 Collector Field

10.9.1 Solar Collector Assemblies

The hardware used in the collector field of a plant with an inorganic heat transfer fluid is essentially the same as the hardware used in the collector field of a plant with Therminol as the heat transfer fluid. This hardware includes the foundations, the support structure, the drive, the mirrors, the heat collection elements, and the local controllers.

The principal differences in the hardware lie in the connections between the rotating heat collection elements and the fixed piping. In a project using Therminol, the most common connection is a ball joint. A cross section of a representative design is shown in Figure 10-1. A threaded retainer loads the two seals against the inner ball. A packing material, consisting of grease and graphite flakes, is injected between the two seals to form a liquid-tight boundary. However, all of the candidate salts discussed in Section 10.3 are oxidizing materials. Over time (days to months) the salt reacts with the grease and the graphite to form CO₂. The packing eventually disappears, and the joint begins to leak. Considerable research effort has been devoted to finding either an organic or an inorganic packing material that compatible with salt and provides a liquid-tight boundary, but to date this has proven unsuccessful.

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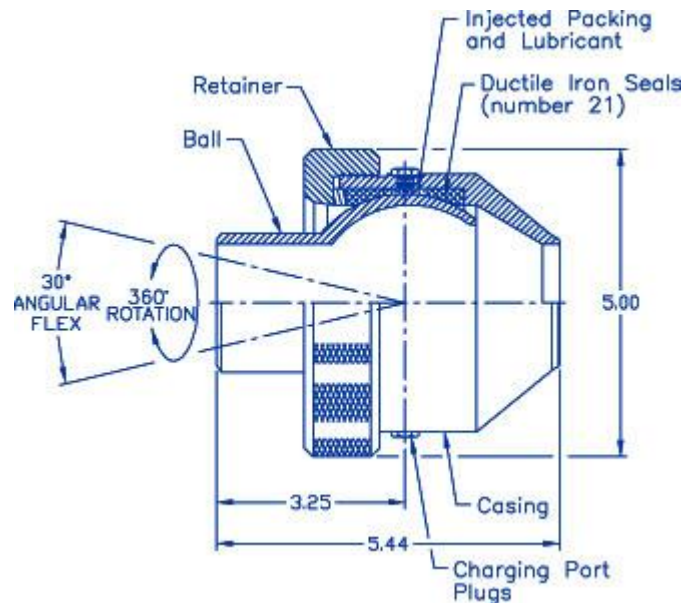


Figure 10-1 Cross Section of a Representative Ball Joint

Instead, the industry has adopted a flex hose arrangement to accommodate 1) the axial thermal expansion and contraction of the receiver tubes, and 2) the rotation between the fixed field piping and the rotating collector assembly. The hose consists of the following:

- An inner bellows, which forms the pressure boundary
- A wire braid, over the inner bellows, to prevent the bellows from crimping
- An outer bellows to protect and to support the wire braid.

A representative flex hose assembly, in its shipping container, is shown in Figure 10-2. The flex hose assembly, installed on a prototype collector, is shown in Figure 10-3. The upper portion accommodates the thermal expansion and contraction of the receiver tubes, and the two lower section accommodate the collector rotation by bending.

10.9.2 Impedance Heating of Receiver Tubes and Flex Hose Assemblies

The absorber tube for the receiver is enclosed in a glass envelope, and a vacuum is drawn in the annular space between the receiver tube and the glass envelope. The receiver tubes require some form of heat addition, other than from solar radiation, for the following purposes:

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Figure 10-2 Flex Hose in Shipping Container



Figure 10-3 Flex Hose Assembly Installed on a Prototype Collector

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- The receiver tubes must be preheated prior to filling with salt to prevent a thermal shock which could break the glass-to-metal seal
- Should salt freeze in the receiver tubes, heat must be added to melt the frozen sections.

Two forms of electric heating have been considered for design of the receiver:

- A conventional heat trace cable, installed on the non-illuminated inside of the receiver tube
- A direct current power supply, which passes a high current through the receiver tubes. The current is converted to heat by the familiar I^2R effect.

The industry has generally not pursued the first approach, for the following reasons:

- The electric cable providing power to the heat trace cable must penetrate the vacuum boundary. The seal for the cable must remain vacuum-tight for decades under thermally cycling conditions. To date, a suitable design for the seal has yet to be identified
- There is a commercial market for parabolic troughs using Therminol as the working fluid; however, there is no commercial market for parabolic troughs using salt as the working fluid. As such, the limited number of commercial receiver suppliers have little motivation to develop a reliable seal for the electric connection.

The impedance heating approach is feasible for receiver tubes due to a combination of a small metal cross section area and a relatively high resistivity for stainless steel. However, based on local jurisdictional requirements, the maximum voltage is limited to either 80 V, based on IEEE requirements¹⁵, or is limited to 50 V, based on NEC Article 110.27¹⁶.

The low voltage allowances result in the following features of an impedance heating system:

- A representative power input to the receiver tubes is 250 W/m. This translates to electric currents in the range of 400 to 600 Amps, depending on the receiver diameter and the wall thickness. The large electric currents result in large transformers, switchgear, and electric cables for the distribution of power to the trough structures, compounded by the need to keep the voltage drops in the range of only a few Volts. For example, a representative size of the electric power cables is illustrated at the top of Figure 10-3, where the combined cross section areas of the three cables approaches the diameter of the receiver tube

¹⁵ "IEEE Recommended Practice for Electrical Impedance, Induction, and Skin Effect Heating of Pipelines and Vessels", Institute of Electrical and Electronics Engineers, Inc., Standard 844-1991, March 1991

¹⁶ 2017 NEC edition of NFPA 70, '110.27(A) Live Parts Guarded Against Accidental Contact'

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- The receiver tube and the flex hose connections are supported by the collector structure. The structure has a much lower electric resistivity than the receiver tubes and the flex hoses. As such, the electric isolation of the receiver tubes and the flex hoses must be essentially perfect. Otherwise, the heating currents will shunt around the receiver tubes and the flex hoses, which decreases the heat input to the receiver tubes and the flex hoses

It can be noted that the electric resistance of the receiver tubes is different than the electric resistance of the flex hose assembly. If the receiver tubes and the flex hoses are to be preheated at the same rate, then separate power supplies, operating at different currents and voltages, will be required for the receiver tubes and the flex hose assemblies.

Design Requirements

As the design of the impedance heating system is fundamentally an electrical problem, each collector loop is considered as a series of electrical resistance elements. The impedance heating system may connect at any point between the resistance elements shown in Figure 10-4, and may be composed of one or multiple power supplies based on the geometry

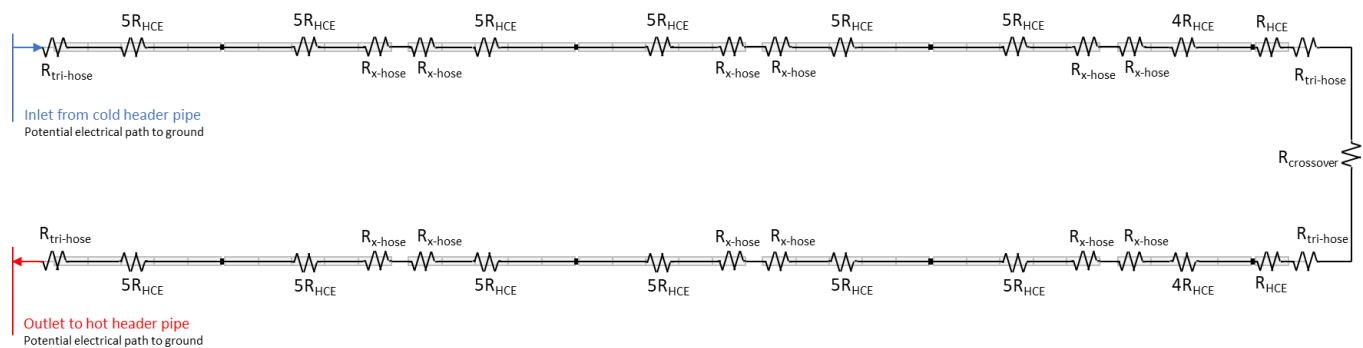


Figure 10-4 Electrical Resistance Network a Collector Loop

The main challenge with impedance heating systems is eliminating paths to ground. The receiver tubes, the flex hose assemblies, and the fixed piping that compose the electrical circuit in Figure 10-4 have numerous paths to ground through the collector structure, and all of the paths need to be electrically isolated. In addition, the field header piping represents additional paths to ground that cannot be easily isolated, but must be considered in the wiring configuration.

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Permanent Systems

A commercial parabolic trough plant may be made up of several hundred collector loops. In principle, permanent impedance systems can be installed on each loop, and the plant could recover from a freeze event in approximately 3 weeks by heating the field piping, the receiver tubes, and the flex hose assemblies simultaneously. In practice, such a system would be limited by the capacity of the plant auxiliary transformers to distribute power from the grid to the various electric heating loads. One study estimates that a maximum of 10 collector loops could be melted at any one time based on the typical capacity of auxiliary transformers in commercial projects ¹⁷. This would result in a freeze recovery time approaching 100 days for the very largest solar fields.

As might be expected, given the large sizes of the electric equipment required for impedance systems, the estimated capital cost to provide a permanent system on each loop is quite high, and may approach \$80/m². This is approximately 50 percent of the unit cost of the collector field, and would render the approach infeasible.

Mobile Systems

An alternate approach to a permanent impedance heating system is a mobile system. The system consists of a number of trailers towed by trucks. On each trailer is a generator, a step-down transformer, and electric cables stored on a series of reels. The cables provide temporary connections between the step-down transformer and permanent electrical connection points in the collector loop.

The truck and each trailer have a common ground bus for connecting the collector structure to the generator neutral. A ground cable also connects to a local permanent grounding well that is bonded to the plant permanent grounding grid.

The estimated cost of a mobile system is on the order of \$1,500,000 ¹⁸. An economic analysis will determine the optimum combination of the number of impedance heating systems and the time to recover from a frozen collector field. For the complete plant, the estimated unit cost of the mobile heating systems is on the order of \$5 to \$10/m² of collector area.

Electric Power Supply Arrangements

The arrangement of the electric connections between the power supply and the receiver tubes, or between the power supply and the flex hose assemblies, has, as its principal goal, the elimination of

¹⁷ C. Prieto, A. Rodríguez-Sánchez, F. J. Ruiz-Cabañas and L. F. Cabeza, "Feasibility study of freeze recovery options in parabolic trough collector plants working with molten salt as heat transfer fluid", *Energies*, vol. 12, no. 12, pp. 1-20, 2019

¹⁸ Shining, R. (Solar Dynamics LLC, Broomfield, CO), "Simplified Melting and Rotation Joint Technology", DOE SunShot Technology to Market Award Number DE-EE0008140, August 2022

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current leakage into the supporting structure. Two configurations have been considered in an effort to reach this goal; midpoint cable; and staggered polarity.

Midpoint Cable The midpoint cable wiring method, illustrated in Figure 10-5, is described in a conceptual design study developed for NREL ¹⁹, and it is described in a freeze protection study which considers a reduced order model to estimate the melt time ¹². This method involves wiring the positive terminal in the middle of the element to be heated, and splitting the negative terminal between the two ends. In this fashion, the ends of two adjacent circuit will always be at the minimum voltage, and the maximum voltage is at the electrical center of the circuit. This wiring configuration can also be advantageous for avoiding current leaks to ground.

In the Figure, the difference between the supply voltage (80 V) and the voltage supplied to the receiver tubes (72 V) is the voltage drop in the electric cables between the power supply and the receiver tubes. Similarly, the difference between the outlet voltage from the receiver tubes (8 V) and the voltage at the power supply (0 V) is the voltage drop in the electric cables between the receiver tubes and the power supply.

Staggered Polarity The staggered polarity method is described in a report looking into alternate methods for freeze protection and recovery ¹⁷. This method involves wiring multiple power supplies in series with opposing polarity to limit the maximum voltage of the total circuit. Only the output terminal of the power supply immediately adjacent to the header piping is grounded. With the staggered polarities and voltage drop of the interconnecting portable cables, the maximum voltage from any section of piping to ground is less than the maximum voltage across any of the power supply terminals.

To illustrate the concept, a collector loop in a potential commercial project is shown in Figure 10-6.

The electric resistance of the loop is represented above in Figure 10-4. A series of portable power supply trailers are placed within the loop, as shown in Figure 10-7.

¹⁹ K. & Associates, "Engineering & Evaluation of a Molten Salt HTF in a Parabolic Trough Solar Field," NREL Contract No. ANN-30441-04, Final Report, 2002

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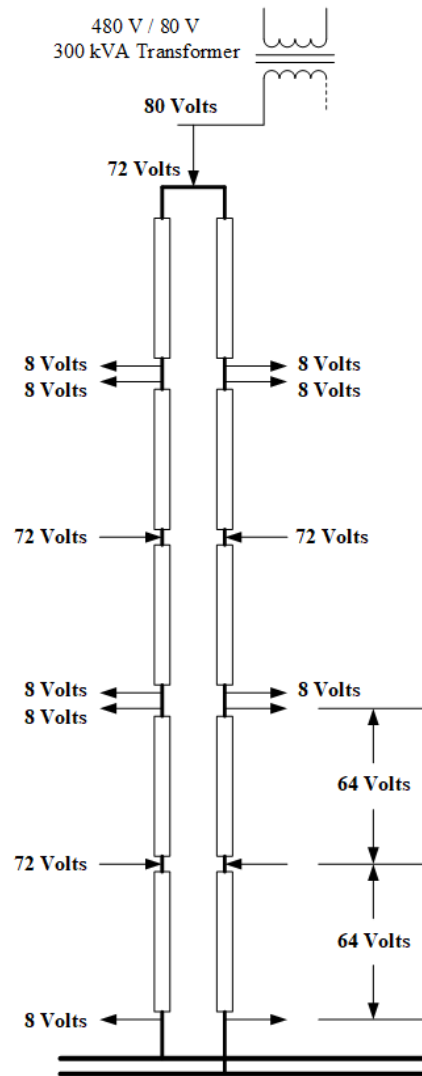


Figure 10-5 Midpoint Cable Wiring Arrangement

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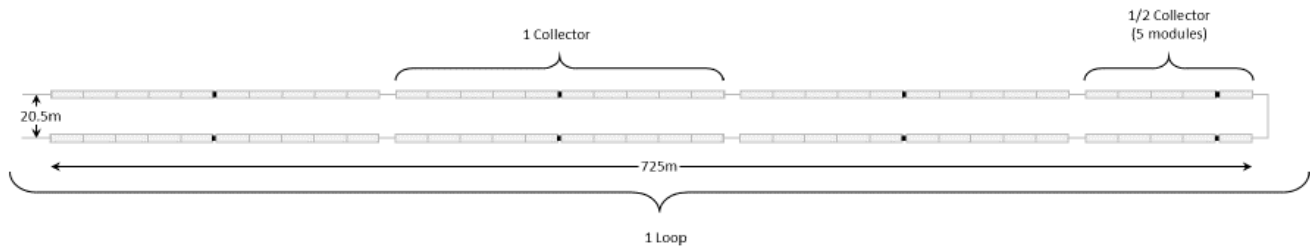


Figure 10-6 Collector Loop in a Potential Commercial Project

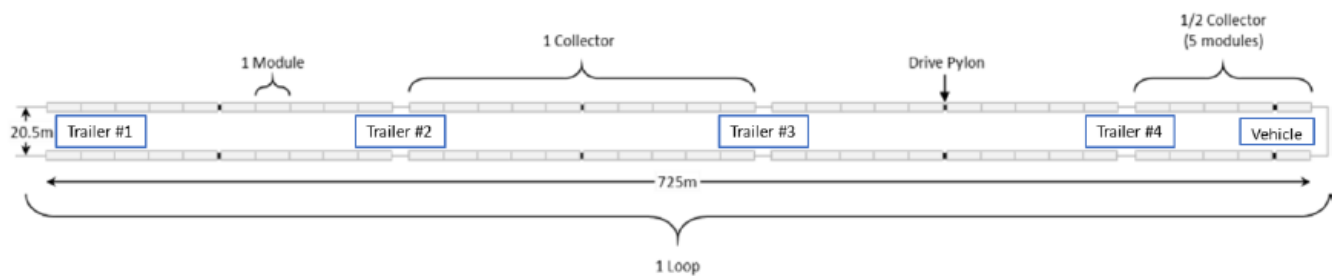


Figure 10-7 Portable Power Supplies for the Impedance Heating System

A conceptual arrangement for the power distribution is shown in Figure 10-8. The green outlines represent the trailers shown in Figure 10-7.

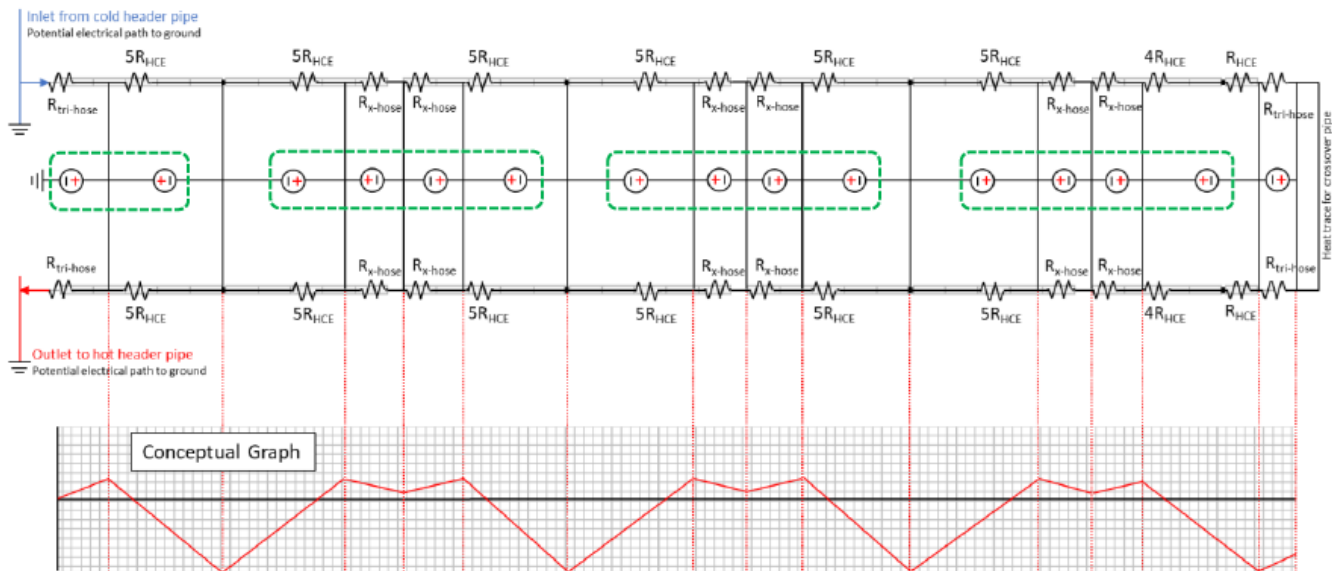


Figure 10-8 Conceptual Power Distribution with Staggered Polarity

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10.9.3 Resistance Heating of the Field Piping

The resistance heating system for the field piping follows the basic design approach discussed in Section 9.8.

As discussed in Section 2.5.3 of Volume 3, Narrative, the heat tracing on the field piping will, at some point in the life of the project, be required to recover the field piping from a frozen condition. This process, as one might imagine, is rather lethargic, with thaw periods ranging from 1 week to 1 month, depending on the diameter of the piping.

Heat tracing is expensive, and the design duty of the system is never generous. As such, the heating rate of the pipe is sensitive to the condition of the pipe insulation. In the plant design phase, an economic analysis will be conducted to determine the optimum combination of the design duty of the heat trace system and the recovery time from a frozen condition. The uncertainties in the analysis are likely be high, as there are no data on the frequency of the freezing events in commercial projects. Nonetheless, the design capacity of the heat tracing should assume some level of degradation in the thermal resistance of the pipe insulation. This is to prevent large deviations between the expected thawing period and the actual thawing period due to slow, but expected, degradation in the quality of the insulation.

10.10 Thermal Storage System

The thermal storage system in a trough project using an inorganic heat transfer fluid is nominally the same as the thermal storage system in a central receiver project. A discussion of the specifications for the thermal storage equipment is presented in Section 6.10 of Volume 2 - Specifications for Central Receiver Projects.

10.11 Steam Generation System

The steam generation system in a trough project using an inorganic heat transfer fluid is nominally the same as the steam generation system in a central receiver project. A discussion of the specifications for the steam generator equipment is presented in Section 6.12 of Volume 2 - Specifications for Central Receiver Projects.

10.12 Steam Bypass System and Air Cooled Condenser Capacities

The criteria for selecting the capacities of the steam bypass system and the air cooled condenser in a trough project using an inorganic heat transfer fluid is nominally the same as the criteria used in a central receiver project. A discussion of the specifications for the steam bypass system and the air cooled condenser is presented in Section 6.14 of Volume 2 - Specifications for Central Receiver Projects.

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11. Current State of the Art - Organic Heat Transport Fluids

11.1 Introduction

The current state of the art in the salt equipment of parabolic trough projects using organic heat transfer fluids is presented in the sections below. Topics include configurations, working fluids, temperatures, rates of temperature change, efficiencies, materials, heat exchangers, pumps, instruments, and valves. The discussion is, in essence, a compilation of functional and prescriptive specifications that incorporate lessons learned from successful commercial plants.

11.2 Configurations

A schematic diagram of a parabolic trough plant with thermal storage is illustrated in Figure 11-1. The Therminol components are shown in red, and the salt components are shown in green. A schematic diagram, showing the salt thermal storage system and the Therminol steam generators, is shown in Figure 11-2.

An elevation view of the storage tanks and the oil-to-salt heat exchangers is shown in Figure 11-3. As discussed in below in Section 11.4, the optimum log mean temperature for the heat exchanger is in the range of 5 to 7 °C. Due to the modest thermal conductivity of nitrate salt, this results in a longer heating length than can be provided in one shell. A representative upper limit on the length of commercial heat exchanger tubes is 30 m (100 ft.), which leads to a requirement for 6 heat exchangers in series.

A plan view of the tanks and the heat exchangers is illustrated in Figure 11-4.

11.3 Working Fluids

The working fluid in the collector field, and the working fluid on the oil side of the oil-to-salt heat exchanger is the eutectic mixture of diphenyl oxide and biphenyl. It is commercially available from Eastman Chemical under the trade name Therminol VP-1.

The working fluid in the thermal storage system, and the working fluid on the salt side of the oil-to-salt heat exchanger, is a mixture of sodium nitrate (NaNO_3) and potassium nitrate (KNO_3). The most common blend is 60 percent by weight sodium nitrate and 40 percent by weight potassium nitrate (64 mole percent sodium nitrate and 36 mole percent potassium nitrate).

A note: This mixture is not the eutectic, which would be a blend of 50 mole percent sodium nitrate and 50 mole percent potassium nitrate. Since the mixture is not the eutectic, the blend has a melting range, rather than a single melting point, as shown in Figure 11-5.

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A 60/40 mixture has been adopted in commercial projects because the sodium nitrate is less expensive than the potassium nitrate, and the moderate deviation from the eutectic does not result in a large increase in the liquidus temperature.

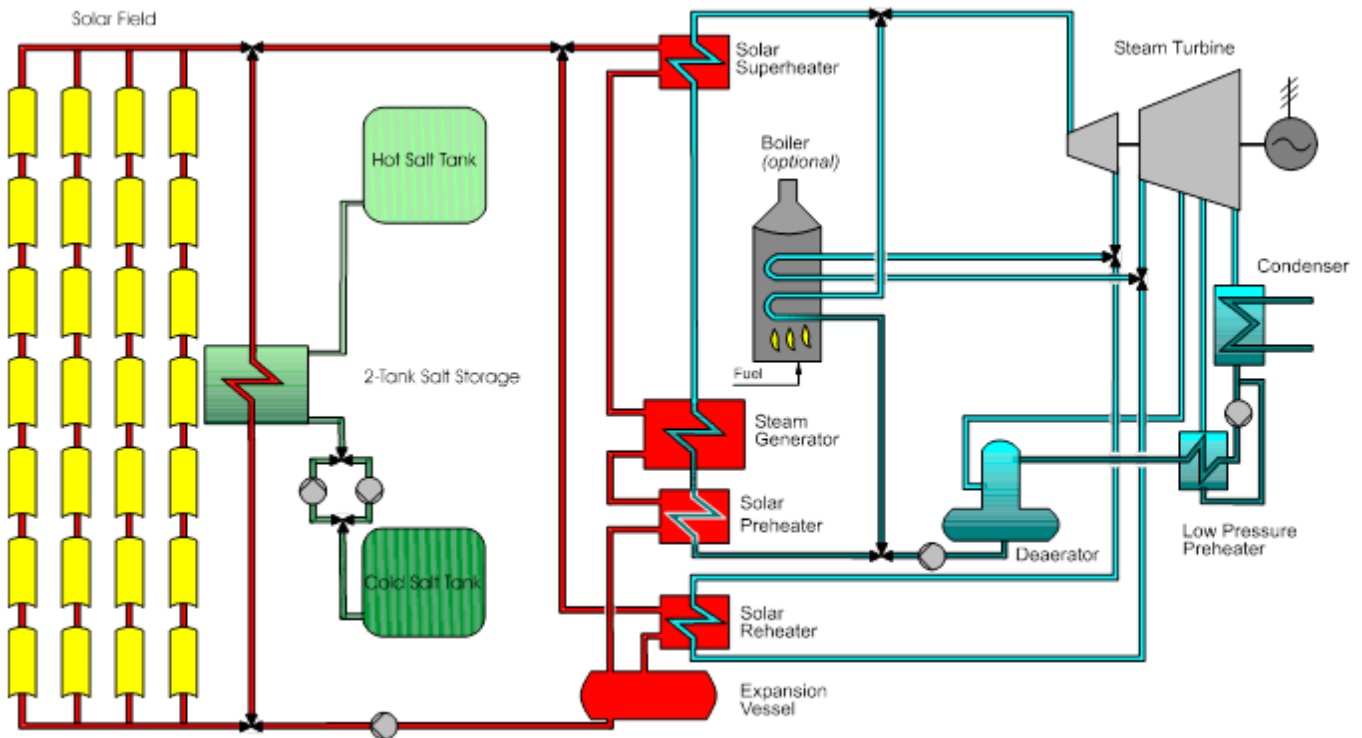


Figure 11-1 Schematic Diagram of a Parabolic Trough Plant with Nitrate Salt Thermal Storage

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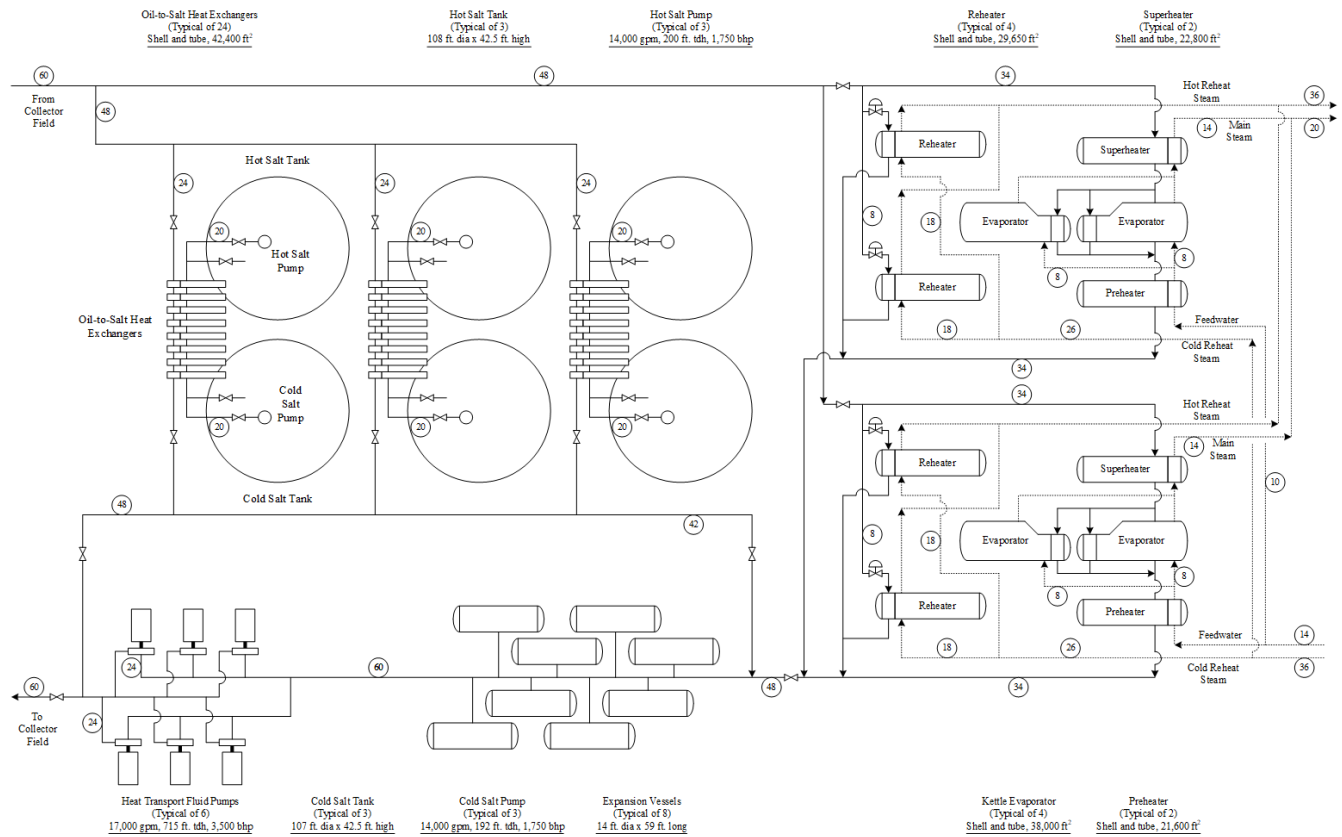


Figure 11-2 Schematic Diagram of Salt Thermal Storage System and Thermanol Steam Generators

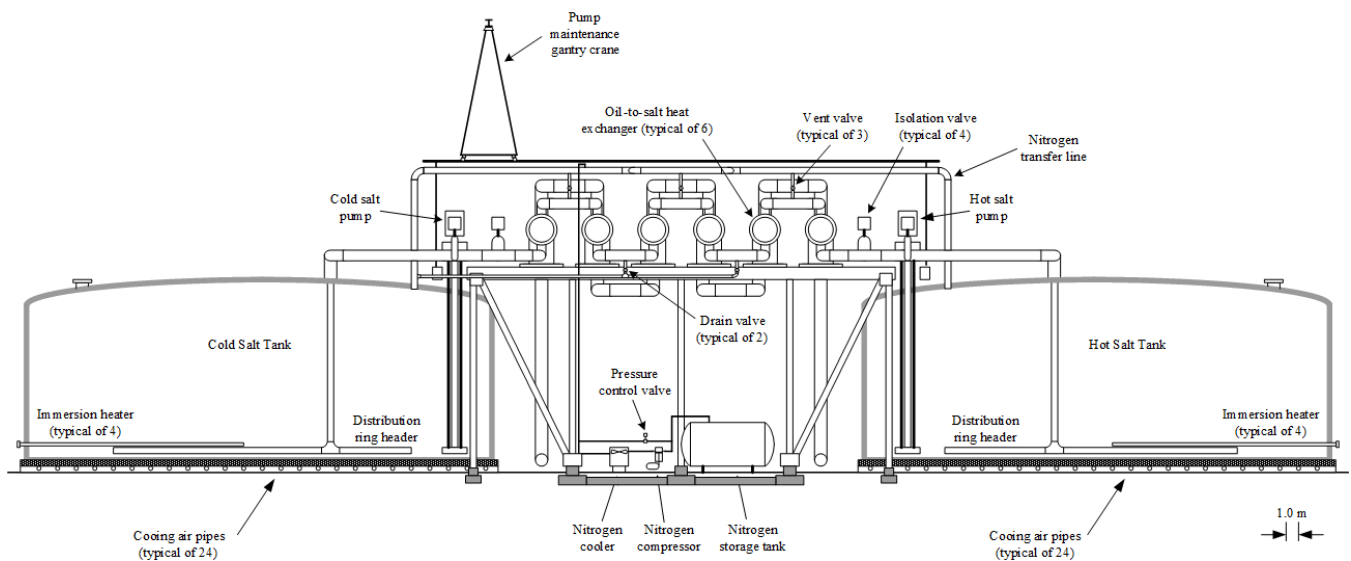


Figure 11-3 Elevation View of Thermal Storage Tanks and Oil-to-Salt Heat Exchangers

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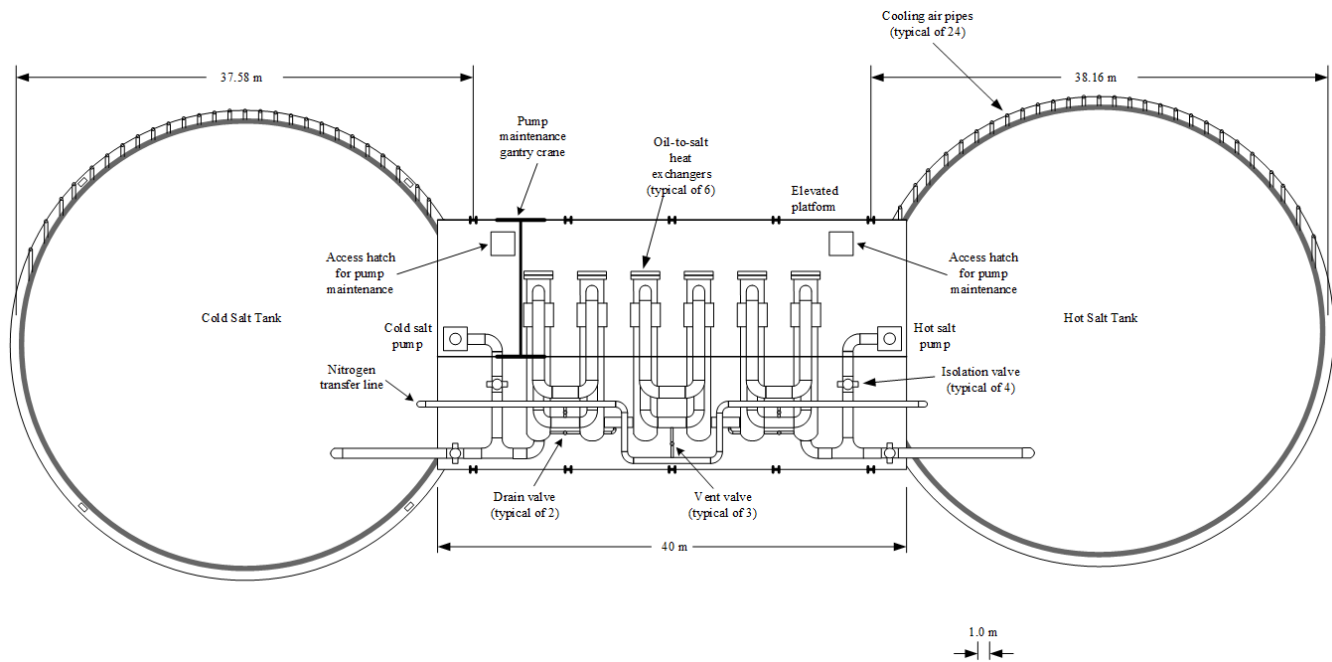


Figure 11-4 Plan View of Thermal Storage Tanks and Oil-to-Salt Heat Exchangers

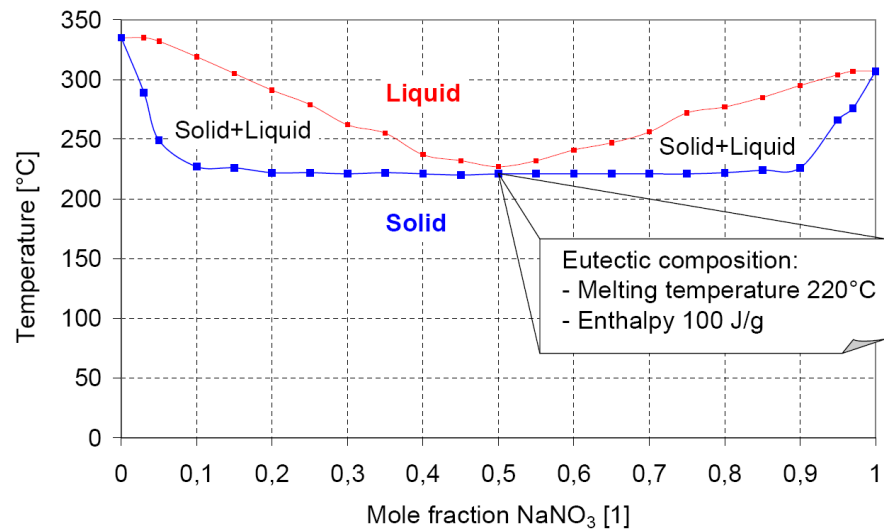


Figure 11-5 Melting Point of Binary Nitrate Salt Mixtures

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11.4 Process Temperatures

11.4.1 Therminol

As with any power plant, the efficiency of the Rankine cycle is maximized by operating at the highest pressure ratio across the steam turbine, and at the highest final feedwater temperature, as practical.

In commercial parabolic trough projects, the source temperature for the Rankine cycle is limited by the maximum allowable operating temperature for the Therminol in the collector field. A representative value is currently 392 °C. At this temperature, the Therminol slowly decomposes, producing a range of light hydrocarbons, such as benzene, and a range of heavy hydrocarbons. The benzene is removed on a semi-continuous basis by exposing the Therminol to a free surface in the expansion vessels, collecting the gaseous benzene released from the liquid, adsorbing the benzene in an activated carbon bed, and transporting the carbon offsite for disposal. The heavy hydrocarbons are separated from the Therminol in a distillation column, and the heavy components are removed from the system in a blowdown stream. The maximum operating temperature of 392 °C is essentially a compromise between power plant efficiency and the operating expense of periodic Therminol makeup.

With a source temperature of 392 °C for the steam generator, the live steam temperature is on the order of 379 °C based on approach temperatures typical of commercial steam generators. With a live steam temperature of 379 °C, and the potential for reheating the steam to 379 °C in a reheater, the live steam pressure can be as high as 100 bar and still acceptable steam qualities (90 to 91 percent) at the exhaust of the low pressure turbine.

With a saturation pressure in the evaporator of about 105 bar, and with a typical commercial pinch point temperature in the evaporator (5 °C) and a typical approach-to-saturation temperatures at the hot end of the preheater (5 °C), the temperature of the Therminol at the cold end of the preheater is essentially defined as 294 °C.

In the storage system, the hot salt temperature is determined by the storage charge condition, and is equal to the design Therminol temperature (392 °C) minus the approach temperature at the hot end of the oil-to-salt heat exchanger. A representative approach temperature is 5 °C, and results from an economic analysis of the storage system, which compares the cost of the oil-to-salt heat exchangers with the cost of the storage inventory. As such, the design temperature of the hot salt tank is a nominal 386 °C.

Conversely, the cold salt temperature is determined by the storage discharge condition. During storage discharge, Therminol is supplied to the steam generator from the storage system at a temperature of 386 °C. Reducing the source temperature to the steam generator reduces the mass flow rate of live / reheat steam, and reduces the live / reheat steam temperatures. Since the turbine normally operates in sliding pressure mode, with the control valves open to 100 percent, reducing the live steam flow rate

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reduces the live steam pressure to values on the order of 92 bar. Reducing the live steam pressure reduces the saturation temperature in the evaporator, which cascades down to a reduction in the Therminol temperature at the cold end of the preheater to a value of approximately 287 °C. A heat transfer analysis on the oil-to-salt heat exchanger shows a corresponding cold salt temperature of 292 °C, and this becomes the temperature of the cold salt tank.

Note that the difference in temperature between the hot tank and the cold tank is a modest 94 °C. This has important implications for the reliability of the storage tanks, as discussed in Section 4.3 of Volume 3 - Narrative.

11.4.2 Silicone Fluids

In addition to Therminol, there is a range of alternate heat transfer fluids based on silicone rather than hydrocarbons. Two high temperature candidates include:

- Syltherm 800, a dimethyl polysiloxane, from Dow Corning Corporation
- Helisol 5A, a polydimethylsiloxane, from Wacker Chemie AG.

According to the suppliers, the silicone fluids can operate at bulk fluid temperatures that are 10 to 25 °C higher than for Therminol. The higher fluid temperatures can, in principle, offer the following benefits:

- Decrease the unit cost (\$/kWh) of the storage system, by increasing the difference in temperature between the hot salt tank and the cold salt tank
- Increase the efficiency of the Rankine cycle, by increasing the temperature of both the live steam and the reheat steam, which allows an increase in the live steam pressure.

A comparison of some of the thermophysical properties of Therminol, Syltherm 800, and Helisol 5A are shown in Table 11-1.

The thermal conductivities, the specific heats, and the viscosities of the 3 fluids are similar, and differences in the designs and the performance of the collector field, the storage system, and the steam generator are likely to be modest. However, the minimum pressure in any equipment containing heat transfer fluid must always be at least equal to the vapor pressure to ensure that the fluid remains in the liquid state. The maximum allowable working pressure for commercial heat collection elements is on the order of 40 bar. As such, the maximum allowable pressure drop in the collector field is as follows:

- Therminol: 40 bar - 11 bar = 29 bar

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Table 11-1 Comparison of Heat Transfer Fluid Properties

	Therminol	Syltherm 800	Helisol 5A
Temperature limit, °C			
Bulk	392	400	425
Film	425	~ 425	~ 450
Vapor pressure, bar	11	14	16
Autoignition temperature, °C	615	385	358
Thermal conductivity, W/m-°C, at 400 °C	0.075	0.064	0.057
Specific heat, kJ/kg-°C, at 400 °C	2.6	2.7	2.5
Viscosity, mPa-sec, at 400 °C	0.14	0.25	0.11

- Syltherm 800: 40 bar - 14 bar = 26 bar
- Helisol 5A: 40 bar - 16 bar = 24 bar.

The allowable pressure drop influences the length of a collector loop. As such, projects using Syltherm 800 or Helisol 5A may require shorter loops than a project using Therminol, which, in turn, can lead to an increase in the number of loops and a second-order increase in the cost of the field piping.

As discussed in Section 10.9.1, differential movement between the heat collection elements and the support structure is accommodated by ball joints. The reliability of the joints is generally very good; however, if the joint does develop a leak, then the heat transfer fluid is exposed to the ambient. The autoignition temperature of Therminol (615 °C) is well above the design fluid temperature (392 °C), and the fluid does not automatically start to burn following a leak. However, the autoignition temperature for the silicone fluids (~ 375 °C) is below the design fluid temperature (400+ °C). As such, a leak is likely to start a fire.

The higher operating temperatures available from silicone fluids may offer benefits which translate to a reduction in the levelized cost of energy. However, the benefits are likely to be modest, particularly in light of the hundreds of project-years of commercial operating experience with Therminol.

11.5 Collector Structure and Reflectors

A trough collector consists of an elongated horizontal structure in the shape of a linear parabola. The structure supports a series of mirrors, each of which represents a segment of a parabola.

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At the focal line of the parabola is a linear receiver. The receiver consists of a metal tube, to which is applied a selective coating. The coating provides a high absorptivity (95 to 96 percent) in the visible spectrum, and a low emissivity (≤ 0.095 percent at 400 °C) in the infrared spectrum.

The metal tube is surrounded by a glass envelope. The envelope has anti-reflective coatings applied to both the inside and the outside of the glass. Air is removed from the annular space between the metal tube and the glass envelope to reduce convection losses. This necessitates the use of a metal bellows, and a glass-to-metal seal, to accommodate the differential thermal expansion between the metal receiver tube and the glass envelope. A representative gas pressure in the annular space 0.0001 mm Hg (0.013 Pa). Getters, which are metallic substances that adsorb gas molecules, are installed in the vacuum space to absorb hydrogen and other gases that permeate into the annular space over time.

Each structure tracks the sun by rotation about a horizontal axis. The drives for the structure are typically a pair of hydraulic rams.

The mirrors are made from a low iron float glass that is silvered on the back and then covered with several protective coatings. The mirrors are formed into a parabolic trough by heating the glass, and then sagging, over parabolic molds in specialty ovens. Ceramic pads are used for mounting the mirrors to the collector structure with a specialty adhesive.

The width of each collector structure ranges from 5 m to 8.2 m, depending on the vendor. A number of collectors are bolted to one another to form a collector assembly. Each assembly has a drive, and the length of the assembly ranges from 100 m to 250 m, again depending on the vendor.

Commercial receivers are available with tube outer diameters of 70, 80, and 90 mm. This results in concentration ratios (collector aperture divided by tube diameter) in the range of 60 to 110. The selected concentration ratio is a function of 1) the optical accuracy of the structure and the mirrors, 2) process temperatures, and 3) allowable fluid velocities in the tubes.

11.5.1 Optical Efficiencies

A representative set of optical efficiencies for the structure and the mirrors is listed in Table 11-2, and a representative set of optical efficiencies for the receiver tube are shown in Table 11-3.

The product of the collector optical efficiency times the receiver optical efficiency is on the order of 0.756. In essence, for every 1,000 photons normally incident on the collector aperture, 756 are absorbed by the receiver tube.

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Table 11-2 Representative Collector Optical Performance Factors

Item	Factor
Tracking error and twist	0.990
Geometric accuracy	0.980
Mirror reflectivity	0.940
Mirror cleanliness	0.950
Product	0.866

Table 11-3 Representative Receiver Optical Performance Factors

Item	Factor
Dust on glass envelope	0.980
Bellows shading of tube	0.963
Glass envelope transmissivity	0.963
Tube absorptivity	0.960
Product	0.872

11.5.2 Heat Losses and Thermal Efficiencies

Thermal losses from the receiver tube include a combination of convection, conduction, and radiation heat transfer. A thermal model of the receiver is illustrated in Figure 11-6. The calculation of the thermal losses, which is rather involved, is detailed in a report from NREL ²⁰.

Examples of calculated receiver efficiencies and unit heat losses (W/m), as functions of the absorptivity of the selective surface and the average heat transfer fluid temperature, are shown in Figure 11-7. Due to the very low gas pressure in the annular space between the receiver and the glass envelope, convection losses from the receiver are generally low. As such, the principal loss mechanism is radiation, which is responsible for the parabolic shapes in the efficiency and heat loss curves as functions of the average fluid temperature.

If Therminol is the heat transfer fluid, then the design collector field outlet temperature is 393 °C. With a maximum receiver tube temperature of perhaps 400 °C, the selective surface chosen for the tubes has a

²⁰ Forristall, R., (National Renewable Energy Laboratory, Golden, Colorado), 'Heat Transfer Analysis and Modeling of a Parabolic Trough Solar Receiver Implemented in Engineering Equation Solver', NREL/TP-550-34169, October 2003

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combination of absorptivity of 0.96 and emissivity of 0.09 to 0.095 which maximizes the efficiency for temperatures up to 400 °C ²¹.

Examples of calculated receiver efficiencies and unit heat losses, as functions of the status of the receiver tube and the wind speed, are shown in Figure 11-8. The values represent receiver tubes for use with Therminol as the heat transfer fluid. The receiver tube can be 1) in an as-new condition with the vacuum intact, 2) in a partially degraded condition with air in the annular space between the tube and the glass envelope (lost vacuum), and 3) in a fully degraded condition with a broken glass envelope.

As noted in Figure 11-8, unit heat losses, at an average tube temperature of 250 °C and over a range of wind speeds, are on the order of the following:

- 150 to 300 W/m with the receiver tubes in an as-new condition
- 300 to 500 W/m with air in the annular space between the tube and the glass envelope
- 1,100 to 2,300 W/m with a broken glass envelope.

²¹ 'Optical and thermal characterization of parabolic trough solar receivers, Shandong Huiyin Energy Co., Ltd., 903, Test Report', Fundación CENER-CIEMAT, Departamento de Energía Solar Térmica, Laboratorio de Ensayos Solares Térmicos, Test Report 30.3429.0-2 Annex 6, 31/08/2018

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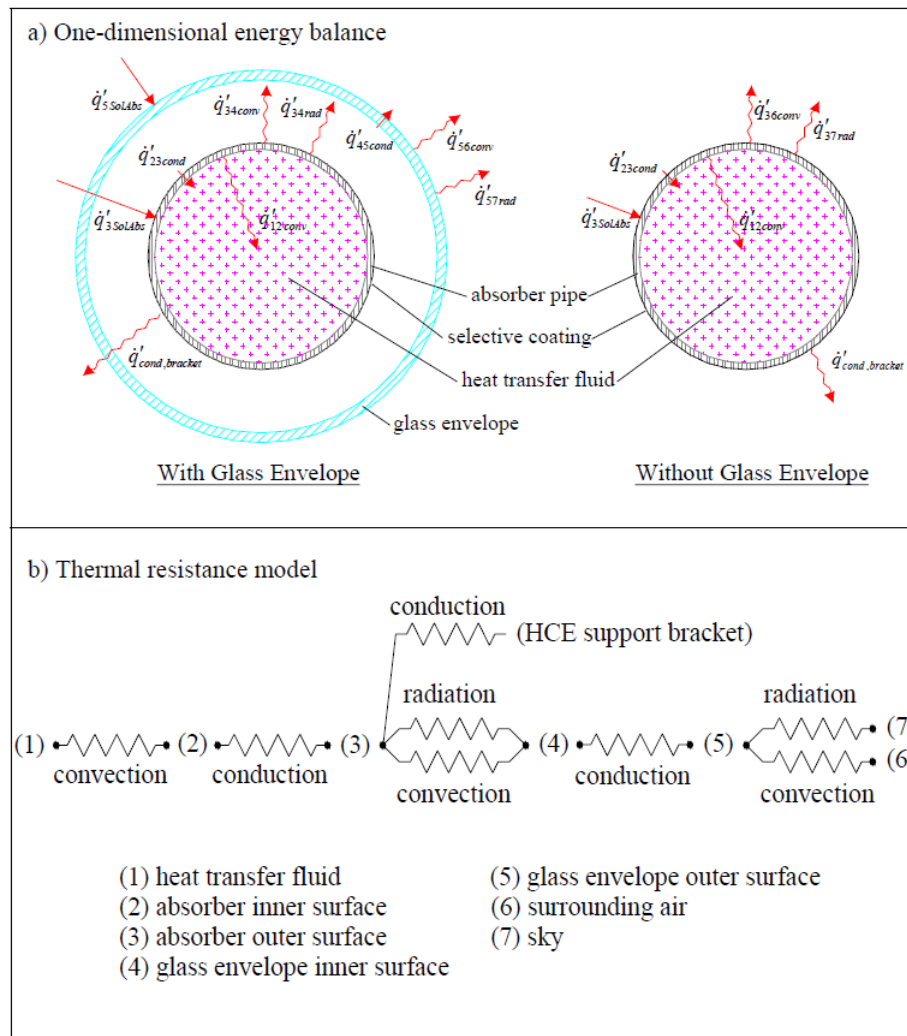


Figure 11-6 Convection, Conduction, and Radiation Heat Exchange in a Receiver Tube

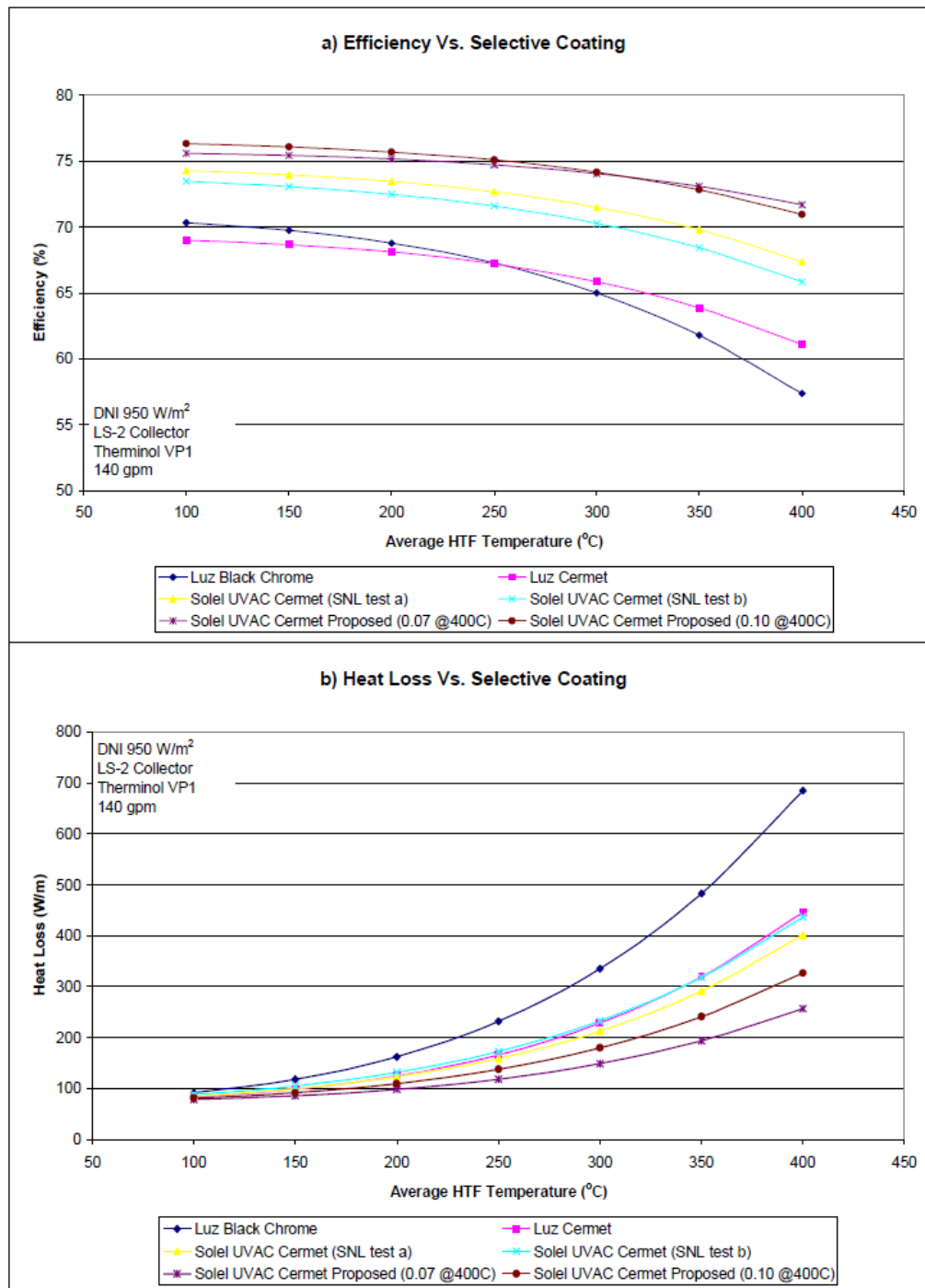


Figure 11-7 Receiver Efficiency and Heat Loss as Function of the Absorber Coating

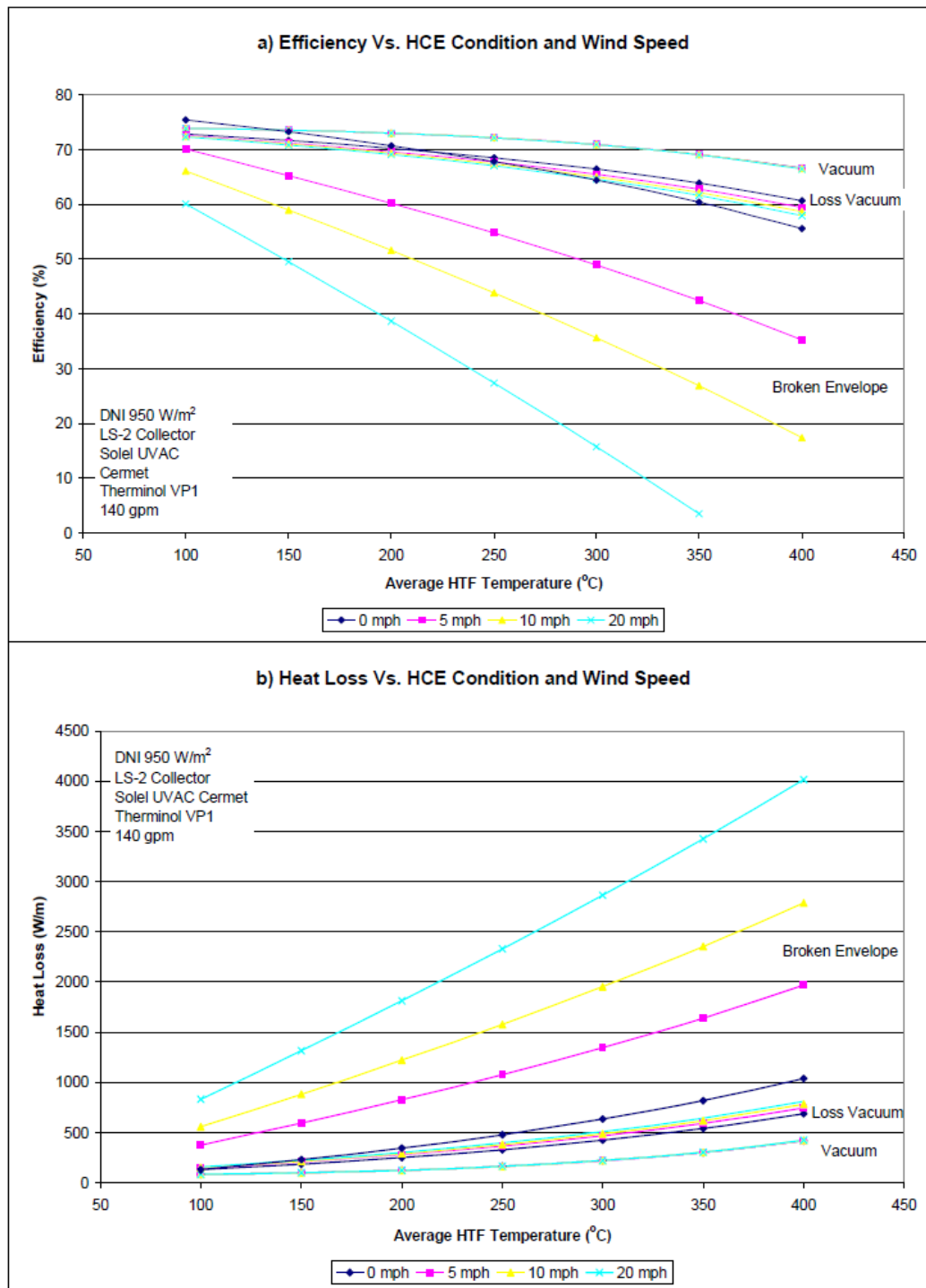


Figure 11-8 Receiver Efficiency and Heat Loss as Function of the Tube Condition

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11.5.3 Collector Drives and Controls

The collector assemblies rotate around the horizontal axis to track the sun. The axis can be aligned either North/South or East/West. The latter offers the best annual sunlight-to-thermal efficiency. However, the former is normally adopted for commercial projects, as it offers a better daily efficiency during the important summer months.

The axis of rotation is located at the collector center of mass to minimize the required tracking power. The drive system uses hydraulic rams to position the collector. A closed loop tracking system relies on a sun sensor for the +/- 0.1 degree accuracy required to properly focus the sun on the receiver tubes. The tracking is controlled by a local controller located on each collector assembly. The local controller also monitors the heat transfer fluid temperature and reports operational status, alarms, and diagnostics to the main solar field control computer.

The collector is designed for normal operation in winds up to 40 km/h and somewhat reduced accuracy in winds up to 56 km/h. The assemblies are designed to withstand a maximum of 113 km/h winds in their stow position, with the collectors aimed 30° below the horizon.

11.6 Designs and Rates of Temperature Change

11.6.1 Oil-to-Salt Heat Exchanger

In most commercial projects, there is a direct connection between the hot header returning from the collector field and the hot end of the oil-to-salt heat exchanger. As such, the rates of temperature change seen in the hot oil header, during both morning startup and cloud transients, are transmitted directly to the oil-to-salt heat exchanger. To a first order, the oil-to-salt heat exchanger must 1) accommodate a rate of temperature change of 10 °C/min, and preferably 12 °C/min, and 2) provide a low cycle fatigue life of at least 30,000 cycles.

Essentially all commercial projects use a shell-and-tube design for the oil-to-salt heat exchanger, including the following geometries:

- U-tube bundle / straight shell, with a single tubesheet. Differential thermal expansion between the tube bundle and the shell is accommodated by bending of the tubes in the U-section of the bundle
- Straight tube / straight shell, with a fixed tubesheet and a floating tubesheet. Differential thermal expansion between the tube bundle and the shell is accommodated by lateral movement of the floating tubesheet.

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The most common approach to attaching the tubes to the tubesheet is 1) strength welding the ends of the tubes to the face of the tubesheet, and 2) plastically expanding the tubes into the tubesheet. The principal sealing mechanism between the high pressure fluid on the tube side (Therminol) and the low pressure fluid on the shell side (salt) is the friction connection between the outside of the tubes and the bores in the tubesheet. The strength weld is the backup fluid boundary in the event that the friction connection relaxes.

To provide a log mean temperature difference of 5 to 7 °C at the design point, large heat transfer areas are required. Even using heat exchangers with the largest shell diameters typically available (2.75 m), 4 to 6 heat exchangers in series are typically required. One alternate approach to multiple shells is to use a flat plate heat exchanger. The corrugated plates are arranged in a stack, with alternating layers of oil flowing in one direction and nitrate salt flowing in the other. The edges of the plates are welded to form a pressure boundary, and an external frame supports the hydraulic forces. The advantages of a plate heat exchanger are very large heat transfer areas in a single unit, and low pressure drops for both fluids. The principal liability of the design are marked changes in metal thickness where the thin plates meet the edge welds, and where the edge welds meet the external frame. To control the transient thermal stresses in these regions, the allowable rate of temperature change is limited to 1.5 °C/min. It is very difficult to comply with this rate, particularly if the hot header from the collector field is connected directly to the hot end of the oil-to-salt heat exchanger. At one commercial project, at least two heat exchangers experienced significant cracking after perhaps 6 months of commercial service.

Nonetheless, this is not to imply that a plate heat exchanger cannot be suitable for commercial use. Specifically, the heat exchanger needs to be operated in a manner similar to the steam generator in a central receiver project. This involves the following steps:

- Prior to daily startup, the heat exchanger is maintained at an uniform temperature of 295 °C by establishing a flow of cold salt at the cold end of the heat exchanger. Salt leaving the hot end of the heat exchanger is returned to the cold salt tank.
- Shortly after sunrise, a flow of Therminol from the hot header of the collector field is sent to the hot end of the heat exchanger. However, if the rate of temperature change in the oil in the hot header exceeds 1.5 C/min, then the flow is attenuated by blending cold oil from the discharge of the field circulation pumps with the oil from the hot header at a mixing station upstream of the hot end of the heat exchanger.
- The flow rate of cold salt to the heat exchanger is adjusted such that the temperature of the salt at the hot end of the heat exchanger increases at the allowable rate of 1.5 °C/min.
- If the salt temperature at the hot end of the heat exchanger is below a threshold value, then the salt is returned to the cold tank to prevent a decay in the inventory temperature in the hot tank. Once the threshold temperature is exceeded, salt is directed to the hot tank.

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- Cloud transients over the collector field will cause a decay in the temperature of the hot Therminol header returning from the field. The temperature decay will be transmitted to the hot end of the heat exchanger. The flow rate of salt will be adjusted to match the duty on the salt side with the duty on the Therminol side. If the salt temperature at the hot end of the heat exchanger falls below the threshold value, then the salt is returned to the cold tank.
- At the end of the day, two scenarios can arise:
 - The storage system will immediately switch from a charge mode to a discharge mode. In this case, the flows on both the salt side and the oil side are stopped to preserve the normal temperature distribution the heat exchanger. The two fluid flow directions can then be reversed without changing the temperature profile in the heat exchanger.
 - The storage system will switch from a charge mode to a discharge mode after some delay period. The heat exchanger is returned to a uniform temperature of 295 °C by essentially reversing the startup process. At the beginning of the discharge process, the heat exchanger is restarted, but with the fluid flows in the reverse direction. This will require the use of a cold salt / hot salt mixing station upstream of the hot end of the heat exchanger.

11.6.2 Storage Tanks

Preheating with Combustion Gases

For the large storage tanks typical of a commercial project, the allowable rate of temperature change for either the cold tank or the hot tank during preheating is 4.5 °C/min. Nonetheless, the more important temperature limits, in terms of ensuring the low cycle fatigue lives of the tanks, are the following:

- 25 °C between any two points on the wall
- 33 °C between any two points on the floor.

Normal Operation

The allowable rate of temperature change for either the cold tank or the hot tank during normal operation is 1 °C/min. As above, the more important temperature limits are the following:

- 25 °C between any two points on the wall.
- 15 °C between any point on the wall and any point on the floor (or the roof)

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- 5 °C between the center of the floor and the perimeter of the floor.

11.7 *Efficiencies*

In a commercial parabolic trough project, a representative storage tank has a diameter of 40 m and a wall height of 14 m. The active storage mass is on the order of 25,000 tons. With a cold tank temperature of 292 °C, and a hot tank temperature of 386 °C, the nominal storage capacity is 930 MWht.

Representative flux fluxes through the insulation for each tank are 75 W/m² through the roof, 60 W/m² through the wall, and 60 W/m² into the foundation. As such, the steady state heat losses from the storage system are about 15 MWht. This results in a daily storage efficiency of approximately $(930 \text{ MWht} - 15 \text{ MWht}) / 930 \text{ MWht} = 0.98$.

11.8 *Materials*

11.8.1 Oil-to-Salt Heat Exchanger

If the heat exchanger is a shell-and-tube design, the shell, the channel, and the tubesheet can be fabricated from carbon steel components. A nominal salt-side corrosion allowance is 3 mm (1/8 in.). In contrast, the tube wall thicknesses required to withstand the absolute and the differential fluid pressures are in the range of 1.65 to 2.1 mm. Since it is impractical to add a corrosion allowance of 3 mm to a thin wall tube, a tube material is selected which requires essentially no allowance for corrosion. The candidate materials include various low chromium ferritic steels, or an L grade of a 300-series stainless steel. The industry has often defaulted to stainless tubes as these are available in longer lengths (30 m, 100 ft.) than ferritic tubes.

As discussed in Section 11.2, 6 heat exchangers in series are typically needed to provide the required heating length. The temperature profile along the heating length during a charge cycle is nominally the same as the temperature profile along the heating length during a discharge cycle. As a cost savings measure, some commercial projects have used carbon steel tubes for the 3 heat exchangers at the cold end of the temperature profile, switching to stainless steel tubes for the 3 heat exchangers at the hot end of the temperature profile.

If the heat exchanger is a plate design, the plates are fabricated from stainless steel, and the supporting structure is fabricated from carbon steel.

11.8.2 Storage Tanks

If the project uses the technical or the refined grade of the salt, the chloride content is relatively low, and the corresponding corrosion rates are moderate. As such, carbon steel is a suitable fabrication material for both the cold tank and the hot tank. Nonetheless, it can be noted that stress corrosion cracking of

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carbon steel has been observed in the tank at one commercial project. The topic is discussed above in Section 5.2.1.

If the project uses the industrial grade of sodium nitrate, rather than the technical or the refined grade, then the chloride content is likely at a level that justifies the use of a low chromium ferritic steel, such as 2¼ - 1 Mo, for the hot salt tank.

11.8.3 Salt Pumps

For both the cold salt and the hot salt pumps, representative materials of construction include the following:

- Motor stand, column, bowls, and suction bell: Carbon steel
- Pump shaft: Low chromium ferritic steel
- Impellers: Martensitic stainless steel
- Shaft bearing: Cast iron.

11.9 Instruments

Recommended approaches for the type and the installation of instruments in salt service are discussed above in Section 9.10.

11.10 Salt Valves

Recommended approaches for the types of valves in salt service are discussed above in Section 9.5.2.

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12. Current State of the Art - Inorganic Heat Transfer Fluids

12.1 Introduction

As discussed in Section 10.3, the most likely candidate for the working fluid in a parabolic trough plant with inorganic heat transfer fluids is the 60/40 binary nitrate salt mixture. For the purposes of the discussion below, the working fluid is assumed to be this binary mixture.

Parabolic trough projects using the binary salt working fluid have similar arrangements to central receiver projects using the same binary salt working fluid. The collector system is supplied with cold salt from a cold salt tank. The salt is heated to the nominal hot side temperature, and delivered to a hot salt tank. Salt from the hot tank is supplied to a steam generator, and the salt leaving the steam generator is returned to the cold tank to complete the cycle. The steam generator supplies live steam and reheat steam to a conventional single-reheat Rankine cycle for electric power production.

To date, trough projects using binary salt as the working fluid have been limited to the following:

- An ENEA research facility near Rome, using one-half of a collector loop
- A 9 MWe equivalent trough field built by Enel near Priolo Gargallo, Sicily. The project has a thermal storage capacity of 8 hours, and supplies steam to an existing power plant
- A 1 MWe demonstration project near Massa Martana, Italy, developed by Archimede Energy and Chiyoda. The plant has 5 hours of thermal storage.

No commercial plants have been built to date, for the following reasons:

- A proven means for freeze recovery in the collector field has yet to be demonstrated. Various approaches have been proposed, including permanent impedance heating systems, portable impedance heating systems, and oscillating the collectors into, and out of focus, for limited periods of time. Nonetheless, each of the options have various liabilities, as follows:
 - The permanent systems are expensive, and can increase the unit cost of the collector field by as much as one-third
 - The time required for portable systems to recover from a completely frozen field ranges from days to weeks
 - The optical heating approach requires several hours of uninterrupted radiation, and if not performed correctly, has the potential for permanently damaging the receiver tubes.

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- Several years ago, the gross Rankine cycle efficiency of plant using Therminol, and operating with single reheat steam conditions of 100 bar / 370 °C / 370 °C, was a nominal 0.375. Today, gross cycle efficiencies in the range of 0.39 to 0.41 are possible due to improvements in the isentropic expansion efficiencies of steam turbines. The improvement in cycle efficiency has reduced some of the motivation to switch heat transfer fluids.

In general, the increase in project risk of a binary salt heat transfer fluid relative to a Therminol heat transfer fluid, particularly in terms of plant availability, may not offset the benefits of a lower cost for the storage system and a higher efficiency for the Rankine cycle. Since there are no commercial projects using salt as the working fluid, the current state of the art discussed below is somewhat conjectural.

12.2 Process Flow Diagrams

A simplified process flow diagram of a parabolic trough plant using salt as the working fluid is shown in Figure 12-1. Unlike trough projects using Therminol as the working fluid, the storage system does not require an intermediate heat exchanger, and the working fluid in the steam generator is salt. A simplified flow diagram on the water/steam side of the steam generator is shown in Figure 12-2.

12.3 Collector System

The collector system in a trough project using salt as the working fluid is nominally the same as the collector system in a trough project using Therminol as the working fluid, as described in Section 11.5.

12.3.1 Absorber Selective Surface

If nitrate salt is the heat transfer fluid, then the design collector field outlet temperature is in the range of 500 to 540 °C. With a maximum receiver tube temperature of perhaps 560 °C, a different selective surface is chosen compared to system based on Therminol. The selective surface for use with nitrate salt sacrifices a portion of the absorptivity (0.92, rather than 0.96) in favor a lower emissivity (0.07, rather than 0.095) at 400 °C. An example of emissivity values, over a range of absorber tube temperatures, for Therminol and nitrate salt applications is shown in Figure 12-3 ²².

²² Ruegamer, T., et al, (SCHOTT Solar CSP GmgH, Mitterteich, Germany), 'Molten Salt for Parabolic Trough Applications: System Simulation and Scale Effects', SolarPACES 2013, Las Vegas, Nevada

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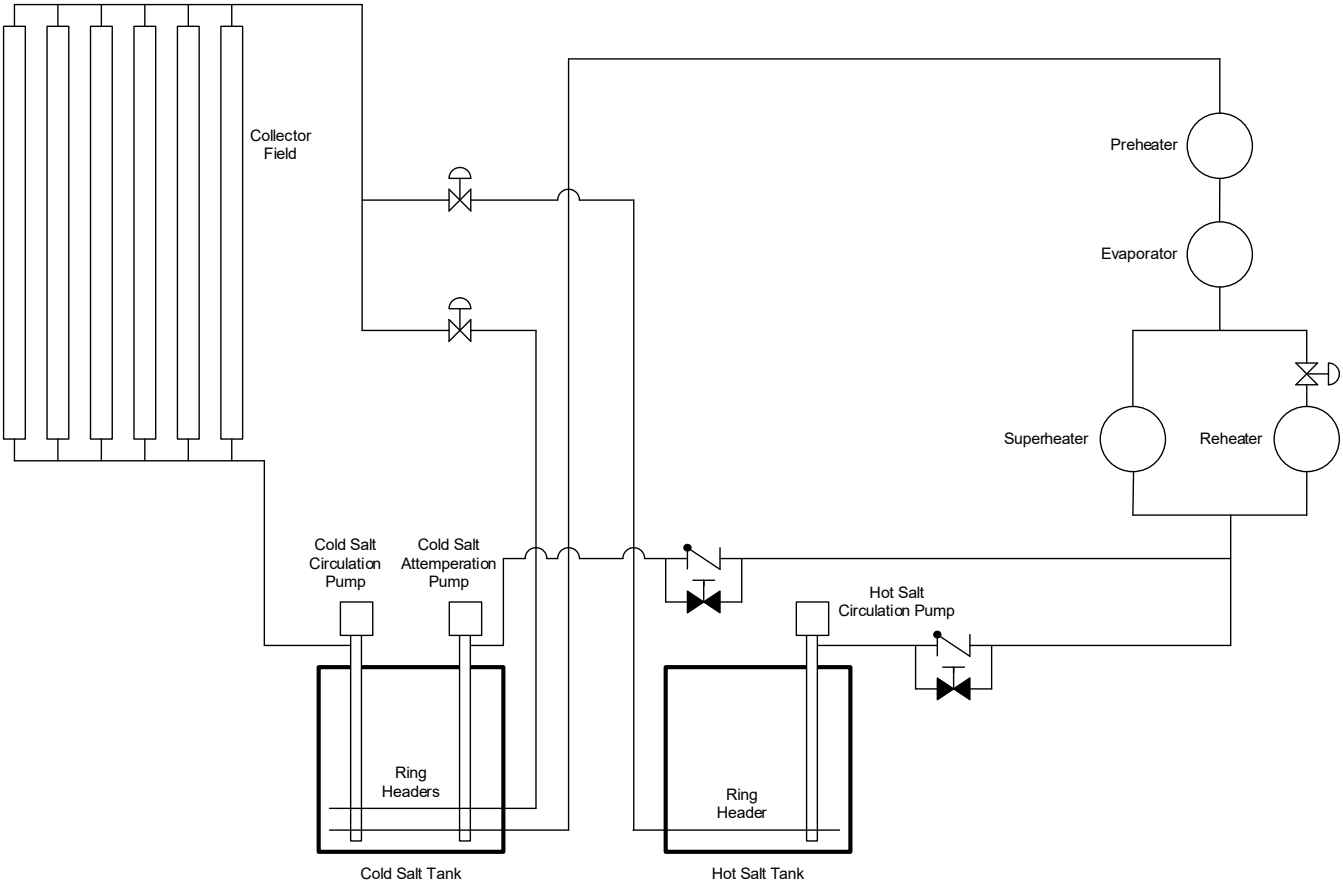


Figure 12-1 Simplified Process Flow Diagram of a Trough Project using Salt as the Working Fluid

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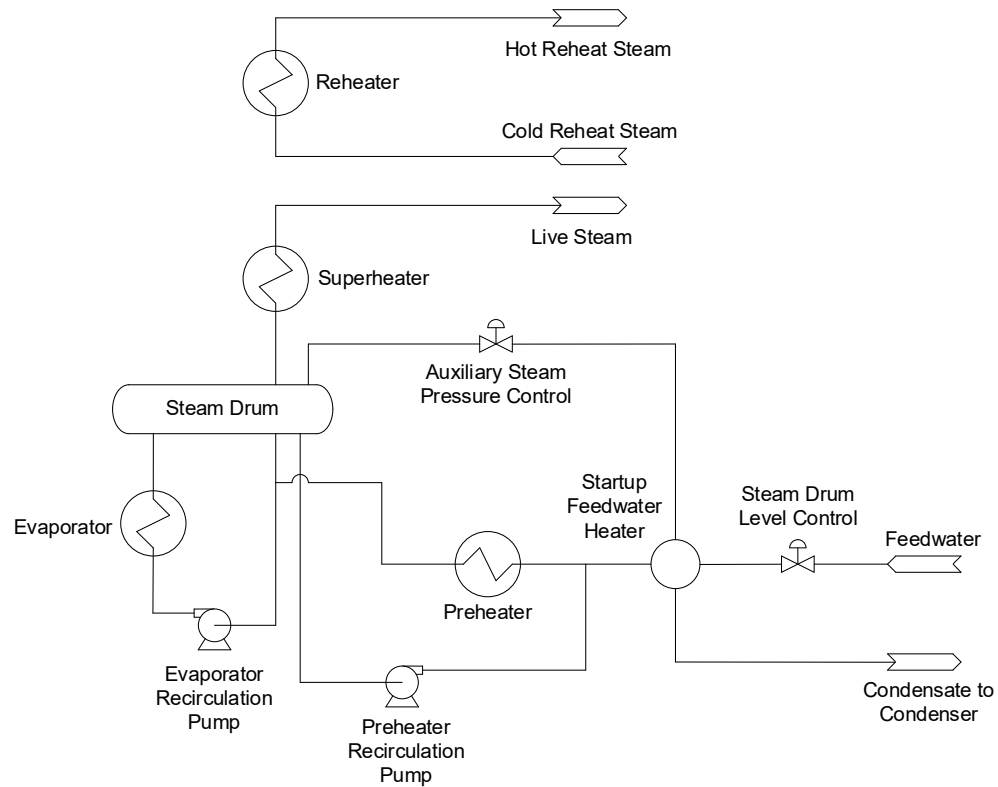


Figure 12-2 Simplified Process Flow Diagram for Water/Steam Side of Steam Generator

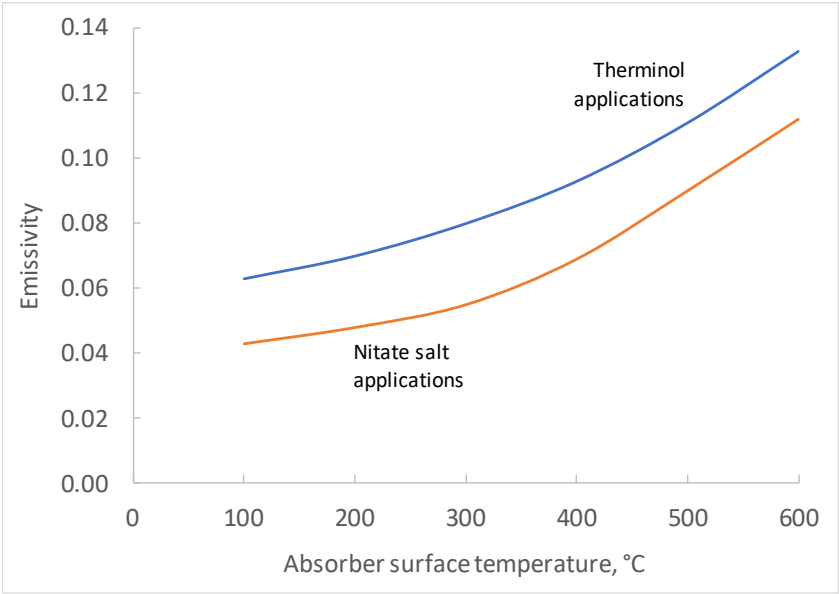


Figure 12-3 Emissivity as a Function of Absorber Surface Temperature

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12.3.2 Impedance Heating of Receiver Tubes

As discussed in Section 10.9.2, the receiver tubes are preheated using portable impedance heating systems. Since the heat losses from a receiver tube are a function of the status of the receiver tube (intact vacuum, lost vacuum, or broken glass envelope), the power required by the impedance heating system is also a function of the status of the receiver tube. Using the unit heat loss estimates shown in Figure 11-8, at a wind speed of 10 miles per hour, the required unit impedance heating capacities are listed in Table 12-1.

Table 12-1 Impedance Heating Requirements as a Function of Receiver Status

Receiver Status	Impedance Duty, W/m	Impedance Current, Amp
New, with vacuum intact	170	440
Air in the annular space	525	770
Broken glass envelope	1,900	1,470

Due to the large electric currents required for an impedance system, the sizes of the transformers, the switchgear, and the power cables are also large. This, in turn, leads to expensive heating systems. The minimum required unit heating duty is on the order of 200 W/m, which is based on the receiver tubes in an as-new condition. However, once in commercial service, the thermal efficiency of the receiver tubes will start to decay, largely as the result of gases entering the annular space between the tubes and the glass envelope. The gases increase the convection heat transfer, and degrade the quality of the selective surface on the receiver tube.

Ideally, the impedance heating system would have the capacity to preheat a tube with air in the annular space; however, the associated costs may be unaffordable. The solution for preheating a receiver which has started to lose its vacuum is to install a temporary insulated cover on those tubes which have a heat loss greater than, for example, 250 to 300 W/m. This approach, in turn, requires the identification of those receiver tubes which have a heat loss greater than 250 W/m. This can be performed by a drone using an infrared camera. The camera can identify those receiver tubes which have outer glass temperatures higher than design values.

For receiver tubes with broken glass envelopes, it is impractical to overcome the heat losses using an impedance system. To preheat these tubes, a temporary insulated cover must be used.

12.3.3 Field Designs using Therminol and Nitrate Salt

In a commercial project, there are several km of receiver tubes in the collector assemblies, and several km of piping in the field supply and return headers. Further, the collector assemblies and the field

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piping are all at nominally the same elevation. As such, once the solar field is filled with the heat transfer fluid, it becomes impractical to drain the solar field.

The principal differences between collector fields using Therminol and nitrate salt as the working fluid include the following:

- Flex hose assemblies (Section 10.9.1) replace the ball joint assemblies to accommodate 1) the thermal expansion of the receiver tubes, and 2) the relative rotation between the collector structure and the fixed field piping.
- An impedance heating system (Section 10.9.2) is used to preheat the receiver tubes prior to filling with salt, and to recover from a situation in which the salt freezes in the receiver tubes
- The field piping uses resistance heat tracing (Section 10.9.3) to preheat the piping prior to filling with salt, for freeze protection under conditions in which the salt is not circulating, and to recover from a freeze event.

Maintenance of Therminol Loops

For maintenance of a Therminol collector loop, the following steps are taken:

- The isolation valves at the inlet and at the outlet of the loop are closed
- A maintenance truck, with a vacuum tank, is located next to the loop
- A temporary piping connection is made between the outlet of the loop and the vacuum tank
- A vent valve at the inlet of the loop is opened, and air entering the vent valve pushes Therminol into the vacuum tank.

To refill the loop, the process is as follows:

- The vacuum tank is vented to the atmosphere
- A temporary piping connection is made between the inlet of the loop and a Therminol pump located below the vacuum tank
- A temporary piping connection is made between the outlet of the loop and the top of the vacuum tank

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- Therminol is circulated from the tank through the loop until all of the air has been removed from the loop.

Maintenance of Nitrate Salt Loops

Draining and refilling a collector loop using nitrate salt is performed in a manner similar to that for a collector loop using Therminol. However, with nitrate salt, the loop will be subject to freezing once the isolation valves are closed. As such, the portable impedance heating system will need to be in place prior to draining the loop. Also, the vacuum tank will need to have some form of electric heating to maintain the salt at temperatures above the freezing point.

12.3.4 Freeze Protection

During all periods in which the collector field is not in normal operation, freeze protection for the collector field (receiver tubes, flex hose assemblies, and field piping) is provided by circulating cold salt from the cold tank, through the collector field, and back to the cold tank. The inventory temperature in the cold tank is maintained above the minimum set point (275 °C) by electric salt heaters. A representative commercial 10 MWe heater²³ is shown in Figure 12-4.



Figure 12-4 10 MWe Electric Salt Heater

²³ <https://www.chomalox.com/-/media/files/literature/en-us/lit-pq132-directconnect.pdf?la=en>

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The heater consists of a series of tubes inserted into a shell, similar to a shell-and-tube heat exchanger. Inside each tube is an electric heating element, and the space between the element and the inside of the tube is filled with a ceramic packing. Heat transfer from the element to the tube is by conduction. The tubes are welded to a tubesheet. The salt flows on the shell side of the heat exchanger; as such, the salt does not come into contact with the heating elements.

The electric heaters, which operate at 4.16 kV, are supplied by the plant medium voltage distribution system or by the standby Diesel generator.

The capacity of the electric heaters is based on a zero thermal input from the field piping heat trace system or from the impedance heating systems. A 10 percent factor is added for thermal losses associated with lost vacuum and broken glass envelopes on the receiver tubes.

12.3.5 Field Piping Heat Trace Diagnostics

Regarding the heat tracing of the field piping, a commercial project will have several kilometers of salt piping. If a heat trace cable fails, then the DCS is able to locate the failed cable, under one of the following conditions:

- If the supply current drops to zero, then the cable has developed an open circuit
- If the cable develops a short circuit, then the breaker protecting the circuit will open
- If the duty cycles of the cables in Zone 1 are consistently higher than the duty cycles of the cables in a similar Zone 2, then one of the cables in Zone 1 may have developed an open circuit.

A potentially larger problem exists if salt circulation is stopped for some period of time, and a local defect in the insulation causes the salt to freeze. Identifying the location of the frozen zone may be difficult, as the heat trace system is still working perfectly. One approach is to install a fiber optic cable within the pipe insulation. The refraction properties of the glass are a function of the optic cable temperature. Measuring the transit time of a light pulse from a source to the low temperature zone, and then back from the low temperature zone to the source, can be used to indicate the distance to the low temperature zone.

12.4 Thermal Storage System

The thermal storage system in a trough project using salt as the working fluid is essentially identical to the thermal storage system in a central receiver project using salt as the working fluid.

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The design of the storage system is discussed in Section 6.10 and Section 7.3 of Volume 2 - Central Receiver Specifications.

12.5 Steam Generation System

The steam generation system in a trough project using salt as the working fluid is essentially identical to the steam generation system in a central receiver project using salt as the working fluid.

The design of the steam generation system is discussed in Section 6.11 and Section 7.4 of Volume 2 - Central Receiver Specifications.

12.6 General Salt System Equipment

The general salt system equipment includes the piping, flanges, fittings, supports, anchors, guides, and valves.

The general salt equipment in a trough project using salt as the working fluid is essentially identical to the general salt equipment in a central receiver project using salt as the working fluid.

The equipment is described in Section 6.5 of Volume 2 - Central Receiver Specifications.

12.7 Instruments

The salt instruments in a trough project using salt as the working fluid are essentially identical to the salt instruments in a central receiver project using salt as the working fluid.

The instrument concepts are discussed in Section 6.6 of Volume 2 - Central Receiver Specifications.

12.8 Heat Tracing System

The heat tracing system in a trough project using salt as the working fluid is essentially identical to the heat tracing system in a central receiver project using salt as the working fluid.

The design of the heat tracing system is discussed in Section 6.7 of Volume 2 - Central Receiver Specifications.

12.9 Salt Pumps

The salt pumps in a trough project using salt as the working fluid are essentially identical to the salt pumps in a central receiver project using salt as the working fluid.

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The design of the salt pumps is discussed in Section 6.8 and Section 7.2 of Volume 2 - Central Receiver Specifications.

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Appendix A. Nitrate Salt Properties

The nitrate salt, which is a mixture of sodium nitrate and potassium nitrate, acts as the receiver coolant, the thermal storage medium, and the heat transport fluid in the steam generator. The salt has several thermophysical properties which make it suitable as a heat transport fluid and storage medium, including:

- High densities, in the range of 1,700 to 1,900 kg/m³
- Acceptable thermal conductivities, in the range of 0.50 to 0.56 W/m-°C
- Acceptable specific heats, in the range of 1.50 to 1.55 kJ/kg-°C
- Low absolute viscosities, in the range of 0.0010 to 0.0036 kg-m/sec
- Very low vapor pressures, on the order of several Pascals
- Low corrosion rates for carbon steels at temperatures up to 400 °C, and low corrosion rates with stainless steels at temperatures up to 600 °C.

The largest difficulty with nitrate salt is a freezing point of approximately 230 °C.

The freezing point of the salt mixture, together with its corrosion characteristics, effectively define an operating temperature range of 250 °C to 600 °C. To provide a safety margin on the freezing point, a lower temperature limit of approximately 275 °C is often used. Together with the characteristics of a subcritical Rankine cycle, the following design parameters are considered representative of a commercial project: 295 °C cold salt tank temperature; 125 bar live steam pressure; 540 °C live steam temperature; 540 °C hot reheat steam temperature; and 565 °C hot salt tank temperature.

The nitrate salt is a mixture of 44 mole percent sodium nitrate (NaNO₃), and 56 mole percent potassium nitrate (KNO₃), which is equivalent to 60 weight percent NaNO₃, and 40 weight percent KNO₃.

Note: The mixture is not the eutectic. The eutectic is a mixture of 50 mole percent NaNO₃, and 50 mole percent KNO₃. For solar applications, the fraction of NaNO₃ is increased to reduce the cost of the salt. Increasing the NaNO₃ fraction raises the melting point from 222 °C for the eutectic to a nominal 238 °C. However, the increase in the melting point can be safely accommodated through careful design of the steam generator.

Temperature range The salt mixture can be used over a temperature range of 260 °C to approximately 621 °C.

Freezing point As temperature decreases, the mixture starts to crystallize at 238 °C, and is completely solid at 221 °C.

Isotropic compressibility (NaNO₃) at the melting point $2 * 10^{-10} \text{ m}^2 / \text{N}$.

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Heat of fusion (based on the average of heat of fusion of each component) $h_{sl} = 161 \text{ kJ/kg}$

Change in density upon melting $\Delta V / V_{\text{solid}} = 4.6\% \Rightarrow V_{\text{liquid}} = 1.046 V_{\text{solid}}$

A list of fluid properties, over a range of temperatures, is shown in Table A-1.

Table A-1 Nitrate Salt Properties Over a Range of Temperatures

Temperature, C	Density, kg/m ³	Specific heat, kJ/kg-°C	Absolute viscosity, kg/m ² -sec	Thermal conductivity, W/m-°C
220	1,950	1.481	0.00578	0.485
240	1,937	1.484	0.00501	0.489
260	1,925	1.488	0.00434	0.492
280	1,912	1.491	0.00376	0.496
300	1,899	1.495	0.00326	0.500
320	1,886	1.498	0.00284	0.504
340	1,874	1.501	0.00249	0.508
360	1,861	1.505	0.00220	0.511
380	1,848	1.508	0.00196	0.515
400	1,836	1.512	0.00178	0.519
420	1,823	1.515	0.00163	0.523
440	1,810	1.519	0.00152	0.527
460	1,797	1.522	0.00143	0.530
480	1,785	1.526	0.00137	0.534
500	1,772	1.529	0.00131	0.538
520	1,759	1.532	0.00127	0.542
540	1,747	1.536	0.00122	0.546
560	1,734	1.539	0.00116	0.549
580	1,721	1.543	0.00109	0.553
600	1,708	1.546	0.00099	0.557

The fluid properties of nitrate salt, each as functions of temperature between 250 °C and 600 °C, are described below. The properties are nominally independent of pressure.

Density, as a function of temperature:

$$\rho (\text{lb}_m / \text{ft}^3) = 131.2 - 0.02221 * T (^\circ\text{F})$$

$$\rho (\text{kg} / \text{m}^3) = 2090 - 0.636 * T (^\circ\text{C})$$

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Specific heat, as a function of temperature:

$$c_p \text{ (Btu / lb}_m\text{- }^\circ\text{F)} = 0.345 + (2.28 \times 10^{-5}) * T \text{ (}^\circ\text{F)}$$

$$c_p \text{ (J / kg - }^\circ\text{C)} = 1443 + 0.172 * T \text{ (}^\circ\text{C)}$$

Absolute viscosity, as a function of temperature:

$$\mu \text{ (lb}_m\text{ / ft - hr)} = 60.28440 - 0.17236 * T \text{ (}^\circ\text{F)} + (1.76176 \times 10^{-4}) * (T \text{ (}^\circ\text{F)})^2 - (6.11408 \times 10^{-8}) * (T \text{ (}^\circ\text{F)})^3$$

$$\mu \text{ (mPa - sec)} = 22.714 - 0.120 * T \text{ (}^\circ\text{C)} + (2.281 \times 10^{-4}) * (T \text{ (}^\circ\text{C)})^2 - (1.474 \times 10^{-7}) * (T \text{ (}^\circ\text{C)})^3$$

Thermal conductivity, as a function of temperature:

$$k \text{ (Btu / hr - ft - }^\circ\text{F)} = 0.253208 + 6.26984 * 10^{-5} * T \text{ (}^\circ\text{F)}$$

$$k \text{ (W / m - }^\circ\text{C)} = 0.443 + 1.9 * 10^{-4} * T \text{ (}^\circ\text{C)}$$

Properties of solid salt are as follows:

Density, ρ

NaNO ₃	2,260 kg / m ³ at ambient temperature
KNO ₃	2,190 kg / m ³ at ambient temperature

Heat capacity, c_p

NaNO ₃	37.0 cal / °C - mol = 1,820 J / kg - °C near the melting point
KNO ₃	28.0 cal / °C - mol = 1,160 J / kg - °C near the melting point

Thermal conductivity, k

KNO ₃	2.1 W / m - °C

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1. Introduction, Report Format, Acknowledgements, and Disclaimer

1.1 Introduction

The work proposed concentrates on a Design Basis Documents / Owner's Technical Specification for parabolic trough and central receiver power plants using nitrate salt as the heat transport fluid and the thermal storage medium. The goals are to 1) distill the successful experience with nitrate salt systems from as many commercial projects as possible, 2) provide technical bases for equipment design/selection that an owner can impose on an EPC contractor, 3) compile this information in one location to provide guidance on salt systems that is as broadly applicable to as many projects as possible, and 4) work toward an industry consensus on a Design Basis Document.

1.2 Project Goal

The principal goal of the project is to develop a Design Basis Document for nitrate salt systems in CSP projects that reflects the lessons learned from earlier commercial projects. The document allows an owner to provide design guidance, and to impose a minimum set of requirements, above and beyond those in the normal Codes and Standards, on an EPC contractor in an effort to avoid the repeat of past mistakes. If successful, this would allow salt systems to achieve the same levels of reliability and availability as Therminol systems in commercial parabolic trough plants.

1.3 Report Format

The report consists of 3 volumes:

- Volume 1 - Specifications for Parabolic Trough Projects
- Volume 2 - Specifications for Central Receiver Projects
- Volume 3 - Narrative

Volumes 1 and 2 include discussions of the following topics:

- Plant functional requirements for the salt systems
- Plant operating states, and transitions between states, for the salt systems
- Risk analysis of the principal salt components
- Plant requirements for the salt systems to meet the functional and the operating requirements

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- Type of specification for the principal salt components: functional; or prescriptive
- Current state of the art for salt systems.

Volume 3 discusses a number of cases in which the salt equipment at commercial parabolic trough and central receiver projects has not met the projected levels of reliability and availability. Possible reasons for the sources of the problems are discussed, as is a range of possible alternate designs that could avoid the known problems.

1.4 Acknowledgements

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1.5 Disclaimer

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

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2. Functional Requirements

2.1 *Commercial Projects*

Central receiver projects use a field of two-axis tracking mirrors (heliostats) to redirect and focus sunlight on a heat exchanger (receiver) located at the top of a tower. The solar flux is converted into heat on the absorber surface, and the heat is transferred to a nitrate salt working fluid. The high temperature salt is delivered to the hot tank of a thermal storage system. The hot salt is pumped to a steam generation system, which converts the thermal energy in the salt to flows of live steam and reheat steam. The energies in the steam flows are converted to electric power in a conventional Rankine cycle.

The thermal rating of the heliostat field and the receiver is greater than the thermal demand of the steam generator. The difference in thermal ratings is retained in the storage system for use during cloudy weather or at night.

2.2 *Nitrate Salt Systems*

The nitrate salt systems include the receiver, the thermal storage, and the steam generation systems.

The following systems are also discussed to provide a picture of the complete plant: collector; electric heat tracing; electric power generation; balance of plant; and master control.

2.2.1 **Collector System**

The collector system consists primarily of heliostats, which redirect and focus sunlight on the receiver. The major system elements include two-axis tracking heliostats, individual heliostat controllers, heliostat array controller, and communication links between the heliostat controllers and heliostat array controller. The number of heliostats varies with the heliostat design, required receiver duty, and project site.

Each heliostat consists of a below-ground foundation, an above-ground pedestal, two-axis drive, support structure, mirrors, and position sensors. The heliostat controller is a small microprocessor located on the heliostat pedestal. Electric wiring for power, control, and grounding are also provided.

The heliostat array controller, resident in the distributed control system, maintains supervisory control over the heliostat field. The array controller includes sun positioning algorithms, X-Y coordinates for each heliostat, receiver coordinates, software for the static aim point processing system, a dynamic aim point processing system, and a beam characterization system. The operator interface is provided through the distributed control system.

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The communication link is a redundant network, typically either copper wire or fiber optic cable, between the heliostat array controller and each of the heliostat controllers. However, a number of projects are starting to use wireless communication between the heliostat array controller and each of the heliostat controllers. The array controllers are typically powered by local photovoltaic panels to eliminate the need for both electric power and control wiring to the heliostat.

2.2.2 Receiver System

The receiver system converts the redirected solar energy from the collector system into thermal energy. The system includes the receiver, a support tower, and receiver circulation pumps. Details on the design and the operation of the receiver system are presented in Section 7.1.

The receiver consists of a group of absorber panels, support structures, inter-panel piping, inlet vessel, outlet vessel, thermal insulation, electric heating, instruments, valves, and controls. Each panel consists of a group of parallel tubes, which are connected at the bottom by a lower header and at the top by an upper header. Each panel includes a support structure, ovens surrounding the headers, thermal insulation, and instruments. The panels are attached to the primary support structure, which positions the panels to approximate a vertical cylinder. Salt enters the first panel at a temperature of 295 °C, then flows through several panels in series to reach the nominal outlet temperature of 565 °C.

The receiver is installed at the top of a reinforced concrete tower. Salt is transported from grade through a riser pipe to the receiver and is returned to grade through a downcomer pipe from the receiver.

The receiver circulation pump with a vertical turbine design is installed on a support structure above the cold salt tank. The pump draws suction from a point near the bottom of the tank.

2.2.3 Thermal Storage System

The storage system stores high temperature salt from the receiver for use by the steam generator, and returns low temperature salt from the steam generator for use by the receiver. The components of a storage system include cold salt tank, hot salt tank, tank foundations, nitrate salt inventory, electric recirculation heaters, and tank insulation. Details on the design and the operation of the storage system are presented in Section 7.3.

2.2.4 Steam Generation System

The steam generation system uses thermal energy in the salt inventory to produce main steam and reheat steam at the conditions required by the turbine-generator. The steam generation system includes the following heat exchangers: superheater; reheater; evaporator; and preheater. Additional components include the steam drum, the water recirculation pumps for the evaporator, and the water recirculation

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pumps for the preheater. Details on the design and the operation of the steam generation system are presented in Section 7.4.

As mentioned, hot salt is supplied to the steam generator from the hot salt tank by means of a vertical turbine pump. The circulation pump is installed on a support structure above the hot salt tank, and draws suction from a point near the bottom of the tank.

Cold salt is also supplied to the steam generator from the cold salt tank by means of a vertical turbine pump. The cold salt is used for (1) attemperation of the hot salt during startup and shut down of the steam generator, and (2) maintaining the temperature of the steam generator at 295 °C during overnight hold periods. The attemperation pump is installed on a structure above the cold salt tank, and draws suction from a point near the bottom of the tank.

2.2.5 Electric Heat Tracing System

The heat tracing system prevents salt from freezing in all process equipment, provides thermal conditioning to the equipment during plant startup, and protects the equipment from excessive thermal gradients and the associated thermal stresses. Details on the design and the operation of the heat tracing system are presented in Section 6.9. This system consists of the following major components:

- Mineral insulated heat trace cables
- Installation hardware, cold leads, and termination kits
- Temperature elements (either thermocouples or resistance temperature detectors)
- Temperature signal conditioning instruments and transmitters
- Power conditioning equipment, including solid-state contactors.

The system control is managed through programmable logic controllers and the distributed control system.

Heat tracing is required on salt heat exchangers, piping, instruments, valves, vents, drains, and pressure relief valves.

In addition to the use of mineral insulated cable heaters, various plant components require other forms of electric heating. Examples include recirculation heaters for the salt storage tanks and radiant heaters for the receiver ovens.

Thermal insulation is part of the system on which it is installed. However, the design and installation details are an integral part of the electric heating system. Therefore, the insulation design must be developed as part of a coordinated package.

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2.2.6 Electric Power Generation System

The electric power generation system converts the energy in the main steam and the reheat steam to electric power for delivery to the grid. This system consists of a turbine-generator, air cooled condenser, condensate system, deaerator, feedwater system, water and steam sampling system, turbine lubrication oil system, and associated pumps and rotating equipment.

2.2.7 Balance of Plant Systems

The balance of plant equipment supports all other plant systems, and includes:

- Switch yard/main power distribution system, including main and auxiliary power transformers
- Emergency and uninterruptible power supplies
- Cranes providing access to receiver panels, salt pumps, and heat exchanger tube bundles
- Fire protection and detection systems
- Plant security system
- Compressed air system
- Potable water system
- Cooling water system
- Service water system
- Nitrogen supply system
- Water treatment system
- Deionized water system
- Sanitary waste and industrial waste systems
- Oil/water separator

The balance of plant includes the electric distribution system supplying the motor control centers, grounding, lightning protection, and lighting with associated raceway, conduit, and wire.

The balance of plant also includes site civil work (grading, drainage, and fences), buildings, and the bridge structures over the salt storage tanks. The beam characterization system target is also included in the balance of plant.

2.2.8 Master Control System

The master control system 1) monitors and controls all process functions during the plant operating states and transitions between operating states, and 2) responds to operator commands. The control system is comprised of the following major subsystems:

- Heliostat array controller, responsible for the operation of the collector field

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- Distributed control system, responsible for all process control other than that of the collector field
- Administrative and data analysis system, which ties all of the plant systems into a common database.

The distributed control system consists of the operator consoles, redundant servers, network server, redundant programmable logic controllers, logic controller remote input/output devices, data historian, network communications, peripherals, and software

The heliostat array controller consists of redundant processors, a data historian, beam characterization system software, heliostat aim point processing software, special instruments, associated network ties, and peripherals. The controller interface is managed through the distributed control system. The individual heliostat controllers are not part of the array controller.

The administrative and data analysis systems are linked to the distributed control system and to the heliostat array controller, with read only access. The analysis systems consist of a management information system, material control and maintenance system, administrative systems, peripherals, and software.

2.3 *Features of Commercial Projects*

Representative features of commercial projects, including site characteristics, labor availability, land, permits, solar radiation, access to transmission, wind, seismic conditions, range of plant sizes, range of storage capacities, water, soil properties, are briefly discussed below.

2.3.1 Site Characteristics

The optimum site would have the following characteristics:

- A contiguous section of land, with a shape and an overall area which allows the collector field to be arranged for the best optical performance. Nonetheless, it can be noted that a commercial project in Nevada uses non-contiguous areas for the heliostat field, and the receiver is designed to accommodate non-symmetrical flux distributions
- A nominally flat surface, with a minimum of elevation changes, to reduce the amount of grading for drainage

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- A location far enough from adjacent mountain ranges that could 1) block solar radiation early in the morning and late in the afternoon, 2) channel winds onto the site, or 3) produce localized cloud buildups that affect solar resources at the site.

2.3.2 Availability of Skilled Labor

Most commercial projects are in desert locations, to provide annual solar radiation levels that are as high as possible. As might be expected, desert areas often have low population densities, with long distances to large cities.

Commercial projects require a skilled labor force in terms of operators, maintenance personnel, and technicians skilled in computers, instruments, mechanics, and welders.

This combination of requirements often presents a problem for commercial projects. The limited population near the project may not have the skills required to properly operate and to maintain the plant. Clearly, incentives can be offered to skilled personnel to relocate to areas near the project, but this will significantly increase the annual staffing cost. Further, there is no guarantee that skilled staff members, once attracted to the project will remain at the project.

It can be noted that this is not a hypothetical problem; it is systemic to the solar industry. In addition, the largest determinants to the annual availability of a project are 1) the technical knowledge of the Owner, 2) the experience of the engineering company, and 3) the skills and the discipline of the staff to operate the equipment within the limits set by the equipment suppliers.

2.3.3 Land

Securing property for the project, either by purchase or by lease, is highly site specific. The ideal situation is one in which all of the land is under the control of one owner, the site has already been disturbed for a commercial purpose, and there are no competing interests for either the land or the groundwater beneath the site.

2.3.4 Permits

All commercial projects will require land use and construction permits.

As might be expected, the duration and the cost of the permit process will depend on the requirements of the permitting agencies having jurisdiction. The minimum duration is perhaps 1 year, and durations up to 3 years have been experienced.

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Items which could postpone, or deny, a project from receiving a permit include 1) visual restrictions due to locations which are close to cities or to airports, 2) limitations on groundwater use, 3) archeological sites or artifacts, and 4) endangered species.

Projects using nitrate salt as the working fluid will generate oxide of nitrogen emissions associated with the thermal decomposition of magnesium nitrate in the salt. However, this is one-time emission during plant commissioning, and there are effective methods for limiting the cumulative release of NO_x.

2.3.5 Solar Radiation

To a first order, the levelized cost of energy is inversely proportional to the annual direct normal solar radiation. The minimum practical annual radiation for a commercial project is in the range of 1,700 to 1,800 kWh/m²-year. Many desert sites in the US and in North Africa have annual radiations in the range of 2,500 to 2,700 kWh/m²-year, while the best sites in the world (Atacama desert in Chile) have annual values on the order of 3,500 kWh/m²-year.

2.3.6 Access to Electric Power Transmission

The site must have access to electric power transmission, both for export to support energy sales and for import to support auxiliary power demands when the plant is not in operation. Feasible distances from the site to major transmission lines range from 1 mile to perhaps 10 miles.

In the United States, the utility connection process and the time required for transmission upgrades can exceed 5 years.

2.3.7 Wind

The heliostats must be designed to survive the highest wind speed expected at the site. A common design wind speed, which is applicable at many commercial sites, is 40 m/sec (90 mph). Nonetheless, collector structures can be designed for essentially any site conditions.

It can be noted that some commercial projects have suffered damage to collector structures due to brief wind gusts at speeds higher than expected. Two options are available to help avoid this condition: 1) install collectors with a stronger structure near the perimeter of the field, or 2) provide wind fences at the perimeter of the field.

Sites have both a peak wind speed and an average wind speed. The optical accuracy of a heliostat is typically guaranteed wind speed; a representative value is 7 to 8 m/sec. For wind speeds above the guarantee value, the heliostat continues to track, but with a reduced optical accuracy and a reduced flux on the receiver. When the wind speed reaches a maximum operating velocity of 11 to 12 m/sec, the heliostats are moved from the track position to the stow position. This protects the structures from

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potentially damaging wind gusts during the somewhat protracted period (2 to 3 minutes) required to stow the field. As such, sites with a high percentage of operating hours with wind speeds below the guarantee value will offer a better solar-to-electric efficiency than sites which operate with a high fraction of wind speeds above the guarantee, but still below the stow, value.

2.3.8 Seismic Conditions

The design of the heliostat is governed by wind loads rather than seismic loads. Unless the site is within a few kilometers of significant fault, the design seismic conditions will not influence the choice of the plant location.

Seismic accelerations will produce sloshing movements in the inventory of the salt tanks. To protect the welded connection between the top of the wall and the bottom of the roof from excessive loads, freeboard allowances are provided, as specified in the API tank design codes.

2.3.9 Range of Plant Sizes

The operation and maintenance staff requirements in commercial projects are relatively insensitive to the size of the project; i.e., the staff required to operate a 100 MWe project is not much larger than the staff required to operate a 30 MWe project. As such, the larger the project, the lower the unit operation and maintenance cost (\$/kWh).

The majority of commercial projects built to date have a gross plant rating of 110 to 135 MWe. In principle, plant ratings greater than 135 MWe are possible. However, as noted below, thermal storage is a necessary feature of commercial plants. With representative solar multiples in the range of 1.6 to 1.8, the corresponding receiver ratings are in the range of 400 to 600 MWt.

With a receiver rating of 600 MWt, the most distant heliostats are 1.5 to 1.6 kilometers from the tower. At these distances, the receiver spillage losses become significant, and the marginal revenue produced by a distant heliostat approaches the marginal cost of the heliostat. This, in turn, effectively sets a practical limit on the size of the receiver (600 MWt), which, in turn, sets a practical limit on the gross plant rating (160 MWe with a solar multiple of 1.6; 140 MWe with a solar multiple of 1.8)

2.3.10 Range of Storage Capacities

Although some commercial tower projects were built with steam receivers (i.e., no thermal storage), central receiver technology can no longer compete on an economic basis with photovoltaic plants for daytime generation. As a result, central receiver plants constructed during the past decade have included storage capacities sufficient to operate the Rankine cycle, at full load, for periods of 6 to 12 hours.

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With thermal storage, solar fields are sized to operate the Rankine cycle at full load during the day plus provide additional energy to charge the storage system so that the plants can continue operating after sunset.

2.3.11 Water

Essentially all recent commercial projects, and all future projects will, use dry cooling for heat rejection from the Rankine cycle. Dry cooling reduces water consumption by approximately 90 percent relative to a plant using wet cooling. The reduced water consumption, in turn, simplifies the permitting process and makes available a wider range of potential site locations. As an example, a commercial solar project in the Atacama desert in Chile has no access to municipal or ground water. All of the water used at the site is brought in by truck from the coast. Such an arrangement would be impossible were the plant to have used wet cooling.

For a plant with dry cooling, the principal water demands include 1) distilled water for mirror washing, 2) makeup water to replace blowdown losses from the Rankine cycle, and 3) makeup water to replace blowdown losses from the wet surface air cooler in the closed cooling water system.

2.3.12 Soil Properties

The properties of the soil influence the plant design in the following areas:

- Heliostat foundations. The foundations for the heliostats can consist of concrete piles, steel piles, or concrete footings (small heliostats). The selected approach is based on the strength of the soil, the site design wind speed, and the presence, or the absence, of rocks in the soil.
- Thermal storage tank foundations. Essentially all commercial salt tanks use a concrete base mat underneath the floor insulation. This is an economical design, but it requires the weight of the static head of the salt to be less than the allowable bearing load for the soil. A representative soil bearing load is 240 kPa (5,000 lb_f/ft²), which, in turn, limits the maximum salt depth to a nominal value of 12 m. If the allowable soil bearing load is less than 240 kPa, or if the salt tank is designed with a inventory height greater than 12 m, friction piles can be installed below the concrete base mat to increase the bearing capacity of the soil. However, the addition of friction piles will significantly increase the cost of the foundation.
- Water table. The elevation of the water table should be significantly lower than the elevation of the concrete base mat for the storage tanks. A low water table helps to prevent liquefaction of the soil during an earthquake, and prevents changes in the vertical heat flux from the tank foundation into the soil.

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- Percolation. Following a rain, the water would ideally percolate into the soil in less than one day. This reduces quantities of standing water, which could otherwise make access to the collector field problematic for maintenance vehicles.

2.4 Nitrate Salt Characteristics

The nitrate salt, which is a mixture of sodium nitrate and potassium nitrate, acts as the receiver coolant, the thermal storage medium, and the heat transport fluid for the steam generator. The salt has several thermophysical properties which make it suitable as a heat transport fluid and storage medium, including:

- High densities, in the range of 1,700 to 1,900 kg/m³
- Acceptable thermal conductivities, in the range of 0.50 to 0.56 W/m-°C
- Acceptable specific heats, in the range of 1.50 to 1.55 kJ/kg-°C
- Low absolute viscosities, in the range of 0.0010 to 0.0036 kg-m/sec
- Very low vapor pressures, on the order of several Pascals
- Low corrosion rates for carbon steels at temperatures up to 400 °C, and low corrosion rates for stainless steels at temperatures up to 600 °C.

The most challenging property of the nitrate salt is its relatively high freezing point of approximately 230 °C. The freezing point, along with its corrosion characteristics at high temperatures, effectively defines an operating temperature range of 250 °C to 600 °C. To provide a safety margin from freezing, a lower process design temperature limit of approximately 290 °C is often used. Together with the characteristics of a subcritical Rankine cycle, the following design parameters are considered typical for a commercial project: 295 °C cold salt tank temperature; 125 bar main steam pressure; 540 °C main steam temperature; 540 °C reheat steam temperature; and 565 °C hot salt tank temperature.

Figure 2-1 shows the phase diagram for the binary NaNO₃ and KNO₃ system. The nitrate salt used in commercial tower projects is a mixture of 64 mole percent NaNO₃ and 36 mole percent KNO₃, which is equivalent to 60 weight percent NaNO₃ and 40 weight percent KNO₃.

Note that the mixture is not at the eutectic composition of 50 mole percent NaNO₃ and 50 mole percent KNO₃. For solar applications, the fraction of NaNO₃ is increased to 60 weight percent to reduce the cost of the salt. Increasing the NaNO₃ fraction has the following effects:

- The liquidus temperature increases from 220 °C (at the eutectic composition) to a nominal 238 °C

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- Starting from 238 °C, cooling the salt results in a mixture of solid particles suspended in a liquid. As the temperature decreases, the fraction of solid particles increases, until the entire mixture becomes frozen at a temperature of 220 °C.

The increase in the liquidus temperature to 238 °C can be accommodated by careful selection of the system operating procedures and by the design of the plant components.

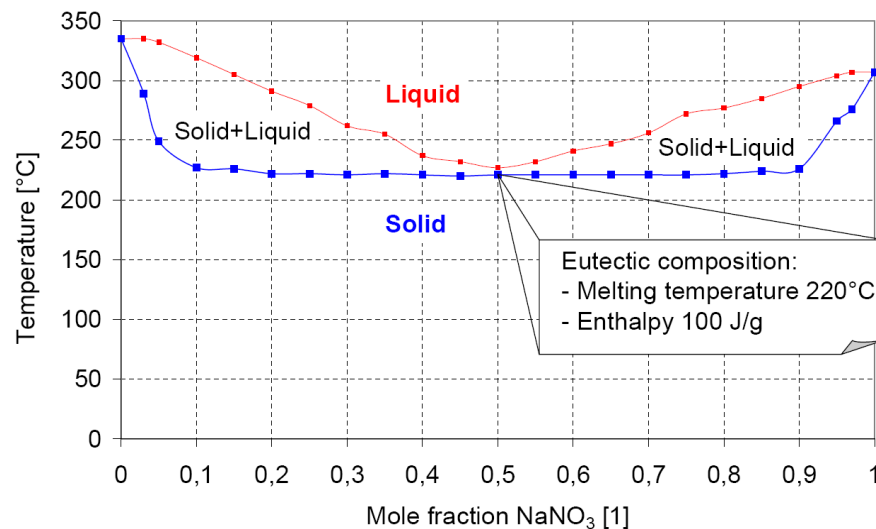


Figure 2-1 Phase Diagram of NaNO₃ - KNO₃ Mixtures

A full range of thermophysical properties for the salt mixture are presented in Appendix A: Nitrate Salt Properties.

Options for the salt procurement specification, including a discussion of purities, corrosive elements, and supply options are outlined in Section 6.3.

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3. Operating States, and Transitions Between Operating States

3.1 Introduction

For the purposes of defining the plant operating states, and the transitions between operating states, a central receiver project can be divided into the following sections: 1) energy collection section, consisting of the collector system and the receiver system; and 2) energy conversion section, consisting of the steam generator system and the electric power generation system.

3.2 Energy Collection Section

3.2.1 States for the Energy Collection Section

The energy collection section operates in one of the following five states (Refer to Figure 3-1, Table 3-1, and Table 3-2):

- Long Term Hold / Overnight Hold: The heliostats are in the stow position, the receiver is drained, and the electric heat trace circuits are inactive
- Standby: The heliostats are focused on the standby aim points, and the receiver pump is in operation. Cold salt is flowing up the riser, through the upper bypass line, down the downcomer, and returning to the cold tank
- Preheat: The receiver electric heat trace circuits are active, the preheat heliostats are focused on the receiver, and the receiver pump is in operation. Cold salt is flowing up the riser, through the upper bypass line, down the downcomer, and returning to the cold tank
- Normal Operation: All of the available heliostats are focused on the receiver, the receiver flow rate is controlled to achieve an outlet temperature of 565 °C, and the electric heat trace circuits are de-energized at the normal operation temperature set points
- Cloud Standby: All of the available heliostats are focused on the receiver, the receiver flow rate is controlled to achieve an outlet temperature of 540 °C under theoretical clear sky conditions, and the electric heat trace circuits are de-energized at the normal operation temperature set points.

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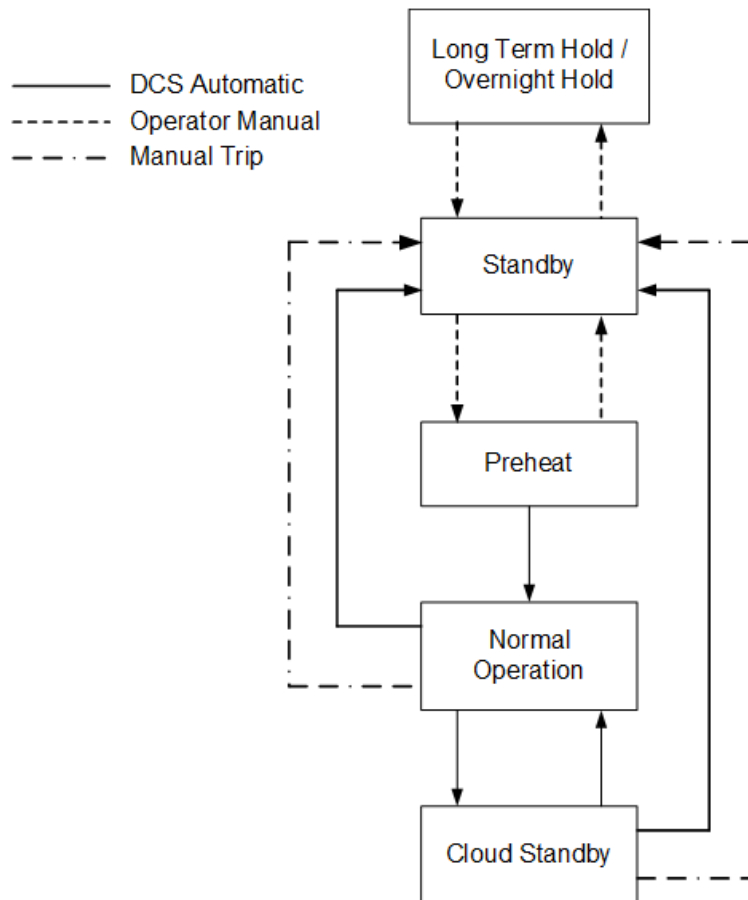


Figure 3-1 Energy Collection Section States and Transitions

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Table 3-1 Energy Collection System States and Equipment Status - Sheet 1 of 2

	Collector Field	Receiver			Riser & Downcomer	Receiver Pump
		Inlet Vessel	Panels	Outlet Vessel		
Long Term Hold / Overnight Hold	Heliostats stowed	Empty	Empty	Empty	Empty	Off
Standby	Heliostats tracking standby aim points	Level control operating at partial pressure	Empty	Empty	Filled, with flow through lower crossover	On
Preheat	Heliostats tracking preheat aim points	Level control at partial ullage pressure	Empty	Empty	Filled, with flow through lower crossover	On
Normal Operation	Heliostats tracking receiver aim points	Pressure control corresponding to clear sky conditions	Filled and design flow established	Filled under level control	Filled and design flow	On
Cloud Standby	Heliostats tracking receiver aim points	Pressure control corresponding to clear sky conditions	Receiver flow controlled to maintain 540 °C outlet temperature under theoretical clear sky conditions	Filled	Filled	On

Table 3-2 Energy Collection System States and Equipment Status - Sheet 2 of 2

	Thermal Storage		Downcomer Flow	Thermal Conditioning	
	Cold Tank	Hot Tank		Tank Heaters	Heat Tracing
Long Term Hold / Overnight Hold	Intermediate to maximum level	Minimum to intermediate level	None	Energized with controllers at Long Term Hold temperature set points	Salt wetted systems active; most systems inactive
Standby	Level variable	Level variable	Flow routed to cold tank	De-energized with controllers at normal operation temperature set points	Salt wetted systems active; flowing systems standby
Preheat	Level variable	Level variable	Flow routed to cold tank	De-energized with controllers at normal operation temperature set points	Receiver heat tracing and ovens active
Normal Operation	Draw-down to heel level	Filling to maximum level	Flow routed to hot tank	De-energized with controllers at normal operation temperature set points	Salt wetted systems active; flowing systems standby
Cloud Standby	Level variable	Level variable	Flow routed to cold tank	De-energized with controllers at normal operation temperature set points	Salt wetted systems active; flowing systems standby

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3.2.2 Transitions Between States for the Energy Collection Section

The nine transitions between the states for the energy collection section are as follows:

- **Long Term Hold to Standby:** The operator moves the heliostats from the stow positions to tracking the standby aim points. The temperatures of the riser, the upper bypass line, and the downcomer are raised to 290 °C. The receiver pump is started, and a flow is established in the riser, the bypass line, and the downcomer
- **Standby to Preheat:** The temperatures of the panel ovens and intra-receiver piping are raised to 315 °C. The preheat heliostats, pre-selected by the heliostat array controller, are moved from the standby aim points to the preheat aim points
- **Preheat to Standby:** The preheat heliostats are moved from the preheat aim points to the standby aim points
- **Preheat to Normal Operation:** The transition consists of the following steps: 1) the receiver is filled by flooding, 2) serpentine flow is established, 3) a flow rate corresponding to clear sky conditions is established, 4) the heliostats are moved from the standby (or preheat) aim points to the normal aim points, and 5) the flow rate is controlled to achieve a nominal outlet temperature of 565 °C
- **Normal Operation to Cloud Standby:** Automatic temperature control is suspended, and the flow rate is controlled to achieve an outlet temperature of 540 °C under theoretical clear sky conditions
- **Cloud Standby to Normal Operation:** Automatic temperature control is resumed, and the flow rate is controlled to achieve a nominal outlet temperature of 565 °C
- **Normal Operation to Standby:** The heliostats are moved from the normal aim points to the standby aim points, the inlet vessel is vented to the atmosphere, and the receiver panels are drained
- **Cloud Standby to Standby:** The heliostats are moved from the normal aim points to the standby aim points, the inlet vessel is vented to the atmosphere, and the receiver panels are drained
- **Standby to Long Term Hold:** The heliostats are moved from the tracking the standby aim points to the stow position, the receiver pump is stopped, the riser and the downcomer are drained, and the electric heat trace circuits are deactivated.

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3.3 *Energy Conversion Section*

3.3.1 States for the Energy Conversion Section

The energy conversion section operates in one of the following six states (Refer to Figure 3-2 and to Table 3-3):

- Long Term Hold: The steam generator is drained, the electric heat trace circuits are inactive, and the electric recirculation water heaters are inactive
- Overnight Hold: The attemperation pump supplies cold salt to the steam generator to maintain heat exchangers temperatures above 275 °C. The steam turbine is rotated by the turning gear. Depending on the requirements of the project, turbine shaft sealing steam may need to be produced to maintain condenser vacuum
- Isothermal Steam Production: The auxiliary steam generator is in service, supplying shaft sealing steam to the turbine. A vacuum is drawn in the condenser. Cold salt is supplied to the hot end of the superheater and to the hot end of the reheater. The salt flows from the cold ends of the heat exchangers combine at a hot salt mixing station upstream of the hot end of the evaporator. The mixing station is illustrated in the process flow diagram Figure 6-7 of Volume 3 - Narrative. Thermal energy in the hot salt is sufficient to establish a steam flow rate equal to 20 percent of the design steam flow rate. The saturation pressure in the evaporator is set by the turbine bypass system such that the saturation temperature is equal to the cold salt temperature. The saturated steam flow from the evaporator passes through the superheater and then through the reheater. Since the salt temperature in the superheater and in the reheater is equal to the saturation temperature, no heat transfer occurs in either heat exchanger. However, during the period in which the steam flow rate increases from 0 percent to 20 percent (slightly above the vendor's minimum), non-uniform steam distributions in the heat exchangers do not produce damaging stress distribution
- Superheated Steam Production: The salt mixing station is transferred from the startup point upstream of the evaporator to the normal point upstream of the superheater / reheater. The mixing station is illustrated in the process flow diagram Figure 6-8 of Volume 3 - Narrative. The steam production rates decays slightly to a nominal 16 percent (the vendor's minimum) due to the superheating duties supplied by the superheater and the reheater
- Turbine Synchronization: A live steam flow rate of (project specific) kg/hr, with a temperature and pressure of (project specific) °C and (project specific) bar, respectively, as specified by the turbine vendor, are established. The turbine-generator is synchronized with the grid, and a minimum turbine output of (project specific) MWe is established

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- Normal Operation: The extraction feedwater heaters are placed in service. A live steam flow rate of (project specific) kg / hr, with a temperature and pressure of (project specific) °C and (project specific) bar, respectively, are established. Turbine output is set by adjusting the speed of the hot salt pump.

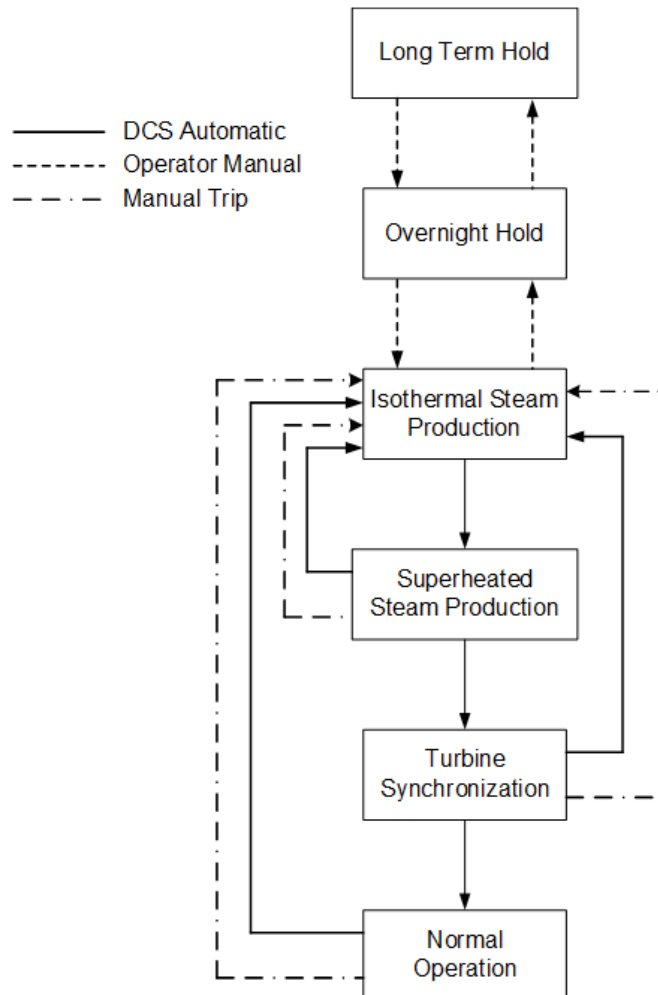


Figure 3-2 Energy Conversion Section States and Transitions

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Table 3-3 Energy Conversion Section States and Equipment Status

	Long Term Hold	Overnight Hold	Isothermal Steam Production	Superheated Steam Production	Turbine Synchronization	Normal Operation
Steam Generator						
Salt side	Empty	Filled	Filled	Filled	Filled	Filled
Water side	Empty	Filled	Filled	Filled	Filled	Filled
Startup feedwater heater	Empty	Filled	Filled	Filled	Filled	Filled
Thermal Storage						
Cold tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Hot tank	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level	Intermediate level
Salt Pumps						
Hot salt	Off	Off	On	On	On	On
Attemperation	Off	Pump maintains minimum system temperature	Salt attemperation as required	Salt attemperation as required	Salt attemperation as required	Off
Water Pumps						
Condensate	Off	Off	On	On	On	On
Feedwater	Off	Off	On	On	On	On
Steam generator recirculation	Off	On	On	On	On	On
Auxiliary Steam						
Electric boiler	Off	Off	Initial demand for turbine seals and condenser vacuum	Off	Off	Off
Steam generator	Off	Off	On	On	On	On
Turbine - Generator						
Turbine	Turning gear	Turning gear	Turning gear	Turning gear	Part load	Full load
Condenser	Empty	Nitrogen	Vacuum	Vacuum	Vacuum	Vacuum
Generator	Off	Off	Off	Off	Part load	Full load
Balance-of-Plant	As required	As required	As required	As required	On	On
Thermal Conditioning						
Tank heaters	Energized	Intermittent	Intermittent	Intermittent	Off	Off
Heat tracing	Off; all systems drained	Activated as required for preheating and protection of equipment from salt freezing	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby	Non-flowing wetted salt systems activated; flowing systems standby
Steam generator water heater	Off; water side drained	On, to preheat heat exchangers to 290 °C	Off	Off	Off	Off

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3.3.2 Transitions Between States for the Energy Conversion Section

The nine transitions between the states for the energy conversion section are as follows:

- **Long Term Hold to Overnight Hold:** The temperatures of the steam generator heat exchangers and the inter-vessel piping are raised to 275 °C by the electric water heaters and the electric heat tracing. The steam generation system attemperation pump is started, and a flow of cold salt is established through the heat exchangers
- **Overnight Hold to Isothermal Steam Production:** The electric auxiliary steam generator is started, supplying sealing steam to the turbine shaft seals. A vacuum is drawn in the condenser. The hot salt pump is started, supplying hot salt to a mixing station upstream of the evaporator. Saturated steam production begins in the evaporator, with a saturation temperature equal to the cold salt temperature. Saturated steam flows through the superheater and the reheater; however, no heat transfer occurs. This allows the steam production rate to increase from 0 percent to a nominal 20 percent without flow maldistributions on the tube sides generating potentially damaging stress distributions. Steam from the steam generator is sent through the turbine bypass system to the condenser
- **Isothermal Steam Production to Superheated Steam Production:** The salt mixing station is transferred from the startup point upstream of the evaporator to the normal point upstream of the superheater / reheater. The flow rates of cold salt and hot salt remain nominally fixed. The metal temperatures of the superheater and the reheater both increase, which results in superheating occurring in both the superheater and in the reheater. Due to the addition of superheating duties, the steam flow rate decays slightly to 16 percent
- **Superheated Steam Production to Turbine Synchronization:** When the temperatures of the live steam and the reheat steam meet the startup requirements of the turbine, a portion of the steam in the bypass system is diverted to the turbine, the turbine is accelerated to synchronous speed, the turbine is synchronized with the grid, and output of the turbine is increased to the point where the generator is not at risk of tripping on reverse power. Steam not required by the turbine flows to the condenser through the bypass system
- **Turbine Synchronization to Normal Operation:** The extraction feedwater heaters are placed into service, the balance of the steam flow in the bypass system is sent to the turbine, the attemperation flow of cold salt is reduced consistent with an the maximum allowable rate of temperature change for the superheater / reheater, and the flow rate of hot salt is increased to the design value

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- **Normal Operation to Isothermal Steam Production:** The speed of the hot salt pump is reduced, which reduces the output of the turbine. When the turbine reaches the minimum allowable load, the turbine is tripped. Tripping the turbine at the full superheat / reheat temperature maintains the turbine metal temperatures as high as possible during the overnight hold period, which reduces the time, the thermal energy, and the metal stresses during the next startup. However, the steam generator is not tripped, and the steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent
- **Isothermal Steam Production to Overnight Hold:** The hot salt / cold salt mixing station is transferred from a point upstream of the superheater / reheater to the startup point upstream of the evaporator. During the transition between mixing stations, the superheating duty decreases relative to the evaporation duty. This results in 1) an increase in the steam flow rate to a nominal value of 20 percent, 2) a decay in the metal temperatures of the superheater / reheater to the cold salt temperature. The rate of change in the metal temperatures can be controlled, in part, by adjusting the saturation temperature in the evaporator. During the transition, the goal is to provide steam flow rates that are high enough to 1) ensure a reasonably uniform flow distribution among the tubes, and 2) reduce the potential for damaging temperature and stress distributions in the superheater / reheater
- **Turbine Synchronization to Isothermal Steam Production:** The transition is essentially a generator trip on reverse power. Although the turbine trips, the steam generator is not tripped, and the steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent
- **Overnight Hold to Long Term Hold:** The attemperation pump is stopped, the salt equipment is drained to the main storage tanks, the electric heat trace system is turned off, the water side of the equipment is drained, and nitrogen is supplied to the heat exchangers for corrosion control.

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4. Risk Analysis

4.1 Introduction

The goal of the risk analysis is to identify the principal equipment items that have a significant influence on the plant availability and the annual plant performance. The evaluation includes an assessment of probability and consequence to identify the most important items to be included in the technical specification.

As background to, and a starting point for, the risk assessment, an evaluation of the technical, commercial, and operating risks in commercial solar projects was recently conducted by NREL¹. The SolarPACES CSP project database² was used to identify the projects that are currently in commercial operation, including 76 parabolic trough plants and 14 central receiver projects. Over the course of the project, the project team held about 50 information gathering sessions, representing nearly two-thirds of the plants operating worldwide.

The principal technical risks associated with central receiver projects are summarized in Table 4-1. The projects surveyed include plants both with and without thermal storage.

To help identify which issues are most important, each issue entered in the database was given an impact score and a risk level. The impact score identified the potential impact of the issue to the project in terms of the effects on plant performance, cost, or schedule. The impact score was ranked as 1 (low) to 5 (high). The risk level was an indication of how likely the problem was to happen. A risk level of 1 meant that the problem was rarely experienced or maybe was only associated with a problem at a single plant. A risk level of 5 meant that it was a common problem or could affect many plants. The scores are multiplied to create a priority score. Priority scores can range from 1 to 25 for each issue. The ranking is, of course, subjective, but it is an attempt to give some quantification to the importance of the issues.

The most significant issues were brought up by multiple participants. The number of “occurrences” does not correspond to the number of times some type of incident or issue occurred at a plant, but rather the number of times it was mentioned by the plant personnel. The number of occurrences in which an issue is mentioned indicates how important the issue is to the stakeholders.

¹ Mehos, Mark, et. al, (National Renewable Energy Laboratory, Golden, Colorado), ‘Concentrating Solar Power Best Practices Study’, Technical Report NREL/TP-5500-75763, June 2020

² <http://www.nrel.gov/csp/solarpaces/>

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Table 4-1 Principal Risks for Central Receiver Projects Identified in the Best Practices Report

System	Component	Issue Type	Occurrence	Weight
Power block	Salt steam generator	Steam generator reliability	13	245
Thermal storage	Salt tanks	Tank design	10	204
Power block	Salt steam generator	Steam generator design	8	174
Project	O&M	Heliostat cleanliness	9	123
Heliostat field	System	Design standards	8	122
Receiver	Downcomer	Downcomer design	8	110
Receiver	Salt piping	Heat tracing	8	110
Heliostat field	Mirrors/facets	Heliostat optical quality	9	105
Receiver	Tower	Tower construction	6	100
Thermal storage	Salt tanks	QA/QC	4	100
Power block	DCS	DCS logic	5	89
Heliostat field	System	Heliostat qualification	6	84
Receiver	Salt piping	Valve design	7	77
Receiver	Control systems	Aiming strategy	3	75
Thermal storage	Salt tanks	Tank foundation	5	73
Receiver	Control systems	Automation	4	70
Receiver	Downcomer	Piping support design	3	65
Receiver	Outlet vessel	Outlet vessel design	4	60
Receiver	Salt piping	Valve reliability	4	60
Project	EPC	EPC execution	3	55
Receiver	Control systems	Receiver reliability	3	55
Receiver	System	Heliostat/receiver integration	5	55
Project	Engineering	Technology scale-up	2	50
Receiver	Receiver	Receiver reliability	7	43
Receiver	Control systems	Infrared camera	3	39
Power block	DCS	Automation	2	30
Project	EPC	Schedule	2	30
Receiver	Tower	Elevator	4	30
Receiver	Receiver	Receiver coating	3	27
Heliostat field	Control	Design specifications	1	25
Heliostat field	Mirrors/facets	Heliostat cleanliness	1	25
Project	EPC	Welding	1	25
Receiver	System	Receiver reliability	3	25
Thermal storage	Salt tanks	Salt heater design	3	25
Heliostat field	Control	BCS calibration	2	24
Heliostat field	Drives	Heliostat availability	4	24

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System	Component	Issue Type	Occurrence	Weight
Heliostat field	Mirrors/facets	Facet blocking	2	18
Heliostat field	Power/wiring	Electrical system design	2	18
Thermal storage	Salt	Corrosion	2	18
Heliostat field	Control	Heliostat/receiver integration	1	15
Heliostat field	Drives	Drive qualification	1	15
Power block	Steam cycle	Valve reliability	1	15
Project	Structure	EPC experience	1	15
Receiver	Control system	Flux meter	1	15
Receiver	Receiver	Automation	1	15
Thermal storage	Salt	Water emulsion	1	15
Receiver	Downcomer	Downcomer control	3	13
Thermal storage	Hot salt pump	Pump design	3	13
Heliostat field	Mirrors/facets	Heliostat availability	2	12
Power block	Steam turbine-generator	Turbine reliability	2	12
Power block	Steam cycle	Pump reliability	3	11
Power block	Salt steam generator	Pump alignment	2	10
Power block	Steam turbine-generator	Generator reliability	2	10
Receiver	Receiver	Receiver design	2	10
Heliostat field	Civil	Heliostat cleanliness	1	9
Heliostat field	Environmental	Heliostat flux hazard	1	9
Heliostat field	Heliostat structure	Pedestal installation	1	9
Heliostat field	Power/wiring	Lightning	1	9
Heliostat field	System	Optics versus cost	1	9
Power block	Auxiliary systems	Water supply	1	9
Power block	DCS	Instrument reliability	1	9
Receiver	Cold salt pump	Pump reliability	1	9
Receiver	Salt piping	Piping design	1	9
Thermal storage	Piping	Insulation quality	1	9
Power block	Auxiliary systems	Fire system design	1	5
Thermal storage	Hot salt pump	Pump reliability	1	5
Heliostat field	Civil	Site preparation	1	3
Power block	Auxiliary systems	Hybrid cooling	1	3
Power block	Electrical	Design specifications	1	3
Receiver	Salt piping	Safety valves	1	3
Thermal storage	Salt tanks	Testing standards	1	3

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A review of the table shows the following:

- The salt equipment topics with the highest weight values are the design and the reliability of the salt tanks and the steam generators
- In contrast to the salt tanks in parabolic trough projects, several failures have occurred in the hot salt tanks in central receiver plants. This can partially be attributed to 1) the large temperature difference between the cold tank and the hot tank, and 2) high rates of temperature change seen at the hot tank during receiver startup and following a receiver trip
- Also, in contrast to the steam generators in parabolic trough projects, problems with the salt steam generators in central receiver projects are considerably more prevalent. This is likely due to the following effects:
 - A requirement for the accurate blending of cold salt with hot salt during the startup and the shutdown of the steam generator
 - A higher freezing point for nitrate salt (220 °C) than for Therminol (12 °C).

4.2 *Equipment Risks*

For the purposes of the Design Basis study, the risk evaluations are based on estimates of the daily probability of the event occurring, multiplied by the number of forced outage days to affect repairs or correct the problem. Potential mitigation or maintenance responses are included in the evaluation. The product of probability and outage is then multiplied by a factor of 330 to account for the number of operating days each year to calculate a value for the annual risk.

Equipment risks for the salt systems are summarized in Table 4-2 through Table 4-8 below. It can be noted that there are essentially no public data on the reliability and the availability of salt equipment and salt systems in commercial projects. This is due to 1) a limited number of commercial projects, and 2) the projects often treating the data as proprietary for commercial purposes. The figures in the tables were largely developed on an ad hoc basis, relying on incomplete information from a limited set of projects. As additional information becomes available, the values in the tables should be reviewed and revised as needed.

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Table 4-2 Equipment Risks Associated with the Salt Receiver

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Incomplete preheating or venting leads to salt freezing in the panels during filling	0.05	2	33	1
Sticking drain valves lead to salt freezing in the panels during draining	0.02	5	33	1
Failures of backwall thermocouples preclude automatic sequence for fill	1	0.1	33	2
Large optical errors in the heliostat field lead to high spillage losses	0.5	0	0	3
Incorrect aim point selection leads to reduction in panel low cycle fatigue life	0.05	7	116	4
Incorrect isolation of panels during overnight hold results in dew point condensation and intergranular stress corrosion cracking	0.002	7	5	5
Salt leakage from valves, tube corrosion, or instruments increases heat losses from the oven insulation, leading to lethargic preheating	0.05	0.1	2	6
Incomplete venting of pump discharge during pump start produces air pocket in riser and receiver trip on perceived loss of flow	0.05	0.1	2	7
Sticking or erratic movement of inlet control valves results in receiver trip on high outlet temperature	0.01	1	3	8
Loss of level signal in inlet vessel results in liquid levels which are too high or too low	0.02	0.05	1	9
Leakage of inlet vessel compressed air solenoid valves results in low pressure in inlet vessel	0.01	1	3	10
Loss of level signal in outlet vessel results in liquid levels which are too high or too low	0.02	0.05	1	9

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Sticking downcomer throttle valves results in liquid levels in the outlet vessel which are too high or too low	0.02	1	7	11

Mitigation or Response:

1. The panels are thawed using a limited flux from the heliostat field. The process is necessarily lethargic, as the primary goal is prevent (or limit) yielding of the tubes during the phase change.
2. The number of backwall thermocouples is in the range of 150 to 200. The probability that all of the thermocouples are operating correctly, which is required to satisfy the permissive for the receiver fill logic, approaches zero. As such, the operator is required to make a qualitative assessment of the completeness of the preheating, which leads to a delay in receiver startup.
3. Once the heliostats are installed, correcting slope errors is difficult. High spillage losses do not result in an outage, but do result in a decrease in the annual sunlight-to-electric efficiency.
4. Adjusting aim points to compensate for high spillage losses can result in tube strains inconsistent with a 30-year low cycle fatigue life. A tube failure requires the replacement of a tube or a panel.
5. Nickel alloys, such as Inconel 625 LCF and Alloy 230, can be susceptible to intergranular stress corrosion cracking in the presence of liquid water. Air saturated with water, either from the ambient or from the vent line to the hot salt tank, must be excluded from the receiver panels.
6. Oven preheating must begin earlier in the morning, and panel preheating will take longer than normal.
7. Receiver flow meters interpret an air flow as a loss of salt flow, which trips the receiver. Restarting the receiver requires draining, re-preheating, and refilling.
8. The stem packing in the control valves must be replaced.
9. The level monitoring and control are switched to the redundant level instrument.
10. The receiver is shutdown to repair the solenoids valve.
11. The receiver is shutdown to repair the control valve.

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Table 4-3 Equipment Risks Associated with the Salt Storage Tanks
(Identical to Table 6-1 in Volume 1 - Parabolic Trough Projects with Inorganic Fluids)

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt leakage	0.001	250	82	1
Foundation settlement	0.0001	250	8	2
Loss of foundation cooling	0.005	1	2	3
Defects in the tank insulation ¹	0.1	0	0	4
Defects in the tank insulation ²	0.0005	250	41	5
Loss of inventory level signal	0.1	0	0	6
Loss of electric salt heaters	0.01	0	0	7

Notes:

1. Local gap, leading to a local increase in the heat loss.
2. Local gap, exposing the outside of the tank to water. In combination with salt spills on the outside of the tank, the carbon steel shell develops stress corrosion cracking.

Mitigation or Response:

1. Transfer the salt inventory to the companion storage tank while repairs are made to the tank which is leaking.
2. Remove the tank floor, lift the tank, remove and replace the foundation, lower the tank, and replace the floor.
3. Move the electric power supply to the redundant fan. The thermal inertia of the foundation will prevent excessive soil temperature during the period required to start the redundant fan.
4. Repair the defects when discovered. This is to avoid thermal gradients in the shell, which have the potential to establish stresses equal to the yield value.
5. The risk for carbon steel tanks is discussed above in Section **Error! Reference source not found..**
6. Use the redundant level instrument(s) until repairs are made to the failed instrument.
7. Replace the failed element(s) in the salt heaters. The temperature of the tank will not meaningfully decrease in the time required to repair the heater.

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Table 4-4 Equipment Risks Associated with the Salt Steam Generator
(Identical to Table 6-2 in Volume 1 - Parabolic Trough Projects with Inorganic Fluids)

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Leakage due to failures of the tube-to-tubesheet connections	0.05	15	247	1
Pitting corrosion in evaporators using stainless steel tubes ¹	0.01	15	50	2
Leakage results in excessive water vapor accumulation on the salt side	0.001	15	5	1
Leakage leading to salt exposure to water vapor	0.01	0	0	3
Leakage leading to water vapor transfer to the ullage in the salt tanks	0.01	0	0	3

Note:

1. Inadequate control over water chemistry leads to flow accelerated corrosion in the carbon steel components of the condensate system. Accumulation of deposits in evaporator tubes leads to cathodic damage to passivation layer on tubes.

Mitigation or Response:

1. Drain the heat exchangers on the salt- and the water/steam-sides, repair the tube leaks by plugging, fill on the water side, preheat by means of the electric water heaters, and fill on the salt side.
2. Maintain dissolved oxygen and pH within values required by the water treatment system.
3. Salt/water reactions are largely benign and commercially acceptable.
4. Water vapor is vented from the ullage spaces during the daily charge/discharge cycles.

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Table 4-5 Equipment Risks Associated with the Salt Pumps

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Loss of receiver cold salt pump	0.0005	2	7	1
Loss of steam generator hot salt pump	0.0005	2	7	2
Loss of steam generator attemperation pump	0.0005	2	7	3

Mitigation or Response:

1. Replace the pump with the warehouse spare.
2. Loss of the hot salt pump trips the steam generator. Replace the pump with the warehouse spare. Once the steam generator cools to the cold salt temperature, the equipment can be maintained at the cold salt temperature by means of the electric water heaters and the heat trace cables on the heat exchangers.
3. Replace the pump with the warehouse spare. The steam generator can be maintained at the cold salt temperature by means of the electric water heaters and the heat trace cables on the heat exchangers.

Table 4-6 Equipment Risks Associated with the Salt Valves

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Sticking cold salt pump discharge isolation valves ¹	0.01	1	3	1
Sticking cold salt pump discharge isolation valves ²	0.001	1	1	1
Sticking cold salt pump minimum flow recirculation valves ¹	0.01	1	3	1
Sticking cold salt pump minimum flow recirculation valves ²	0.001	1	1	1
Sticking hot salt or attemperation pump discharge isolation valves ¹	0.01	2	7	2
Sticking hot salt or attemperation salt pump discharge isolation valves ²	0.001	2	1	2

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Sticking hot salt or attenuation pump minimum flow recirculation valves ¹	0.01	2	7	2
Sticking hot salt or attenuation pump minimum flow recirculation valves ²	0.001	2	1	2
Sticking receiver vent and drain valves ¹	0.005	5	8	3
Sticking receiver vent and drain valves ²	0.0005	5	1	3
Sticking heat exchanger vent and drain valves ^{1,3}	0.005	5	8	4
Sticking heat exchanger vent and drain valves ^{2,3}	0.0005	5	1	4
Internal leakage in downcomer throttle valves	0.01	1	3	5
Internal isolation valve leakage in the steam generation system ⁴	0.001	5	2	2
External leakage in receiver system isolation valves ¹	0.01	5	17	6
External leakage in steam generator isolation valves ¹	0.01	5	17	6

Notes:

1. Failure rate with conventional valve stem packings.
2. Failure rate with bellows valve stem seals.
3. In conjunction with water leakage rates which are high enough to adversely affect the flow distributions on the salt side; i.e., vapor binding.
4. Internal leakage prevents accurate control over heat exchanger temperature during transient conditions.

Mitigation or Response:

1. Shut down the receiver, drain the riser, repair or replace the valve, and restart the receiver.
2. Drain the heat exchangers on the salt side, repair or replace the valve, and refill the heat exchangers. The steam generator can be maintained at the cold salt temperature by means of the electric water heaters and the heat trace cables on the heat exchangers.
3. Sticking panel drain valves will lead to salt freezing in the panels. The thawing process is intentionally lethargic to reduce the potential for plastic deformations of the tubes, and the associated reduction in low cycle fatigue life, during the phase change.

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4. If the heat exchangers cannot be drained, then repairs can begin only after the heat exchanger temperatures have decayed below 200 °C. Repair or replace the valves, preheat on the water side, and then fill on the salt side.
5. Shut down the receiver, drain the downcomer, repair or replace the valve, and restart the receiver.
6. Drain the piping, replace the stem packing, and replace any insulation or heat trace cables that have been damaged by exposure to the salt.

Table 4-7 Equipment Risks Associated with the Salt Instruments

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Inaccurate, or sporadic loss of, flow meter readings ¹	0.05	0.1	2	1
Inaccurate, or sporadic loss of, flow meter readings ²	0.05	0	0	2
Salt freezing in the pressure instruments	0.05	0.5	8	3
Inaccurate, or loss of, temperature readings	0.01	0.5	2	4

Notes:

1. Receiver system
2. Systems other than the receiver system

Mitigation or Response:

1. Loss of flow signal is interpreted as a loss of flow, and the control system responds by tripping the heliostat field and the receiver. Drain the receiver system, and restart the receiver.
2. Switch to manual operation until output signals from flow meters are corrected.
3. Repair the insulation and/or the heat tracing on the instrument stubs.
4. Replace the thermocouple, or switch to the redundant dual-element connections.

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Table 4-8 Equipment Risks Associated with the Heat Tracing
(Identical to Table 6-5 in Volume 1 - Parabolic Trough Projects with Inorganic Fluids)

	Daily probability, fraction	Outage duration, days	Annual product	Mitigation or Response
Salt freezing in the pressure instruments	0.05	0.1	2	1
Salt freezing in the heat exchanger vent and drain lines	0.01	2	7	2
Damaged insulation and marginal heat trace capacity result in lines, when frozen, cannot thaw ¹	0.01	3	10	3

Note:

1. Problems primarily associated with the small diameter lines, such as the heat exchanger vent lines, the heat exchanger drain lines, and the instrument stubs.

Mitigation or Response:

1. Repair the insulation and/or the heat tracing on the instrument stubs.
2. Repair the insulation and/or the heat tracing on the vent and drain lines.
3. Repair the insulation and/or the heat tracing on the salt lines, followed by a waiting period for thawing.

A review of the annual product values shows that the major risks have, to a first order, a value greater than 10 and the minor risks have a value less than 10. The largest risks to the equipment items are summarized below in Table 4-9. Detailed discussions of each of the first 4 of the 10 principal risks are presented in the sections which follow. The fifth through the tenth principal risks are identical to the corresponding risks discussed in the risks analyses for trough projects using organic and inorganic heat transfer fluids.

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Table 4-9 Summary of the Highest Equipment Risks

Equipment Item	Risk	Annual Product, Probability * Duration
Receiver	Incomplete preheating or venting leads to salt freezing in the panels	33
Receiver	Sticking drain valves leads to salt freezing in the panels	33
Receiver	Failure of backwall thermocouples precludes automatic fill sequence	33
Receiver	Incorrect aim point selection leads to reduction in panel low cycle fatigue life	116
Thermal storage ¹	Leakage in salt tanks	82
Thermal storage ²	Stress corrosion cracking of carbon steel in the cold salt tank	41
Salt steam generator ¹	Leakage due to failures of the tube-to-tubesheet connections	247
Salt steam generator ¹	Pitting corrosion in evaporators using stainless steel tubes	50
Salt valves ²	Valve steam leakage past conventional stem packings	17
Heat tracing ²	Damaged insulation and marginal heat trace capacity result in lines, when frozen, cannot thaw	10

Notes:

1. Same as the risks for parabolic trough plants using inorganic fluids
2. Same as the risks for parabolic trough plants using organic fluids or inorganic fluids

4.2.1 Risks Associated with the Receiver

Incomplete Preheating or Venting Leads to Freezing in the Panels

Preheating and Venting Process

Each receiver panel consists of a lower header, a series of curved jumper tubes which connect the header with the bottom of the tubes in the panel, the absorber tubes, a series of curved jumper tubes which connect the top of the tubes in the panel with the upper header, and the upper header. The bottom header and the lower jumper tubes are located inside a lower oven. Similarly, the top header and the upper jumper tubes are located inside an upper oven.

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Prior to filling with salt, the panels are preheated to a nominal temperature of 300 °C by a preheat flux of 35 to 60 kW/m² from the heliostat field. The headers and the jumper tubes are preheated by radiant heaters located inside the oven.

The panels are flood filled from the bottom. The panels are considered full when a liquid level is established in the outlet vessel above the panels.

Non Ideal Preheating Conditions

Under conditions of clear skies and calm winds, preheating of the panels and the ovens typically proceeds smoothly over a period of 30 to 45 minutes. However, if the skies are cloudy, or if the wind speeds are above calm, then preheating can be accompanied by the following characteristics:

- Rapid changes in the solar radiation can lead to rapid changes in the local value of the preheat flux
- Movements of the preheat heliostats, produced by changes in the wind speed or changes in the wind direction, can result in local variations of the flux on the absorber
- Heat losses from the oven due to wind infiltration can extend the preheat period, or result in oven temperatures below the desired fill value of 300 °C. The problems are compounded 1) if rain water or salt has leaked into the oven and saturated the insulation, or 2) not all of the radiant heaters are available for service.

Sources of Local Freezing

Local freezing can develop during the fill process under the following conditions:

- The temperature of the salt entering the absorber tubes is less than 300 °C due to difficulties in preheating the lower oven
- Local absorber temperatures are less than 300 °C due to random variations in both the preheat flux and the preheat duration
- During the fill process, salt velocities in the tubes are on the order of 0.05 to 0.1 m/sec, and the salt residence time in the absorber section can be as long as 2 minutes.

If the temperature of the salt entering the bottom of the tubes is only slightly above the freezing point, then local variations in the preheat fluxes, the convective wind losses, or the residence time of the salt in the absorber could result in the formation of salt plugs or frozen tubes.

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Incomplete Venting and Local Freezing

During daily startup, the receiver panels, the drain lines, and the vent lines are flood-filled from below. The goal is to remove all of the air from the system prior to the start of normal operation. However, there is no direct means of determining if the air has been vented during filling. As a result, the decision to change the operating state from 'Fill' to 'Operate' is largely at the discretion of the operator.

Under conditions of clear skies and little wind, the operator can assign a generous time to the Fill process, which results in a low risk of retaining air in the receiver. However, under conditions of cloudy skies on a windy day, the operator will be motivated to make the transition to Operate as soon as possible to limit the potential for salt freezing in the panels.

If air is trapped in the receiver, it will likely reside at the top of the upper header in each panel, or at the top of the jumper lines between adjacent panels. Following the transition to Operate, the flow changes from all up-flow to serpentine, and the flow rate increases by at least an order of magnitude. Any air trapped in those headers or jumper lines located upstream of the downflow panels will be pushed down into the panels. This can cause a maldistribution among the parallel flow paths, largely driven by the buoyant forces of air in the tubes. If the flow stalls in some tubes during non-stable preheat conditions, the risk of freezing is high.

Thawing Process

In the phase change from solid to liquid, the volume of salt increases by 4.6 percent. Were the salt to be constrained in a tube during the melting process, the tube would experience a strain of 2.3 percent, which is well beyond the yield strain. A strain of this magnitude would significantly reduce the low cycle fatigue life of the tube, and could lead to a future forced outage to replace the tube.

To thaw frozen salt in a tube or in a panel, the following steps are required:

- The location and dimensions of the frozen zone must be defined. This can be done by preheating the panel to a nominal temperature of 200 °C, removing the heating flux, and then observing the temperature decay rate using an infrared camera.
- A thawing procedure is developed, which defines the heating heliostats, the heating flux distribution, and the aim points for the heliostats. The goal is to heat the zone below the frozen section, and then to melt the frozen zone from the bottom moving up. This approach provides an exit path for the salt as it melts, and reduces the potential for melting salt in a constrained volume.

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- As the thawing proceeds, the location and dimensions of the frozen zone will change. As such, the heating heliostats, the heating flux distribution, and the aim points for the heliostat will need to be periodically revised.

Due to the essentially infinite number of potential frozen salt distributions in the absorber, and the potential range in the skills and the patience of the operating staff to develop a thawing procedure that minimizes the damage to the tubes, the risks to the receiver associated with frozen salt are significant.

Sticking Drain Valves

If a panel drain valve fails to open at the end of the day, then all of the salt in the panel will freeze within a few minutes of sunset.

A panel thawing process will need to be undertaken, as discussed in the section above.

Failure of the Backwall Thermocouples

An array of 16 to 20 thermocouples is installed on the back of select tubes in the panel. The goal of the thermocouple measurements is to provide a two-dimensional image of the temperature distribution in the absorber panel. Commercial receivers have 14 to 16 panels, which translates to 224 to 320 backwall thermocouples. The thermocouples are typically tack welded to the tubes.

In some commercial projects, the control system logic requires that all backwall thermocouples show a minimum preheat temperature (say, 275 °C) before a permissive is given to fill the receiver. However, the tubes are subject to daily temperature cycles of 275 to 550 °C, and exposed to rates of temperature change as high as 6 °C/sec following a receiver trip. These conditions cause one of two conditions to occur:

- The thermocouple tack welds crack, and the contact between the tube and thermocouple is lost. Natural convection air flows between the back of the tubes and the front of the panel insulation result thermocouple readings that are 50 to 75 °C lower than the tube temperatures.
- The bimetallic connection inside the thermocouple fails, which causes the temperature value to go to full scale.

The failure rates are higher than might be expected; perhaps 1 percent of the thermocouples per month. As such, soon after the receiver begins commercial operation, the permissive to fill the receiver can no longer be satisfied.

Clearly, this is commercially too restrictive. Instead, the operator will be required to make a decision as to fill the receiver, or continue with preheating, based on the distribution of the readings from the

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thermocouples which are in service. The distribution will be random and arbitrary, based on when and where the thermocouples fail. Further, the judgement criteria will need to evolve as the thermocouples fail or as the thermocouples are repaired.

The ability of the project to fill the receiver as quickly, and as reliably, each day will depend on the skills, the experience, and the patience of the operator. This, in turn, represents a risk to the project, as the capabilities of the an operator are much more qualitative than logic in the control system.

Incorrect Aimpoint Selection

Aim Point Calculation

Each heliostat has an assigned aim point on the receiver. The location of the aim point changes with the time of the day, and the day of the year, to satisfy the following conflicting goals:

- Assign all of the aim points to the equator of the receiver. This minimizes the spillage losses, but maximizes the peak incident flux on the receiver.
- Uniformly spread the aim point over the height of the panel. This minimizes the peak incident flux, but maximizes the spillage losses.

The design software for the heliostat field (HFLCAL, Tietronix/R-Cell) calculates the aim points maps that keep the spillage losses as low as possible consistent with the tube strain limits associated with a low cycle fatigue life of 30 years. The aim point maps are based on heliostat slope errors and heliostat pointing errors provided by the heliostat vendor. The project will require the vendor to warrant the maximum errors.

Receiver Design Approach

Thermal losses from the receiver are held to a minimum by operating the receiver at flux levels as high as possible. Since the receiver is designed to operate for the life of the project, the accumulated tube fatigue and creep damage must be less than the damage associated with tube rupture. To satisfy this requirement, the receiver can be designed based on the following approach:

- The tubes are exposed to the incident flux only from the front side. Further, the tube is constrained to remain in the plane of the absorber. As a result, when in service, the front of the tube is placed in compression, and the back of the tube is placed in tension.
- When the receiver is operated for the first time at the design flux, the compressive strains at the front of the tube near the equator of the receiver exceed the yield value, and the tensile strains at the back of the tube exceed the yield value.

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- At the end of the day, the receiver cools to ambient temperature. The compressive stresses at the front of the tube relax, but the plastic compressive strains which occurred earlier in the day result in residual tensile stresses at the front of the tube. Similarly, at the end of the day, residual compressive stresses are established at the back of the tube.
- When the receiver is operated for the second time, compressive strains are again established on the front of the tube near the equator of the receiver. The strains are at, or slightly exceed, the plastic limit. However, the strains are lower than the strains established on the first day due to the residual tensile stresses. Similarly, the tensile strains at the back of the tube are lower than the tensile strains on the first day due to the residual compressive stresses. This effect is known as elastic shakedown.
- When the receiver is operated for the third time, the strains are similar to those experienced on the second day.

The process, which repeats on Day 3 and beyond, is illustrated as the series of hysteresis loops shown at the upper left of the Bree diagram in Figure 4-1. As long as the stresses remain in the region of Low Cycle Fatigue, then the receiver will last the life of the project.

Heliostat Optical Errors

In commercial projects, the heliostats are typically constructed in an assembly building. One step in the fabrication process is installing the mirrors on the structure, and canting the modules to the slant range selected for that heliostat. Once the canting is complete and verified, the heliostat structure is transported from the assembly building to the assigned pedestal. However, the transportation loads transferred from the truck to the heliostat structure can change, often on a random basis, the cant of one or more of the modules.

At some commercial projects, the heliostat optical errors have exceeded the warranted values. Once a heliostat is installed, it is often possible to correct the drive pointing errors, but it is difficult to correct the mirror slope errors (i.e., adjust the mirror cant). If the slope errors are larger than expected, then the receiver spillage losses will be higher than expected and the annual plant performance will be less than the value estimated in the performance model.

One method for reducing the spillage losses is to move the heliostat aim points closer to the equator of the receiver. However, this necessarily increases the peak incident flux near the equator, which, in turn, increases the tube strains at the center of the panel. If the tube stresses increase to the point of ratcheting, as shown in the Bree diagram, then the center of the panel can fail due to rupture. This consequence is not theoretical, as one commercial project experienced tube failures in an effort to compensate for high spillage losses.

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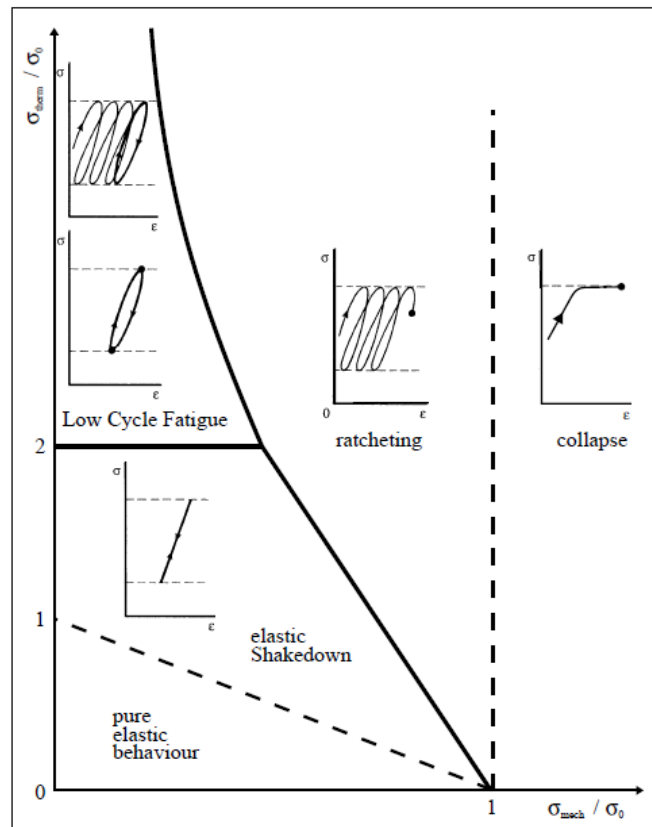


Figure 4-1 Bree Diagram of Primary Mechanical Stress and Secondary Thermal Stress

Recommendations

If, after construction, the receiver spillage losses are higher than expected due to heliostat optical errors that are higher than expected, then the heliostat aim points can be adjusted to compensate. The first step is to measure the drive pointing errors and the mirror slope errors throughout the field. Once these are known, revised incident flux maps can be calculated over the course of the year. Revised heliostat aim points can then be recalculated, which reduce the spillage losses to values as low as possible while maintaining the low cycle fatigue life at the required value of 30 years.

With an increase in the heliostat optical errors, the annual performance of the receiver will decrease. The reduced performance will need to be reflected in the projected energy sales to the local utility. Stated another way, the project cannot expect to compensate for an increase in the heliostat optical errors by reducing the low cycle fatigue life of the receiver.

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4.2.2 Leaks in Stainless Steel Hot Salt Tanks

A discussion of the risks associated with leaks in stainless steel hot salt tanks is presented in Section 6.2.1 of Volume 1 - Specifications for Parabolic Trough Projects.

4.2.3 Leaks in Carbon Steel Salt Tanks

A discussion of the risks associated with leaks in carbon steel salt tanks is presented in Section 5.2.1 of Volume 1 - Specifications for Parabolic Trough Projects.

4.2.4 Leaks in the Steam Generator

A discussion of the risks associated with leaks in a salt steam generator is presented in Section 6.2.2 of Volume 1 - Specifications for Parabolic Trough Projects.

4.2.5 Salt Valves

A discussion of the risks associated with salt valves is presented in Section 5.2.3 of Volume 1 - Specifications for Parabolic Trough Projects.

4.2.6 Heat Tracing

A discussion of the risks associated with heat tracing is presented in Section 5.2.4 of Volume 1 - Specifications for Parabolic Trough Projects.

4.3 *System Risks*

The discussion above, in Section 4.2, evaluated risks associated with the salt equipment. However, there are also risks at the project level which can influence the availability of the plant. Two such risks, which are identical to the project risks listed in Table 5-8 and Table 6-8 in Volume 1 - Parabolic Trough Specifications, are presented in Table 4-10.

4.3.1 Process Design

The process design requirements for a central receiver project are essentially the same as the process design requirements for a trough project using organic heat transfer fluids, as discussed in Section 5.3.1 in Volume 1 - Specifications for Parabolic Trough Projects.

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Table 4-10 System Risks for the Salt Equipment

	Annual probability, fraction	Annual outage duration, days	Annual product	Mitigation or Response
Process design is not consistent with the performance requirements in the project financial model	0.5	20	10	1
Operating personnel do not have an understanding of the plant that is sufficient to prevent damage to the equipment	0.5	20	10	2

Mitigation or Response:

1. Research the process design in other commercial projects. Copy those features which are successful, and avoid those features which are not effective.
2. Provide sufficient funds to attract, and retain, a mechanical engineer at the site for the first 5 years of commercial service. Provide ad hoc training to the operators in topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

4.3.2 Selection of Operating Personnel

A discussion of the risks associated with the selection of the operating personnel in parabolic trough plants using organic heat transfer fluids is presented in Section 5.4 of Volume 1 - Specifications for Parabolic Trough Projects. The risks associated with the selection of the operating personnel in parabolic trough plants using inorganic heat transfer fluids is presented in Section 6.4 of Volume 1.

Central receiver projects necessarily operate through daily thermal cycles. But, the plant has a number of features which make the design more fragile in cycling conditions than parabolic trough plants using inorganic heat transfer fluids. These features include the following:

- The difference in temperature between the cold side (295 °C) and the hot side (565 °C) is a meaningful 270 °C. This difference is almost 3 times the difference in a parabolic trough project. During startup and shutdown of the steam generator, hot salt is blended with cold salt. The blending proportions are continuously adjusted such that the vendor limits on rate of temperature change and thermal shock are met. The blending fractions are determined by the distributed control system, and are adjusted by changing salt pump speeds and control valve positions. However, automatic control is not a guarantee of accurate control. For example, a common pump and valve arrangement includes 1) two hot salt pumps operating in parallel with one cold salt attemperation pump, 2) a common minimum flow recirculation valve for the hot salt pumps, and 3) system control valve sizes selected to provide a small pressure drop at the

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design condition. At the low flow rates characteristic of startup and shutdown, the large control valves can provide only a small pressure drop. This obligates the hot salt and the attemperation pump to operate at their minimum speeds. As such, small adjustments in the speed of either pump, or small adjustments in the positions of either of the two minimum flow recirculation valves, can produce undesirable and random changes in the blending proportions of hot salt and cold salt. The result can be step changes in the mixed salt temperature, or oscillations in the mixed salt temperature. Either condition can impose transient thermal stresses on the heat exchangers which are outside of the vendor limits. In an effort to protect the equipment from low cycle fatigue damage, the operator may be obligated to suspend automatic control, and to make manual adjustments to pump speeds and control valve positions. The low cycle fatigue life of the equipment will then depend on the skill of the operator to interpret, and manipulate, a range of independent variables and remain within the transient limits set by the vendor. As might be expected, some operators will be more successful than others.

- The receiver tubes normally operate at strain levels which are slightly greater than the elastic limit. The low cycle fatigue life of the material is very sensitive to the level of plastic strain, and the degree to which plastic strains can accumulate; i.e., ratcheting. The strains are a function of the incident flux distribution, the thermal losses from the panel surface, and the convection heat transfer coefficients inside the tubes. The second two characteristics are well understood and predictable. However, the incident flux distribution is a function of the optical errors of the heliostats and the aiming strategy. The ability to characterize the optical errors varies from project to project. If the optical errors are not well understood, or if the errors are larger than anticipated, then it is difficult to select the aimpoint distributions that make accurate tradeoffs between receiver output and spillage losses. To some degree, the low cycle fatigue life of the receiver may be unknowingly, and significantly, reduced by even modest attempts to control spillage losses. At one commercial project, revisions to the aimpoints resulted in the melting of several tubes.

Recommendations

The recommendations regarding plant personnel for central receiver projects are the same as the recommendations for personnel as trough projects using inorganic heat transfer fluids, with one exception. The former type of projects are more complex, and more brittle, than the latter due to equipment operating at plastic strains levels (receiver), and due to larger temperature differences between the cold side and the hot side. The project must, rather than should, provide sufficient funds to attract, and retain, a mechanical engineer at the site for the first 5 years of commercial service. The engineer would perform the following:

- During the period required to refine the logic in the distributed control system, provide guidance to the operators to protect equipment from low cycle fatigue damage during startup, shutdown, and transient conditions

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- Provide ad hoc training to the operators in topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

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5. Operating Requirements

Operating requirements for central receiver projects include the following topics:

- Selection of design parameters, such as temperatures, flows, allowable fluxes, rates of temperature change, low cycle fatigue lives, and design Codes
- Evaluation of (Capital cost) versus (Capital cost + Operating cost). The evaluation considers items such as material selection, fabrication techniques, and installed redundancy
- Outline of hardware recommendations and specifications.

5.1 Receiver System

5.1.1 Performance Requirements

Temperatures

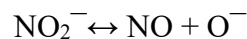
The operating temperatures of interest are the receiver outlet temperature and the receiver inlet temperature.

Receiver Outlet Temperature

The receiver outlet temperature should, in general, be as high as possible to provide a Rankine cycle efficiency that is as high as possible. However, the outlet temperature is limited by the thermal stability of the salt. The nitrate ion is in chemical and thermal quasi-equilibrium with the nitrite ion, as follows:



The nitrite ion, in turn, is in quasi-equilibrium with the oxide ion:



The reactions only reach quasi-steady state conditions for the following reasons:

- The reactions are influenced by the temperature. The peak temperatures in the system are the film temperatures in the receiver. However, steady-state reaction conditions are never achieved because 1) the film region of the tube is only a small fraction of the tube cross section area, 2) over the course a year, the residence time of the salt in the film region is only a small fraction of

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the overall period. Since steady-state conditions are never reached, the reactions do not proceed to completion

- NO generated in the second reaction is periodically vented to the atmosphere from the storage tanks
- Oxides produced from the second reaction are consumed by the oxidation of iron, nickel, and chromium in the system.

Experience from commercial projects has shown that oxide ($O^{\ominus}$) levels produced with a receiver outlet temperature of 565 °C provide an acceptable compromise in the following mutually exclusive features: 1) corrosion rates for carbon steels, stainless steels, and nickel alloys; and 2) Rankine cycle efficiencies.

Potential for Receiver Temperatures Above 565 °C

Oxides ions, together with chloride ions, are the principal corrosion agents in nitrate salt. If the oxide ion concentrations can be reduced, then it should be possible to operate the receiver at temperatures above 565 °C and still maintain acceptable metal corrosion rates. This, in turn, would result in a reduction in the unit cost of the storage system (\$/kWh) and an increase in the Rankine cycle efficiency.

One method for reducing the oxide ion concentrations is to reduce the nitrite ion concentrations. This can be accomplished by shifting the reactions, noted above, to the left by enriching the ullage gas concentrations of O_2 and NO. Adding O_2 and NO to the ullage gas is a straightforward exercise. However, there are a number of practical problems to be solved, as follows:

- O_2 is dangerous, as it supports combustion and it can more readily form an explosive atmosphere than air
- NO is a distinct hazard to operating and maintenance personnel
- NO, in the presence of water, forms nitric acid (H_2NO_3). As the plant ages, leaks in the steam generator are likely, if not inevitable. The effect on the corrosion characteristics of the carbon steel in the cold tank in the presence of nitric acid gas is currently unknown
- There is a cost associated with supplying O_2 and NO to the project, and venting O_2 and NO to the atmosphere during the daily tank charge and discharge process may be too expensive for commercial consideration. One method to reduce the operating cost is to withdraw the gas mixture from the hot tank during a discharge cycle, compress and store the gas, and then return the gas mixture to the cold tank during a charge cycle. However, the salt has a vapor pressure, and storing the compressed ullage gas in a pressure vessel at ambient temperature will lead to the

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accumulation of solid salt in the vessel. To prevent this condition, 1) the entire ullage gas recovery and storage system must be maintained at a temperature of at least 275 °C, and 2) the ullage gas compressor must be able to operate with inlet gas temperatures of at least 275 °C

- On a point related to leakage of the steam generator, water vapor introduced into cold tank must be vented to the atmosphere to maintain the tank ullage gas pressure below the relief valve setting. This, in turn, implies that some means of separating H₂O from O₂ and NO, at a nominal temperature of 275 °C, will need to be identified.

Receiver Inlet Temperature

The cold salt temperature is defined by the design conditions for the Rankine cycle, as follows:

- The unit work performed by the expansion of steam is increased by increasing the pressure ratio across the turbine. However, the expansion ratio cannot be so high as to result in excessive moisture (more than 8 or 9 percent) in the exhaust of the low pressure turbine. With a hot salt temperature of 565 °C, a representative live and reheat steam temperature is 540 °C. These temperatures, in turn, result in live steam pressures of 125 bar to a maximum of perhaps 140 bar
- The efficiency of the Rankine cycle is improved by increasing the final feedwater temperature. However, raising the final feedwater temperature increases the salt temperature at the cold end of the preheater in the steam generator, which, in turn, reduces the temperature difference between the cold salt tank and the hot salt tank. Optimum final feedwater temperatures are typically in the range of 220 to 240 °C.
- To provide a heat transfer area in the evaporator of the steam generator that can be safely warranted by the vendor, a pinch point of 5 °C is typically selected
- To help prevent water evaporation (steaming) at the hot end of the preheater during low load operation, the preheater is typically designed with a 5 °C approach to saturation.

The requirements above typically result in cold salt temperatures in the range of 295 to 305 °C, depending on the final selection of the live steam pressure and the final feedwater temperature.

Pressures

Heating the salt from an initial temperature of 300 °C to a final temperature of 565 °C requires a heating length of 150 to 200 m, as defined by the following:

- Thermodynamic properties of the salt (specific heat, thermal conductivity, viscosity)

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- Allowable film temperatures in the receiver tubes (generally 600 to 610 °C).

With a flow circuit divided into a series of panels, representative receiver pressure drops are on the order of 25 to 30 bar. Since the outlet of the receiver is vented to the atmosphere, receiver inlet pressures are in the range of 26 to 31 bar.

Flow Velocities

The allowable salt velocities in the tubes are generally limited by erosion considerations. However, the erosion rates of nickel alloy tubes are not well understood. In many commercial designs, the peak velocity is limited to about 4 m/sec, as this is taken to be a 'safe' value.

It can be noted that the number of hours each year in which the peak velocity is reached are limited, as follows:

- The design flow rate only occurs in the hours between 10:00 am and 2:00 pm, with the direct normal radiation at, or above, the design value
- The highest velocities occur following a receiver trip, but the number of trips each year is generally limited to at most several.

Turn Down Ratio

To capture as much of the annual site radiation as possible, the receiver would, under ideal conditions, be able to operate with an infinite turndown ratio. However, this is not practical, for the following reasons:

- At the minimum flow rate, the pressure drop across the panel must be high enough to uniformly distribute the flow among the 40 to 60 parallel tubes
- In the downflow panels, the pressure drop in each tube must be high enough to prevent flow reversals promoted by buoyancy effects.

In many commercial designs, the maximum turn down ratio is about 6:1.

On a related point, the power incident on the receiver is a function of the season and the time of the day. On a winter morning, the Northwest quadrant of the receiver absorbs a much higher power than the Southeast quadrant. To help balance the powers among the 4 quadrants, 2 flow circuits are typically used. In one circuit, the flow enters on the North side, passes through the panels of the Northwest quadrant, crosses over to the East side, and then passes through the panels of the Southeast quadrant.

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The other flow circuit is a mirror image of the first. This flow arrangement allows the receiver to operate above the maximum turn down ratio for a higher percentage of the operating hours each year.

Vessel Capacities

Commercial receiver designs typically use an inlet vessel and an outlet vessel.

Inlet Vessel

In the event of a loss of electric power to the receiver pump drives, the heliostat field must be defocused immediately to prevent permanent damage to the panels. The field is defocused by moving the heliostat aim points from the panels to a group of imaginary points in the air space near the receiver.

The time required to move the heliostats is on the order of 20 seconds. Should site power be lost, an additional 30 seconds is required to start a standby Diesel-generator, establish the normal operating generator speed, and cycle the switchgear through the field transformers supplying electric power to the heliostat drives.

To supply an emergency flow of salt to the receiver during a 50 second defocus period, a receiver inlet vessel is provided. The vessel contains a sufficient volume of salt to supply the receiver, at the design flow rate, for a nominal period of 60 seconds. Above the salt inventory is an space, filled with compressed air. The pressure and volume of the air space are selected such that the flow of salt, during the heliostat defocus period, is adequate to protect the receiver from damage due to overheating.

Outlet Vessel

The outlet vessel is located above the top of the receiver panels, and provides a level signal to the throttle valves at the base of the downcomer. This arrangement ensures that both the receiver and the downcomer are always flooded.

The throttle valves at the base of the downcomer are typically specified as fail open. However, it is possible for the valves to move to the closed position if the operator places the valves in manual, and issues a closed command. A high-high, or a high-high-high, level in the outlet vessel will initiate a trip of the receiver, and, in turn, a trip of the heliostat field. The salt inventory in the inlet vessel will pass through the receiver to the outlet vessel. To prevent the outlet vessel from overflowing, the inventory of the outlet vessel must be the sum of the inventory at the trip condition, plus the inventory of the inlet vessel.

Receiver Vent Line

The outlet vessel is vented to the atmosphere, for the purposes of the following:

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- Prevent a backpressure on the flow entering the receiver, which could restrict the flow and damage the receiver due to overheating
- Prevent a backpressure on the receiver, which could cause the tubes to exceed the maximum allowable working pressure.

With this arrangement, it is possible to the receiver to be in normal operation, and for the operator to simultaneously close the throttle valves at the base of the downcomer. Within a period of 1 to several minutes, the available spare volume in the outlet vessel will flood with salt, and the vessel will start to overflow. Salt flowing on to the top of the receiver will likely result in damage to any electric equipment inside the receiver, and poses a danger to the operating personnel on the ground.

To avoid this situation, the receiver uses a vent line. The vent line connects the top of the outlet vessel with the top of the hot salt tank. This ensures that the outlet vessel is vented to the atmosphere, and any salt overflowing the vessel is captured in the hot tank. Due to Code requirements on the allowable operating pressure of the outlet vessel, the vent line must be open and it can have no internal restrictions; i.e., valves.

Although the vent is simply an open pipe, the design requirements are stringent. Salt entering the vertical line will accelerate due to gravity. Since the pressure loss is less than 1 m of head per meter of vertical length, the flow will quickly transition to two-phase flow. Further, once the flow reaches the first elbow, a portion of the kinetic energy will be converted to momentum forces on the pipe supports and anchors. The momentum forces, which will be much higher than those associated with single-phase flow, will be difficult to predict. Further, the two-phase nature of the flow can lead to oscillations in the loads at the elbows, pipe vibrations, and fatigue damage to the pipe, supports, and anchors.

Thermal Efficiency

Unlike the receivers in parabolic trough projects, the receiver in a central receiver project is exposed directly to the environment; i.e., there is no vacuum jacket around the receiver tube to reduce convection heat transfer. As such, the receiver uses the following approaches to reach efficiencies as high as practical:

- Reduce the absorber area. The reradiation and the convection losses are, to a first order, proportional to the area. The area is reduced until the tube strains produced by the incident flux result in tube fatigue life equal to the life of the project (i.e., 30 years).
- Use a selective surface coating on the tubes. The most common coating is Pyromark, which provides a nominal absorptivity, when new, of about 96 percent.

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At the design point, the receiver efficiency is on the order of 88 percent. Additional details on the receiver efficiency, over a range of operating conditions, are presented in Section 7.1.14.

Rates of Temperature Change

The largest temperature transient in the receiver occurs under the following conditions:

- The receiver is operating at or near the design power, with an outlet temperature of 565 °C
- A trip occurs in either the receiver or the heliostat field. The heliostat field defocuses in 20 to 50 seconds, and the incident power on the receiver decays to zero in this period
- Since the receiver is operating at or near the design power, the ullage pressure in the inlet vessel is at or near the design value. In response to the trip signal, the circuit inlet flow control valves move to the full open position. This maximizes the flow to the receiver during the period in which the heliostats are moving to the defocus positions. The combination of the design ullage pressure in the inlet vessel and the fully open control valves results in a receiver flow rate equal to, or slightly greater than, the design value
- The thermal mass of the metal in the receiver (mass * specific heat) is extremely small relative to the heat transfer capability of the salt moving through the receiver (mass flow rate * specific heat). As a result, following a trip, the rate of change in the metal temperature is on the order of 360 °C/min.

Receiver Panels

All of the pressure parts in the receiver panels are designed to accommodate a rate of change of 360 °C/min. The most problematic locations are the following:

- Where there is a step change in the metal thickness; i.e., the transition from the panel header to the panel tube
- Where there is a step change in the tube wall thickness; i.e., the support for the tube clip welded to the back of the tube wall.

Outlet Vessel

The flow entering the outlet vessel will experience a rate of temperature change of 360 °C/min. However, a rate of change in the metal temperature of 360 °C/min has the potential to damage a piece of equipment as large as the outlet vessel. As such, some means of flow distribution is provided to

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promote mixing with the vessel inventory, and limit the rate of temperature change in the metal temperature.

Fatigue Life

During the 30-year life of the project, the receiver operates under the following range of cyclic conditions:

- 12,000 full-range startup and shutdown cycles, in which the metal temperature at the cold end of the receiver changes from ambient to 295 °C, and the metal temperature at the hot end of the receiver changes from ambient to 565 °C. Once the receiver is filled and flowing with cold salt, the rate of temperature change at the hot end of the receiver is on the order of 0.5 °C/min
- 20,000 partial-range cloud transient cycles, in which the metal temperature at the cold end of the receiver remains constant at 295 °C, and the metal temperature at the hot end of the receiver changes from 295 °C to 565 °C. The rate of temperature change at the hot end of the receiver is on the order of 0.5 °C/min
- 150 trip cycles, in which the metal temperature at the cold end of the receiver remains constant at 295 °C, and the metal temperature at the hot end of the receiver changes from 565 °C to 295 °C. The rate of temperature change at the hot end of the receiver is approximately 360 °C/min.

The receiver is designed to withstand the cumulative creep and fatigue damage associated with these cycles.

5.1.2 Equipment Design

Materials

Tubes

The tube material of choice in commercial project is Alloy 230. This is also known as Haynes 230 if the material is supplied by Hayes International. The alloy is a solid solution of nickel, chromium, tungsten, molybdenum, cobalt, and various trace elements. The receiver panels at Gemasolar, Crescent Dunes, Cerro Dominador, and Noor III were fabricated with Alloy 230.

The other candidate material is Inconel 625LCF. This is also known as Haynes 625SQ if the material is supplied by Haynes International. The LCF stands for low cycle fatigue, and the alloy is essentially Inconel 625 with various trace elements removed to improve the fatigue properties. An advanced receiver panel, fabricated from Inconel 625LCF, was installed and operated at the Solar Two demonstration project.

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Both materials offer a favorable combination of corrosion resistance and the ability to accommodate demanding combinations of creep and fatigue damage. However, Inconel 625LCF, on the time span of years, forms dendrites of intermetallic compounds. This has the effect of strengthening the material, but reducing the ductility.

Headers

The upper and the lower headers for the panels are typically fabricated from Type 347H stainless steel. Stainless steel is not required for the panels near the cold end of the receiver. However, stainless steel is often used for all of the panels to provide a common fabrication approach and to provide interchangeability among the panels.

To help accommodate the differences in the coefficients of thermal expansion between Alloy 230 and Type 347H stainless steel, a transition piece can be installed between the end of the tubes and the ends of the nozzles on the headers. A representative material is Inconel 825.

Redundancy

Commercial receiver designs have little redundancy, as follows:

- There is one inlet vessel and one outlet vessel
- There are two flow circuits, operating in parallel. However, both flow circuits must be in service for the receiver to operate, and the multiple circuits do not provide a level of redundancy
- There are typically 16 receiver panels. However, all of the panels must be in service for the receiver to operate, and the multiple panels do not provide a level of redundancy
- There is one flow control valve for each receiver circuit, one vent valve per pair of panels, and there is one drain valve per pair of panels. All of the valves must be in service for the receiver to operate.

The only redundancy provided is generally limited to thermocouples and to heat trace circuits.

5.2 Thermal Storage System

The thermal storage system includes the cold salt tank, the hot salt tank, and the salt inventory. The mass of the salt inventory, and thereby the volumes of the tanks, is largely defined by the number of hours in which the Rankine cycle is required to operate, at full load, from the storage system.

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5.2.1 Performance Requirements

Cold Salt Tank

The cold salt tank performs 5 primary functions:

- Provides a volume sufficient to store the entire salt inventory of the plant
- Receives salt, at a nominal temperature of 295 °C, from the steam generator
- Receives salt, at temperatures in the range of 295 to 520 °C, from the receiver during receiver startup
- Provides a source of cold salt to the receiver
- Provides a source of cold salt to the steam generator during startup and shut down.

The normal operating temperature of the tank is the temperature of the salt at the cold end of the steam generator preheater (295 to 305 °C, depending on the Rankine cycle design). However, the design temperature of the tank is typically selected to be 370 °C. This allows the temperature of inventory to increase by as much as 75 °C during extended receiver startup periods typical of cloudy weather.

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². The insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

- Limit the heat losses from the floor of the tank to values in the range of 60 to 75 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Hot Salt Tank

The hot salt tank performs 4 primary functions:

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- Provides a volume sufficient to store the entire salt inventory of the plant
- Receives salt, at a nominal temperature of 565 °C, from the receiver
- Receives salt, at temperatures in the range of 450 to 565 °C, from the receiver during receiver startup and following a receiver trip. This requirement is notably quite stringent. The tank inlet distribution system must provide sufficient mixing with the bulk inventory to prevent potentially damaging values of intra-tank temperature differentials. The differentials may only persist for short periods (seconds), but it is the instantaneous rates of temperature change that lead to low cycle fatigue damage
- Provides a source of hot salt to the steam generator.

The normal operating temperature of the tank is the temperature of the salt at the hot end of the receiver (565 °C). However, the design temperature of the tank is typically selected to be the high-high trip temperature for the receiver (600 °C).

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/m². As with the cold tank, the insulation types and thicknesses are based on an economic optimization comparing the capital cost of the insulation with the capital cost of the additional heliostats required to compensate for the heat losses through the insulation.

The foundation of the tank is provided with thermal insulation, and with cooling beneath the foundation. The goals of the foundation design include the following:

- Limit the heat losses from the floor of the tank to values in the range of 70 to 85 W/m²
- Limit the temperature of the soil directly beneath the foundation to values no higher than 75 °C. At this temperature, desiccation of the soil, and oxidation of the organic materials in the soil, are controlled to values which limit the potential for differential tank settlement.

Inlet Flow Distribution System

Distribution Ring Header

In essentially all commercial projects, the flow is introduced into the cold tank and the hot tank by a circular or an orthogonal ring header located just above the floor. The ring is centered in the tank, and the diameter of the ring is approximately 50 percent of the diameter of the tank. A series of holes or mixing eductors are located along the circumference of the ring to promote mixing between the incoming flow and the bulk inventory.

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The design is simple and inexpensive. However, it is not particularly effective in promoting mixing, for the following reasons:

- Mixing is a function of the fluid velocity leaving the distribution header. At low flow rates, effective mixing occurs only within some limited distance from the header. In a representative commercial project, the diameter of the tank is 44 m, the diameter of the header ring is 21 m, and the diameter of the header pipe is 0.6 m. As such, effective mixing at low flow rates only occurs over perhaps 30 to 50 percent of the surface area of the floor
- During receiver startup, the inventory level in the hot tank is typically near the minimum level. Further, the temperature of the incoming flow is at least 20 °C, and may be as much as 100 °C, lower than the temperature of the bulk inventory. Due to buoyancy effects, the incoming salt will largely descend towards the floor. If the depth of the inventory is low, then the temperature of the incoming salt may only increase by a small amount before the flow reaches the floor. This can result in local, and potentially damaging, thermal stresses due to temperature gradients.

It can be noted that the problems with buoyancy in the hot tank are generally not replicated in the cold tank. During receiver startup, salt, at temperatures in the range of 295 °C to perhaps 510 °C, are introduced into the cold tank. However, this typically has moderate effects on the tank, as follows:

- During morning startup, the inventory level in the cold tank is typically full, or close to full. As such, the thermal inertia of the inventory (mass * specific heat) is high, and the inventory can accept salt at relatively high temperatures without a significant change in the inventory temperature
- Since the initial temperature of the bulk inventory is close to 295 °C, the flow of relatively hot salt rises into the bulk inventory. This has the effect of damping the transient temperature changes imposed on the floor and the wall.

Ring Header Supports

In all commercial projects, there is a mechanism for supporting the distribution header. This typically done with vertical pipe supports, approximately 0.5 to 1 m high, located between the floor and the header.

In some projects, the supports are welded to the floor. However, as noted above, when the temperature of the incoming flow is different than the temperature of the bulk inventory, differential thermal expansion between the ring header and the floor will place a combination of shear and bending loads on the floor. The floor is typically thin (6 to 8 mm), and the ability of the floor to resist bending loads, without deformation, may be limited. To avoid this condition, reinforcing plates can be installed between the bottom of the pipe supports and the top of the floor. However, there is now a step change in

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the thickness of the floor, and this location can act as a stress concentration when the tank undergoes daily thermal expansion and contraction cycles. The problem can be compounded if the welds for the reinforcement plates overlap with the welds for the floor plates.

Alternately, in other projects, the supports are placed on, but are not welded to, the floor. This eliminates the potential for the supports to impose bending loads on the floor. However, erosion has been observed at the bottom of the pipe supports. This is likely due to movement of the ring header associated with the following:

- Daily thermal expansion and contraction cycles
- Two-phase flow in the vertical line between the top of the tank and the inlet to the ring header. At essentially all flow rates, the pressure drop in the vertical line will be less than 1 m of head per meter of change in elevation. This will produce a two-phase flow, with the potential for oscillations in the flow rate of liquid entering the ring header and vibrations in the line.

In future commercial projects, alternate approaches to flow distribution should be adopted. Potential candidates include the following:

- Provide multiple distribution ring headers, spanning a larger fraction of the floor surface area
- In the hot salt tank, introduce salt at locations inside the roof or near the top of the wall. This helps to isolate receiver startup transients from the floor, and eliminates the potential of the sharing of forces between the header supports and the floor. The ring header would be replaced with an array of small spray nozzles to avoid problems with trying to support a single, large ring header from the roof or from the top of the wall. A couple of notes: 1) this approach to flow distribution in the hot tank was demonstrated at the Solar Two project, and 2) the approach is likely limited to the hot tank; i.e., introducing high temperature salt (500 °C) in the cold tank during receiver transients could expose the tank wall to potentially high temperatures.
- Clearly, introducing salt at the top of the inventory in the hot tank will produce a vertical stratification in the inventory due to 1) a permanent layer of stagnant salt located below the elevation of the suction bells for the salt pumps, and 2) continuous transfer of heat into the foundation. To prevent or to remove the stratification, some form of bulk mixing will be required. Candidates methods might include the following:
 - Pump recirculation, which supply salt to either a conventional distribution ring or a grid of injection points near the floor
 - Gas compressors, which draw suction from the ullage space in the tank and supply the (slightly compressed) ullage gas to a grid of injection points near the floor

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- Tank-scale mixing devices, such as a large vertical plate rotating about a vertical axis in the tank, or a large horizontal perforated plate, which vertically traverses the inventory.
- It can be noted that gas reinjection and tank-scale mixing devices have yet to be demonstrated at a commercial scale.

5.2.2 Equipment Design

In commercial projects, salt inventories typically range from 20,000 to 40,000 metric tons. Since the vapor pressure of salt is very low (several Pa), the lowest cost storage tank is a flat bottom design with a self-supporting dome roof. The tank is vented to the atmosphere.

At projects sites representative of commercial projects, the allowable soil bearing load is on the order of 240 kPa (5,000 lb_f/ft²). The lowest cost foundation design is one in which the tank is placed directly on compacted, but parent, soil. Given the density of the salt (1,900 kg/m³), the tallest column of salt which can be supported by the parent soil is about 12.7 m. This, in turn, effectively sets a nominal limit on the allowable wall height of 12.5 m. To store the required salt inventory, the tank diameter is selected accordingly.

To a first order, the section thicknesses of the wall courses can be calculated from the hoop stress formula (Thickness = Pressure * Diameter / (2 * Allowable stress)) and the allowable stresses listed in Section II of the Code. For the stainless steel used in the hot salt tank, there are no fabrication restrictions noted in the Code. However, for the carbon steel used in the cold salt tank, a post weld heat treatment is required for section thicknesses greater than 38 mm (1.5 in.). Specialty contractors can provide heat treatment services, but the process is not without risks:

- The treatment temperature is high enough (650 °C) that the strength of the material decreases to the point where creep deformations are possible
- The iron in the steel can chemically reduce the carbon dioxide in the air. Carbon is infused into the surface of the metal, which increases the strength, but reduces, the ductility of the material.

If the salt inventory requires a cold tank wall thickness greater than 38 mm, but the engineering contractor would like to avoid the need for heat treatments, then the inventory can be divided into two 50-percent capacity tanks to reduce the wall thickness to a value less than 38 mm.

It can be noted that the engineering contractor can determine that the operating conditions of the cold tank are outside of the limits specified in API Standard 650. As such, the cold tank is not, strictly speaking, an API Standard 650 tank, but the tank can be designed based on API Standard 650 with exceptions. If so, the requirement for the post weld heat treatment of plate thicknesses greater than 38 mm can be avoided. Naturally, this design approach would require the concurrence of the Owner.

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5.3 *Steam Generation System*

The steam generator converts the thermal energy in the hot salt to a combination of live steam and reheat steam that is supplied to the Rankine cycle for power generation.

5.3.1 **Process Requirements**

The steam generator receives a flow of feedwater from the Rankine cycle. The feedwater is at a pressure of 130 to 145 bar, and at a temperature of 220 to 245 °C, depending on the cycle design. The nominal flow rate is 1 kg/sec per MWe of electric power production.

The steam generator preheats the feedwater to the saturation temperature, converts the feedwater to saturated steam, and superheats the steam to a nominal temperature of 540 °C.

A portion of the saturated water (~ 1 percent) is removed from the evaporator section for water chemistry control.

After the live steam has expanded in the high pressure steam turbine by a ratio of 4 to 5, the steam is returned to the steam generator for reheating. The steam is reheated to a nominal temperature of 540 °C, and then expanded by a factor of about 200 through the intermediate- and the low-pressure sections of the turbine.

Daily Startup and Shutdown

The steam generator operates in a daily cycle, as follows:

- During overnight hold, cold salt from the attemperation pump flows through each of the heat exchangers. The cold salt is returned to the cold tank
- In the first phase of startup, hot salt is mixed with cold salt at a point upstream of the hot end of the evaporator. This establishes a flow of saturated steam in the superheater and in the reheater equal to about 20 minimum of the design flow rate. The flow rate is slightly greater than the minimum allowable flow rate specified by the vendor (16 percent). No heat transfer occurs in either the superheater or the reheater during this phase
- In the second phase of startup, the relative flow rates of hot salt and cold salt remain fixed, and the salt mixing station is moved from a point upstream of the hot end of the evaporator to the normal point upstream of the hot ends of the superheater / reheater. Superheating now occurs in the superheater and in the reheater. Due to the superheating duties, the steam flow rate decreases to 16 percent

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- The relative proportions of hot salt and cold salt are adjusted by changing the relative speeds of the hot salt pump and the attemperation pump. When the flow of cold salt has reached a value of zero, the steam generator is in normal operation
- The process is essentially reversed to shut down the steam generator.

Minimum Flow Rate

During the startup and the shutdown process, the flow rates on both the shell-sides and the tube-sides of the heat exchangers must satisfy the minimum allowable flow rates (a nominal 16 percent) specified by the vendor.

The flows on the shell sides of the heat exchanger can satisfy the vendor requirement by setting the sum of the hot salt flow rate and the cold salt flow rate to a value that is always equal to, or greater, than the vendor limit. If the evaporator is a forced recirculation design, then the flows on the tube sides of the preheater and the evaporator can satisfy the vendor limits, as follows:

- A preheater recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the preheater. The combination of the feedwater flow rate plus the recirculation flow rate always satisfies the vendor requirement
- An evaporator recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the evaporator. The combination of the feedwater flow rate plus the recirculation flow rate always satisfies the vendor requirement
- During startup, there is no mechanism to recirculate steam in the superheater and the reheater. As such, the steam flow rate necessarily passes through the range of 0 percent to the vendor minimum of 16 percent. At the flow rates in this range, the steam is unlikely to be uniformly distributed among the several hundred tubes in each heat exchanger. Non-uniform flow distributions can produce non-uniform temperature distributions, which, in turn, can lead to unpredictable stress distributions. However, if the superheater and the reheater are operated with the same temperatures on the tube side and the shell side, then no heat transfer occurs and potentially damaging stress distributions can be avoided. As noted above, the use of a salt mixing station upstream of the hot end of the evaporator allows the minimum tube-side flow requirement to be met, at which point the mixing station can be safely moved to the normal point upstream of the superheater / reheater.

On a related point, the duty of the steam generator necessarily passes through the range of 0 percent to 16 percent during both startup and shut down. Establishing the minimum allowable tube- and shell-side

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flow rates, or establishing isothermal operation, throughout the duty range of 0 percent to 16 percent helps to protect the heat exchangers from unpredictable, and potentially damaging, stress distributions.

Maximum Allowable Rate of Temperature Change

The heat exchanger consists of a number of parts which are joined either by welding or by plastic deformation. In addition, the parts generally span a wide range of section thicknesses, ranging from less than 2 mm for the tube walls to perhaps 200 mm for the tubesheet. During transient conditions, the thin metal sections change temperature more quickly than the thick metal sections. The different response times produce different rates of thermal expansion, which lead to local thermal stresses. The transient stresses are often additive to the normal process stresses due to temperature and pressure.

The sum of the normal process stresses and the transient thermal stresses can exceed the allowable stress values listed in Section II of the Code. As such, the fatigue life of the heat exchanger can be less than infinite. To maintain a low cycle fatigue life consistent the life of the project, the vendor will define a maximum allowable rate of temperature change. Typical values range from 8 to 12 °C/min.

It can be noted that operating the heat exchangers at rates of temperature change higher than the vendor's limit will produce transient stresses which are above those than consistent with a fatigue life of 30 years. Further, the decrease in the fatigue life associated with an increase in the transient thermal stress is exponential, and even small excursions in the rate of temperature change can significantly reduce the fatigue life of the equipment.

5.3.2 Performance Requirements

The steam generator must meet the following performance requirements:

- The saturated steam from the drum complies with the steam quality standards set by the turbine vendor
- The heat exchanger designs meet the Tubular Equipment Manufacturers Association requirements for items such as flow distribution, tube vibration, and tubesheet temperature gradients, and allowances for spare tubes
- The heat exchangers have a fatigue life consistent with at least 10,000 startup and shutdown cycles (i.e., 30 years * 330 cycles per year).

At some commercial projects, it has proven difficult to meet the vendor's limit on the allowable rate of temperature change startup and shutdown. To provide a margin against potentially inaccurate process temperature control, it may be beneficial to provide a fatigue life of 50,000 to 100,000 cycles for the heat exchangers.

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5.3.3 Equipment Design

All commercial steam generator designs are based on a common arrangement, as follows:

- The evaporator uses natural or forced recirculation, in combination with a separate steam drum. A solar project necessarily cycles each day, and the condensate water chemistry in the condenser hotwell typically degrades during the overnight hold period. Reestablishing the water chemistry during morning startup for a drum-type evaporator is less complex and less time consuming than reestablishing the more stringent water chemistry required for a single-pass-to-superheat steam generator
- A preheat heat exchanger, separate from and upstream of the water side of the evaporator, performs two functions:
 - Raises the temperature of the feedwater close to the saturation point, which allows the evaporator to operate under isothermal conditions on the water side
 - Reduces the temperature of the salt leaving the steam generator to a value which is less than the saturation temperature, which effectively reduces the cost of the thermal storage system
- The cold reheat steam temperature (285 to 300 °C) is the same order of magnitude as the saturation temperature in the evaporator (335 to 350 °C). As such, the temperature of the salt at the cold end of the reheater is the same order of magnitude as the temperature of the salt at the cold end of the superheater. As a consequence, the salt flow from the cold end of the reheater can be combined with the salt flow from the cold end of the superheater, and the combined flow can be directed, in series, through the evaporator and the preheater. In this manner, energy can be extracted from the salt flow through the reheater to the same extent as energy can be extracted from the salt flow through the superheater. This, in turn, provides the lowest cold salt temperature of any of the candidate steam generator configurations
- On the salt side of the steam generator, the superheater operates in parallel with the reheater. With this arrangement, the hot reheat steam temperature can be as high as the live steam temperature. This, in turn, maximizes the live steam pressure (up to the point where the moisture content in the turbine exhaust reaches maximum allowable values of 8 to 9 percent). Since the unit work of the steam passing through the turbine is $\int (\text{Specific volume}) * d(\text{Pressure})$, maximizing the live steam pressure maximizes the cycle efficiency.

In general, the shell-side fluid is selected to be salt, and the tube-side fluid is selected to be water/steam. This arrangement is adopted for two reasons:

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- Should the salt freeze in a heat exchanger, the thawing process will involve a change in the salt volume of about 4.6 percent. Were the salt to be constrained in a tube or in a vessel, the change in tube or vessel diameter would be on the order of 2.3 percent. This is beyond the yield strain for both carbon and stainless steels, and plastic deformations of this amount can significantly reduce the fatigue life of the equipment. One method to reduce the potential for damage during thawing is to place the salt on the shell side. The thawing process would involve the following steps:
 - Raise the temperature of the salt lines to and from the vessel to at least 275 C. This ensures the salt in the lines is liquid, and any salt melted in the exchanger has an exit path from the vessel
 - Activate the electric heat tracing on the shell. Radial heat transfer into the shell will melt the salt on the inside surface of the vessel, and the melting salt has an exit path from the heat exchanger through the salt lines
 - Simultaneously supply heat to the tube bundle by means of the electric water heaters. In the preheater and in the evaporator, the heat will be supplied by the recirculation of water. In the superheater and in the reheater, the heat will be supplied by a flow of saturated steam from the drum. The temperature on the water side must be maintained within a few degrees of the temperature on the outside of the shell. This is to prevent the establishment of radial temperature gradients in the tubesheet that could either damage the tubesheet or flex the tubesheet to the extent that the tube-to-tubesheet connections could relax
- The water/steam operates at much higher pressures than the salt, and thin wall tubes are more adept at withstanding high pressures than thick wall vessels.

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6. Functional / Prescriptive Specifications

6.1 Introduction

Plant and equipment specifications fall into two broad categories:

- Functional specifications describe what the system or the equipment needs to do, consistent with the minimum legal requirements of the local jurisdictions. The details of how this is to be accomplished is developed by the engineering contractor. This allows the engineering contractor to define the process and the equipment designs such that the functional requirements can be met at the lowest cost
- Prescriptive specifications, which are developed by the Owner, prescribe to the engineering contractor how the functional requirements of the project are to be met. Prescriptive elements include items such as the process design, Code Section selection, fabrication techniques, materials selection, and component details. This arrangement ensures that the favorable experience from a previous project is repeated, and helps to avoid situations in which an inexperienced contractor repeats mistakes from earlier projects.

It can be noted that the use of prescriptive specifications often requires the following:

- The Owner must have a thorough understanding of the process and the technical features of the plant. For example, the heat trace designs in commercial projects often use 1 dual-element thermocouple in each zone as the input to the temperature set point controller. However, a heat trace design, which reflects the experience from previous projects, might use 2 to 4 thermocouples, distributed along the length of the zone, as inputs to the controller. This arrangement can help to detect defects in the insulation, local areas with temperatures below the freezing point of the salt, local areas which are receiving too much heat input from adjacent zones, and leakage past isolation valves.
- The Owner is prepared to accept what is likely to be an increase in the capital cost as a means of achieving the goals set in the performance and in the financial models. For example, the Owner may require the use of all-welded construction in salt heat exchangers. This fabrication approach is more expensive than the conventional approach; i.e., tube rolling followed by strength welding of the tube to the tubesheet. However, an all-welded design is more tolerant of the transient stresses associated with 1) daily startup and shutdown, and 2) mistakes made by the operators. In turn, the improvement in plant availability more than compensates for the marginal increase in the cost of the heat exchangers due to an all-welded design.

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In the sections below is a list of the Design Codes and Standards, followed by a discussion of recommended prescriptive specifications.

6.2 *Design Codes and Standards*

Reference codes and standards are listed below for the US market. If the plant is constructed outside of the US, an equivalent matrix will need to be developed for the host country.

<u>Designation</u>	<u>Title</u>
API	Standard 650, Welded Steel Tanks for Oil Storage
ASME	B31.1, Power Piping
ASME	Section I, Rules for the Construction of Power Boilers
ASME	Section II, Materials
ASME	Section III, Division 1, Subsection NH, Class 1 Components in Elevated Temperature Service
ASME	Section V, Non Destructive Examination
ASME	Section VIII, Division 1, Rules for the Construction of Pressure Vessels
ASME	Section VIII, Division 2, Alternative Rules for the Construction of Pressure Vessels
ASTM	A105, Specification for Forgings, Carbon Steel, for Piping Components
ASTM	A181, Specification for Forgings, Carbon Steel for General Service
ASTM	A182, Specification for Forged or Rolled Alloy-Steel Pipe Flanges, Forged Fittings, and Valves and Parts for High-Temperature Service
ASTM	A192, Specification for Seamless Carbon Steel Boiler Tubes for High-Pressure Service
ASTM	A193, Specification for Alloy-Steel and Stainless Steel Bolting Materials for High-Temperature Service
ASTM	A194, Specification for Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service
ASTM	A213, Specification for Seamless Ferritic and Austenitic Alloy-Steel Boiler, Superheater, and Heat Exchanger Tubing
ASTM	A216, Specification for Steel Castings, Carbon, Suitable for Fusion Welding, for High-Temperature Service
ASTM	A240, Specification for Heat-Resisting Chromium and Chromium-Nickel Stainless Steel Plate, Sheet, and Strip for Pressure Vessels
ASTM	A249, Specification for Welded Austenitic Steel Boiler, Superheater, Heat Exchanger, and Condenser Tubes
ASTM	A312, Specification for Seamless and Welded Austenitic Stainless Steel Pipe
ASTM	A325, Specification for Structural Steel Bolts, Steel, Heat Treated, 120/125 ksi Minimum Tensile Strength
ASTM	A351, Specification for Castings Austenitic Austenitic-Ferritic (Duplex) for Pressure-Containing Parts

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ASTM	A36, Specification for Carbon Structural Steel
ASTM	A387, Specification for Pressure Vessel Plates, Alloy Steel, Chromium-Molybdenum
ASTM	A403, Specification for Wrought Austenitic Stainless Steel Piping Fittings
ASTM	A500, Specification for Cold-Formed Welded and Seamless Carbon Steel Structural Tubing in Rounds and Shapes
ASTM	A506, Specification for Steel, Sheet and Strip, Alloy, Hot-Rolled and Cold-Rolled, Regular Quality and Structural Quality
ASTM	A516, Specification for Pressure Vessel Plates, Carbon Steel, for Moderate- and Lower-Temperature Service
ASTM	A53, Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated Welded and Seamless
ASTM	A556, Specification for Seamless Cold-Drawn Carbon Steel Feedwater Heater Tubes
ASTM	B443, Specification for Nickel-Chromium-Molybdenum-Columbium Alloy (UNS NO6625) Plate, Sheet and Strip
ASTM	B444, Specification for Nickel-Chromium-Molybdenum-Columbium Alloy (UNS NO6625) Pipe and Tube
ASTM	B446, Specification for Nickel-Chromium-Molybdenum-Columbium Alloy (UNS NO6625) Rod and Bar
NEC	National Electric Code
NEMA	National Electrical Manufacturers Association
NFPA	National Electric Code (NEC), National Fire Protection Association (NFPA)
TEMA	Tubular Exchanger Manufacturers Association, 8th Edition TEMA Standards
UBC	Uniform Building Code

6.3 *Nitrate Salt Specification*

6.3.1 **Grades**

The component salts are available in refined, technical, and industrial grades. The refined grades have the lowest impurities and the highest cost, while the industrial grades have the highest impurities and the lowest cost. The technical grades are generally suitable for central receiver applications, offering an acceptable compromise between purity and cost.

A typical technical grade specification should include the following items:

- Minimum nitrate concentration of 99 percent by weight
- Maximum total chloride ion concentration from all sources of 0.2 percent by weight.

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6.3.2 Total Chloride Concentration

An example of the calculation of the total chloride ion concentration in a mixture is as follows: If the chloride ion concentrations in the sodium nitrate and potassium nitrate are 0.85 and 0.20 weight percent, respectively, then the chloride ion concentration in the mixture is $(0.6 \text{ weight fraction NaNO}_3)(0.85 \text{ percent}) + (0.4 \text{ weight fraction KNO}_3)(0.20 \text{ percent}) = 0.59 \text{ percent}$.

For this example, the 0.6 maximum weight percent chloride ion concentration for the nitrate salt mixture is satisfied.

For all of the compounds which contain chlorine, the chloride ion concentrations in weight percent shall be calculated from the weight percent of the compound in the nitrate salt, as follows:

$$\left[\frac{\text{Atomic Weight of Chlorine}}{\text{Molecular Weight of Compound}} \right] (\text{Weight Percent of Compound})$$

For example, a sodium chloride (NaCl) concentration of 0.9 weight percent is converted to a chloride ion concentration of 0.55 weight percent by multiplying the sodium chloride concentration by 0.607, as follows:

$$\left[\frac{35.457 \frac{\text{gm}}{\text{gm - mole}} \text{ for Cl}}{\left(22.997 \frac{\text{gm}}{\text{gm - mole}} \text{ for Na} + 35.457 \frac{\text{gm}}{\text{gm - mole}} \text{ for Cl} \right)} \right] (0.9 \text{ Percent})$$

Similarly, potassium perchlorate (KClO₄) concentration in weight percent shall be converted to chloride ion concentration in weight percent by multiplying the potassium perchlorate concentration by 0.256, and sodium perchlorate (NaClO₄) concentration in weight percent shall be converted to chloride ion concentration in weight percent by multiplying the sodium perchlorate concentration by 0.290.

6.3.3 Allowable Contaminants

Maximum contamination from all sources, by weight, will be:

- Nitrite: < 0.02 percent
- Carbonate: < 0.10 percent
- Sulfate: < 0.01 percent
- Hydroxyl alkalinity: < 0.20 percent
- Perchlorate: < 0.085 percent
- Magnesium: < 0.035 percent.

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The specification should also state that notification shall be provided for any contaminants not listed above which exceed a concentration of 0.04 percent by weight.

6.3.4 Supply Options

The salt may be supplied in one of two forms:

Option 1: The sodium nitrate is delivered in separate bags from the potassium nitrate. The project is responsible for mixing the two components in the required 60/40 mixture, melting the solid mixture, and loading in salt in the storage system

Option 2: The salt supplier mixes the components, melts the solid mixture, solidifies the liquid mixture in the form of a prill, and delivers the prills to the site. The project is responsible for melting the prills and loading the salt in the storage system.

Option 1 can offer a slight discount in the price of the salt. However, the material handling requirements and costs can be substantial. In addition, the project may need to process a one-time emission of oxides of nitrogen. Specifically, the non-synthetic grades of sodium salt contain trace amounts of magnesium. (Synthetic sources of sodium nitrate do not contain magnesium, and the reactions noted below do not need to be considered by the project.) Magnesium is soluble in nitrate salt, appearing as magnesium nitrate. Upon heating, the magnesium nitrate decomposes via the following irreversible reaction:



The magnesium oxide is insoluble in the salt, and forms a solid precipitate. The NOx has a very low solubility in the salt, and essentially all of the NOx appears as a gas in the ullage space above the liquid level. A typical commercial project might require 30,000 metric tons of salt. The magnesium concentration in the technical grade of the salt is a nominal 0.035 percent by weight, which translates to about 10,000 kg (410 kg mole) of magnesium. The decomposition reaction produces 2 moles of NOx per mole of magnesium, which results in the generation of about 38,000 kg of NOx. Some form of NOx capture and treatment system will be required to both meet environmental regulations and to ensure worker safety. The least complex approach is probably a water spray column, in which water reacts with NOx to form nitric acid, N2NO3. The acid is neutralized with a base, and then shipped offsite for disposal.

Under Option 2, the process of melting and mixing the components to form prills will promote the magnesium decomposition reaction. As such, the one-time production and treatment of NOx will be the responsibility of the salt supplier rather than the responsibility of the project. This division of responsibility will be reflected in the cost of the salt delivered to the project.

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As part of the procurement process, the engineering contractor will need to conduct an economic analysis to determine which of the two supply options offers the lowest total cost to the project.

6.3.5 Deviations from a 60/40 Mixture

Under either Option 1 or Option 2, the final composition is likely to vary slightly from the desired 60/40 mixture. In principle, property tables and relationships could be developed for the exact salt mixture at the project. However, the properties are not strongly influenced by mixture fractions, which are often in the range of 58/42 to 62/48 for commercial projects. An example is the melting point, as shown in Figure 6-1.

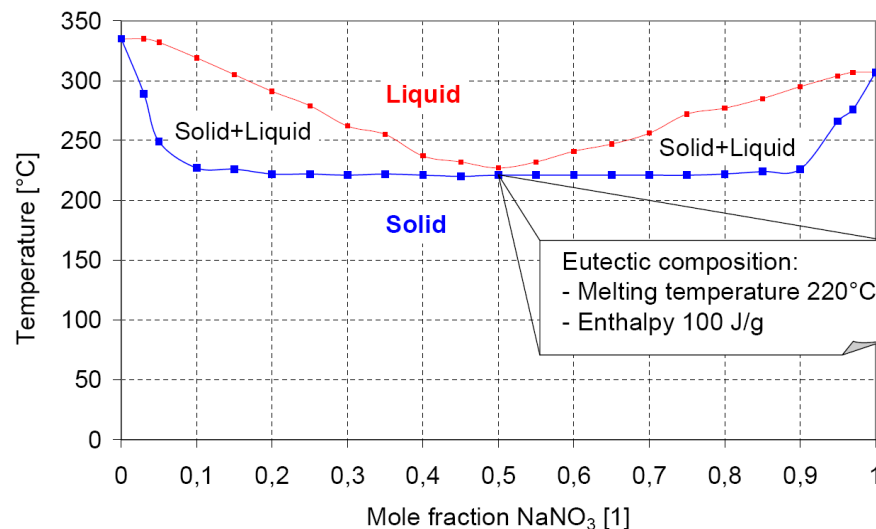


Figure 6-1 Melting Point of Binary Nitrate Salt Mixtures

Other properties, specifically the density, can be estimated by adjusting the 60/40 properties based on molar volumes, as described by Bradshaw³.

6.4 Nitrate Salt Handling and Melting Specification

Nitrate salt is hygroscopic, and absorbs moisture from the air during transit and when stockpiled for melting.

³ Bradshaw, R. W., (Sandia National Laboratories, Albuquerque, New Mexico), "Effect of Composition on the Density of Multi-Component Molten Nitrate Salts", Sandia Report SAND2009-8221, December 2009

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In trough projects, the design temperature of the hot salt tank is on the order of 395 °C. During the salt melting process, the temperature of the liquid salt is raised to something less than the design temperature of the hot tank; perhaps, 350 °C.

The salt, as delivered, contains 0.01 to 0.05 percent magnesium nitrate, depending on the purity. At temperatures of 550 °C and above, the magnesium nitrate, in a period of several days, decomposes via the following reaction:



The magnesium oxide precipitates as a solid, and the NO₂ and the O₂ leave the salt as gases.

The magnesium nitrate concentration is only a fraction of a percent; however, each mole of magnesium nitrate produces two moles of NO₂. For a plant with a storage mass of 35,000 metric tons, the total NO₂ production will range from 2,000 to 11,000 kg, depending on the purity of the salt.

The salt, during the melting process, is loaded into the storage tanks. Depending on the size of the inventory and the capacity of the melters, the melting process can take a few to several weeks. During the melting period, the temperature of the salt in the storage tanks is in the range of 275 to 350 °C, depending on the capacity of the tank electric heaters. As such, the magnesium decomposition rate is much slower than would occur at a temperature of 550 °C; certainly on the time span of months. Further, the water than is adsorbed by the salt in transit and during site storage does not immediately leave the salt after melting. Anecdotal evidence suggests that the time required to release the water from the salt is measured in weeks. As such, within the liquid inventory of the salt, NO₂ produced from magnesium decomposition comes into contact with H₂O released from the salt. The two combine to form nitric acid, H₂NO₃. The nitric acid is released from the salt inventory in the form of a gas. Further, the gas will condense on any surface with temperatures below about 80 to 90 °C. Since nitric acid has a dew point higher than water vapor, any warm surface on which the vapor condenses will essentially be pure acid. Condensed nitric acid samples collected from a commercial project during commissioning showed pH values in the range of 1 to 2. Nitric acid liquid in contact with any carbon steel in the storage system results in high corrosion rates, with the highest rates measured in mm per month.

The specification for salt handling and melting must include the following provisions:

- To prevent damage to the carbon steel equipment which is exposed to the ullage gas in the storage tanks, the equipment must be maintained at a temperature of at least 250 °C for a period of several months after the salt is loaded into the storage tanks. A temperature of 250 °C prevents the condensation of both nitric acid and salt vapors.
- If the equipment cannot be maintained at a temperature of 250 °C, then the equipment must be fabricated from a material which is resistant to nitric acid corrosion, such as stainless steel.

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6.5 General Salt System Equipment

6.5.1 Piping

Design Codes

The nitrate salt piping can be designed to the requirements of either ASME B31.1 - Power Piping or ASME B31.3 - Process Piping.

Receiver System

B31.1, compared with B31.3, is more conservative in terms of material specifications, stress intensity factors, stress design factors, and radiographic inspections. In the interests of providing reliable piping systems, at least two commercial central receiver projects have selected B31.1 as the design code for the salt piping in the receiver system. This includes the cold salt pump discharge piping, the riser, the inter-panel piping, the vent lines, the drain lines, and the downcomer.

Steam Generation System and Rankine Cycle

B31.1 is the required design specification for boiler external piping; i.e., the piping upstream of, or downstream from, a steam generator to the first flanged, welded, or threaded connection. As a practical matter, the following piping is also often designed to B31.1:

- The cold salt and the hot salt piping to, within, and from the steam generator
- The water/steam piping segments in and around the steam generator and the turbine-generator, which operate at combinations of high temperature and high pressure.

Materials

The cold salt piping is typically ASTM A106 Gr B or Gr C, with a nominal corrosion allowance of 1.6 mm (1/16 in.). The hot salt piping will be a 300-series stainless steel, with a nominal corrosion allowance of 0.7 mm (1/32 in.). The candidate materials include both L-grade materials (Type 304L and Type 316L), and H-grade materials (Type 304H, 316H, and 347H). Recent commercial projects have generally selected Type 347H, but this may not always be the preferred choice. A discussion of the advantages, and the disadvantages, of each option is presented in Section 8 of Volume 3 - Narrative.

Stainless Steel Material Specification

All stainless steel materials are to be supplied in a solution annealed condition.

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Stabilization Heat Treatment of Type 347H Piping

The pipe shall conform to ASME SA-312, Grade TP347H.

After solution annealing of the pipe, an additional stabilization heat treatment is to be performed per ASME SA-312 Supplementary Requirement S6. The stabilization process occurs at a temperature which is lower than the temperature which dissolves niobium carbide, but it is higher than the temperature which dissolves chromium carbides, such as Cr_{23}C_6 . The material is held at the stabilization condition for a period which is long enough to promote the formation of the desired niobium carbides at the expense of the undesired chromium carbides.

The stabilization steps are as follows:

- Raise the temperature to the stabilization temperature at a minimum rate of 280 °C/hour
- Hold at the stabilization temperature of 885 °C $\pm$ 14 °C for a minimum period of 2 hours
- Remove the pipe section from the heat source and allow the assembly to air cool to ambient conditions.

The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Stabilization Heat Treatment of Type 347H Wrought Fittings

All wrought stainless steel pipe fittings shall conform to ASME SA-403, WP347H, Class S (Seamless) per ASME SA-403.

After solution anneal of the fittings, an additional stabilization heat treatment shall be performed per ASME SA-403 Supplementary Requirement S2, except it shall be done in a temperature range of 845 to 870 °C and the fittings cooled in air.

The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Stabilization Heat Treatment of Type 347H Forged Flanges and Fittings

All forged stainless steel pipe flanges and fittings shall conform to ASME SA-182, F347H.

After solution anneal of the fittings, an additional stabilization heat treatment shall be performed per ASME SA-182 Supplementary Requirement S10, except it shall be done in a temperature range of 845 to 870 °C and the fittings cooled in air.

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The stabilization step may be performed during the ramp down cycle of the solution annealing process.

Post Weld Heat Treatment of Type 347H Pipe and Fittings

It is acceptable for the stabilized piping and fittings to receive an additional stabilization treatment during the post weld heat treatment of spool assemblies in the shop, and during the post weld heat treatment of the field weld that attach to these pipes and fittings.

It can be noted that post weld heat treatment of stabilized stainless steel pipe and fitting is not required by the ASME Code. The use of a post weld heat treatment is a decision to be made by the Owner as a means of reducing, but not eliminating, the potential for intergranular stress corrosion cracking in the heat affected zones of the welds. To make an informed decision regarding the use of post weld heat treatment, the Owner must, to some extent, be an expert in the Owner's technology.

Connections, Sizes, and Schedules

End connections for salt piping will be butt welded joints, except in the following locations:

- Salt pump discharge nozzles. Flanged connections using ring-type joints are permitted.
- Salt tank nozzle connections to the gas space inside the roof.

The minimum system piping diameter shall be DN 100 (4 in.) to reduce the potential for salt freezing due to damaged insulation or due to reduced heat trace capacity.

For both cold salt and hot salt service, the minimum schedule for DN 100 (4 in.) lines is Schedule 40. The goal is to provide sufficient bending stiffness to limit sagging and to prevent sections in the line which may not fully drain.

For lines larger than DN 100, the minimum Schedule is 20. The goal is to provide enough stiffness at the ends of pipe such that normal handling of the pipe does not cause the ends to become oval, which would make lineup and fitment prior to welding problematic.

All pipe will be specified as seamless to provide as much resistance to creep and fatigue damage as possible.

6.5.2 Flanges

Flanges in salt service may be of the following types: ring type joint; or hub type. No other types of flanges, such as raised face flanges, will be used due to the potential for relaxation of the bolted connection and leakage past the gasket due to the excellent wetting characteristics of the salt. Ring type

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joint gaskets and hub rings will be ASTM A182 Gr F21 for low temperature service, and ASTM A240 Gr 304 for high temperature service.

Stud bolts used at temperatures below 400 °C will be specified as ASTM A193 Gr 7. For temperatures above 400 °C, or pressure containing components with external heat tracing, stud bolts will be specified as ASTM A193 Gr B8R, with ASTM A194 Gr 8R heavy hex nuts.

6.5.3 Fittings

Butt weld fittings will be used in salt service in all pipe sizes.

The following fittings are not suitable for salt service:

- Compression fittings. Salt in contact with the threads and the ferrules will develop corrosion layers. Over time (months), the corrosion layers will merge, which will make disassembly of the fitting impossible
- Flat face and raised face flanges. The flanges rely on a gasket to provide the seal. However, salt is an excellent wetting agent, and no gasket material has yet been identified that is impervious to salt intrusion and seepage. Further, should salt come into contact with the flanges bolts and nuts, corrosion between the threaded regions will make disassembly impossible
- Socket welded connections. The wall thickness of the socket is the roughly the same as the wall thickness of the adjacent piping. As such, the combined wall thickness of the pipe / socket is nominally double the wall thickness of the piping. The plant operates under daily thermal cycles, with rates of temperature change as high as 360 °C/min. A pipe / socket combination subject to a thermal transient will establish a radial temperature gradient through the pipe / socket. The gradient will be the largest near the bottom of the socket due, in part, to the air gap between the outside of the pipe and the inside of the socket. The gradient will be the smallest at the weld connecting the outside of the pipe and the end of the socket. The cyclic radial temperature gradients will establish cyclic longitudinal gradient in the fitting, which will impose longitudinal sheer and bending stresses on the welds. Over time, this can result in a low cycle fatigue failure of the fitting weld.

6.5.4 Pipe Supports, Anchors, and Guides

Pipe supports, anchors, and guides are subjected to daily thermal expansion and contraction cycles.

At the Solar Two project, insulated pipe supports, using an external metal clamp with a calcium silicate insert between the pipe and the clamp, had an effective life of perhaps 1 year before the insert worked

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loose from the clamp. Alternate designs, using a radial fin to positively locate the insert within the clamp, also failed due to fracturing of the insert.

Following the Solar Two project, changes have been made to insulated pipe supports, including the use of refractory inserts with a higher crush strength. If a project plans to use insulated pipe supports, then surveys should be conducted of commercial projects where the proposed pipe supports are already in use. This will assist the engineering contractor in 1) determining which locations are suitable for insulated supports, and 2) specifying the refractory materials for the support.

Nonetheless, there are likely to be locations where insulated supports will not provide the required stiffness, strength, or cyclic lifetime in locations which are difficult to access (i.e., along the length of the downcomer). An alternate design for the supports, anchors, and guides uses a metal web, welded directly to the pipe. The web, in turn, is welded to a base plate. A portion of the web is removed to reduce conduction heat transfer through the web. The basic design is illustrated in Figure 6-2.

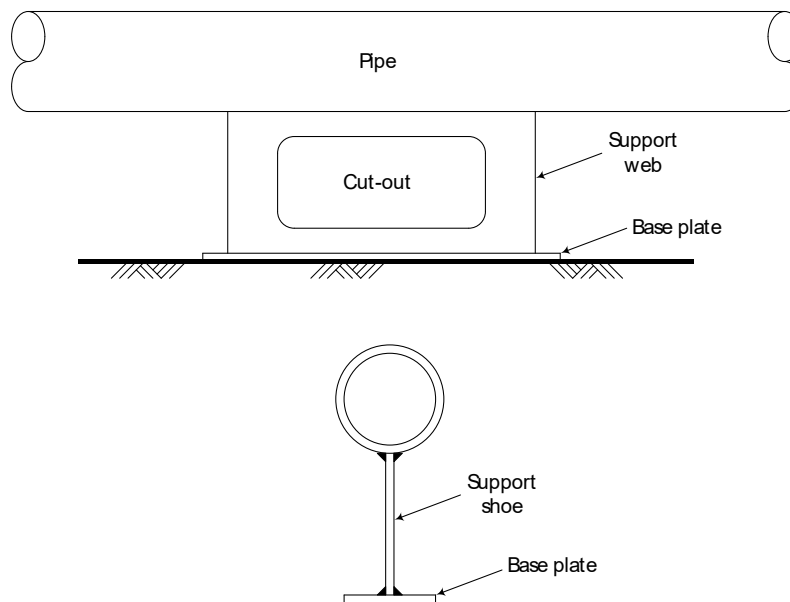


Figure 6-2 Pipe Support, Anchor, and Guide

The heat losses at the support will be greater than the heat losses in the adjacent pipe. As such, additional heat trace will need to be provided at the supports to ensure adequate pipe temperatures.

For pipe supports handling strictly vertical loads, without the possibility of a lateral component, conventional spring hangers are recommended.

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6.6 *Control and Isolation Valves*

In general, the use of control and isolation valves in salt service should be held to the minimum.

6.6.1 Valve Types

A salt corrosion layer develops on the plugs and the seats of all valves. For gate and ball valves, the two sealing surfaces slid across one another. When the corrosion layers develop, particularly with the valves in the closed position, the corrosion layers can lead to binding and erratic valve motion. As such, gate and ball valves are not suitable for salt service. For globe and offset butterfly valves, the plugs move perpendicular to the seats, and the development of the corrosion layers are less likely to interfere with the motion of the valve.

Globe and offset butterfly valve seats and disks will have Stellite® faces. Nonetheless, hardened surfaces will corrode in salt service. Over time, the valve will fail to provide leak-tight service, and the equipment downstream of the valve must be able to operate safely with a small, but continuous, flow of salt. For example, vent valves for heat exchangers should be located at the top of the vent line to allow leakage past the valve to drain to a storage tank.

A number of items can be noted from the experience with salt valves at commercial projects:

- The commercial experience with triple offset butterfly valves in isolation service has been decidedly mixed. The valves have a low ratio of ‘thickness to diameter’. Pipe loads transferred to and through the valve body can be higher than expected due, for example, to 1) forcing the pipe into alignment with the valve body prior to welding, or 2) deteriorations in the pipe supports, anchors, or guides. This, in turn, can lead to flexing and distortion of the valve seat, in which case the valve can develop an internal leak. At one commercial project, leakage rates after about 2 years of service were such that the triple offset valves were replaced by globe valves
- In general, globe valves provide more consistent sealing characteristics because the valve body is rigid, which 1) reduces deflections of the body and the seat due to piping forces, and 2) keeps the plug and the seat in alignment.
- For a given pipe size, globe valves are likely to be more expensive, and in some cases much more expensive, than triple offset butterfly valves. However, if the plant experiences a forced outage to replace one or more triple offset butterfly valves, then the lost revenue can be on the order of \$1,000,000 per week. In general, the improvement in the plant availability associated with selecting globe valves during the design phase will justify the marginal increase in the capital cost for the globe valves.

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6.6.2 Stem Sealing

For globe valves, the preferred stem seal, by a considerable margin, is a metal bellows. The bellows, although expensive, provide a hermetic stem seal. Hermetic seals are important, as leakage past the valve stem will expose the heat trace cables on the valve body and the adjacent piping to salt. The cables operate at temperatures which are high enough ($> 650\text{ }^{\circ}\text{C}$) to decompose the salt. Several of the decomposition products are various oxides, which aggressively corrode the outer metal covering on the heat trace cables. Corrosion lifetimes can often be measured in weeks. Once the covering corrodes, the internal heating cables are exposed to moisture, and can fail in a matter of days. Failed cables lead to immovable valves and frozen salt piping, which can necessitate forced outages, lasting hours to days. A project economic analysis will often show that the marginal improvement in plant availability and revenue, due to a properly functioning heat trace system, justifies the marginal expense for bellows stem seals.

The bellows should have a minimum fatigue life of 10,000 cycles, and should be replaceable with the valve in place. The bellows region will require both heat tracing and insulation. If available, a pre-engineered bellows enclosure should be procured from the valve supplier.

The bellows is designed to withstand the maximum salt pressure expected in the valve body. In some applications, the maximum pressure may occur following a tube leak in a heat exchanger.

The primary bellows stem seal should be supplemented with a backup conventional stem packing, as described below.

For offset butterfly valves, bellows stem seals are not an option, and a conventional stem packing must be used. However, carbon and graphite materials used in standard bonnet packings are only marginally acceptable in salt service. The salt is an oxidizing material, and reacts with the carbon and graphite to form CO_2 . To date, the valve stem packing which offers the longest, but nonetheless marginal, service life consists of alternating layers of the following:

- Wire-reinforced graphite braid packing over a fiberglass core: Style 1200-PBI from Garlock Engineering, or Style 387I from John Crane, Inc.
- Fiberglass-filled Teflon® washers.

The allowable temperature range for the materials is $260\text{ }^{\circ}\text{C}$ to $315\text{ }^{\circ}\text{C}$. The lower temperature limit is to prevent salt from freezing; the upper limit is to prevent the Teflon from decomposing. Teflon, as it decomposes, produces fluorine gas, which aggressively attacks the valve stem material. Note the stem packing relies on graphite, and will require periodic replacement, regardless of the valve temperature.

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If required by the valve design, bonnet gaskets should be either metallic ring type (carbon core with a soldered silver sheath), or welded spiral wound.

Valves for service at temperatures above 400 °C will be specified with extended bonnets to ensure that the packing temperature does not exceed 200 °C in the absence of supplemental heat from the bonnet heat trace system. A heat trace zone, dedicated to the bonnet region, maintains the packing temperature in the desired range of 275 °C to 300 °C.

6.7 *Check Valves*

A conventional check valve can be provided in the discharge line of a salt pump for the following purposes:

- Should a pump trip, the check valves prevents a reverse flow through the pump, and protects against an overspeed condition in reverse rotation
- During steam generator startup, hot salt is blended with cold salt by operating the hot salt pump in parallel with the cold salt attemperation pump. At the start of the blending process, the pressure at the discharge of the hot salt pump is slightly lower than the pressure at the discharge of the attemperation pump. The speed of the hot salt pump is then increased until the discharge pressure of the hot salt pump is a few Pa higher than the discharge pressure of the attemperation pump. A check valve at the discharge of the hot salt pump 1) prevents a reverse flow though the hot salt pump when the discharge pressure of the hot salt pump is less than the discharge pressure of the attemperation pump, and 2) helps to accurately control the relative flow rates of hot salt and cold salt when the hot salt pressure is only slightly greater than the cold salt pressure.

As with all salt valves, corrosion layers will develop on the flap, the seat, and the pivot of the check valve. If the valve remains in the closed position for an extended period of time (weeks to months), then the corrosion layers on the internal components can merge, which would leave the check valve immovable.

As with globe and triple offset butterfly valves, the flap in the check valve moves perpendicular to the seat, which reduces the potential for the corrosion layers on the flap and the seat to merge. Nonetheless, the only effective means of preventing problems with corrosion is to periodically open and close the valve.

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6.8 Instruments

6.8.1 Temperature Measurements

Either thermocouples or resistance temperature detectors can be used for temperature measurements. However, the accuracy of thermocouples is generally sufficient for process control in commercial projects. The additional accuracy that may be provided by resistance temperature detectors generally does not warrant the additional expense of the detectors.

Dual element thermocouples are commonly used to reduce repair times should one of the elements fail.

Industry standard thermowells are used for fluid temperature measurements. External thermocouples, pushed against the outside of the pipe or equipment by a spring, are not acceptable. The small, but inevitable, natural convection air flows around the outside of the thermocouple housing lead to inaccurate measurements.

Temperature measurements in various locations will not be amenable to the use of thermowells. Examples include valve bodies, valve bonnets, and the outside of a pipe for heat trace control. In these cases, the thermocouple will be welded to the pipe or equipment by means of a welding tab. The thermocouple is covered with a stainless steel tent, which is then tack-welded to the equipment.

Thermocouple extension wire used with salt components must use high temperature ceramic fiber insulation.

6.8.2 Flow Measurements

Flow meter options include vortex shedding, ultrasonic, and venturi.

Vortex shedding meters are generally preferred over ultrasonic meters due to higher reliabilities, less signal noise, and lower instances of dropped signals. However, vortex meters are limited to a maximum process temperatures of 450 °C, and a maximum pipe size of DN 400. As such, the meters are restricted to use on the cold side of the system.

Ultrasonic meters do not have a restriction on maximum pipe diameter, and can be used on both the cold side and the hot side of the system. For pipe sizes of 75 mm and smaller, the refraction angles may need to be increased to achieve the desired accuracy.

In principle, the pressure transmitters described below in Section 6.8.3 can be used in combination with a differential pressure transducer to measure flow based on differential pressure. The differential pressure can be developed using conventional flow devices, including venturis, flow nozzles, and orifice

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plates. A potential design of a flow venturi in combination with diaphragm pressure transmitters is shown in Figure 6-3.

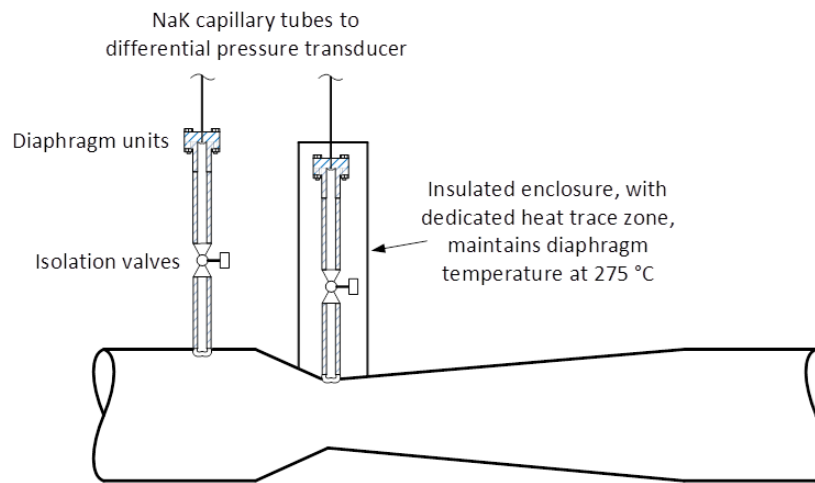


Figure 6-3 Flow Venturi with Diaphragm Pressure Transmitters

Venturis, flow nozzles, and orifice plates are available from a number of commercial vendors, and can operate over the full range of temperatures and flow rates expected in a commercial project. However, it not known if a vendor has combined a conventional flow device with diaphragm units and NaK capillary tubes to develop a differential pressure flow device for use in salt service. In principle, the reliability and the accuracy of such a device should be suitable for commercial use, and could offer an alternate to ultrasonic flow meters in high temperature ($> 450\text{ }^{\circ}\text{C}$) service.

Accuracy and Reliability

To compensate for the (at times) lower availability of ultrasonic flow meters, one of two design approaches can be followed:

- Ultrasonic meter vendors can provide 2 or 3 independent transmitter / receiver combinations in a single pipe spool piece. A 1 out of 2, or a 2 out of 3, voting logic can be used to obtain a flow signal
- Different types of flow meters can be arranged in series; i.e., a vortex unit in series with an ultrasonic unit. In this way, a 1 out of 2 voting logic can be used to compensate for dropped signals or an obvious problem with calibration.

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The latter arrangement will, of course, be met with resistance on the part of the engineering contractor, as it will increase the total length of pipe for the required diameters upstream, and the diameters downstream, of the meters. However, the control logic for the receiver likely includes a provision to trip both the heliostat field and the receiver on a loss of flow to the inlet vessel, or a loss of flow to either of the two flow circuits. If the receiver is tripped, then the receiver is typically drained. Resetting a trip of the heliostat field, and restarting the receiver, will lead to an outage lasting at least 90 minutes and as much as 3 hours. Since this type of trip happens several times a year, the marginal cost to install redundant flow meters, or different flow meters in series, can likely be recovered by the improvements in plant availability, electric output, and annual revenues.

6.8.3 Pressure Measurements

The pressure instrument with the fewest liabilities is a diaphragm unit in combination with a remote transducer. A capillary tube, filled with NaK, connects the diaphragm with the transducer. NaK is the preferred capillary fluid. It has a very low vapor pressure, which reduces the potential for changes in the diaphragm temperature to influence the pressure readings. The fluid also has a low freezing point (-12 °C), which eliminates the need to heat trace the capillary tube. Nonetheless, temperature compensation is still required for accurate readings.

The diaphragm should be located in a position, in combination with a dedicated heat trace zone and thermal insulation, which maintains a constant temperature of 275 °C during all operating modes.

An example of an installation in a vertical process line is shown in Figure 6-4. The dimensions of the external loop are selected such that a diaphragm temperature of 275 °C is maintained, even if the temperatures in the process line reach or exceed 500 °C. During normal operation, the isolation valve is closed. If it is necessary to drain the system, the valve is opened.

Experience at commercial projects indicates that using globe valves, rather than triple offset butterfly valves, in the external loops provides better isolation characteristics. This, in turn, helps to prevent damage to the diaphragm unit from overheating.

As an alternate approach, there are commercial diaphragm units designed for operation at temperatures as high as 600 °C (<https://www.badotherm.com/products/diaphragm-seal-systems/>). At two commercial projects, the Badotherm units have operated as intended.

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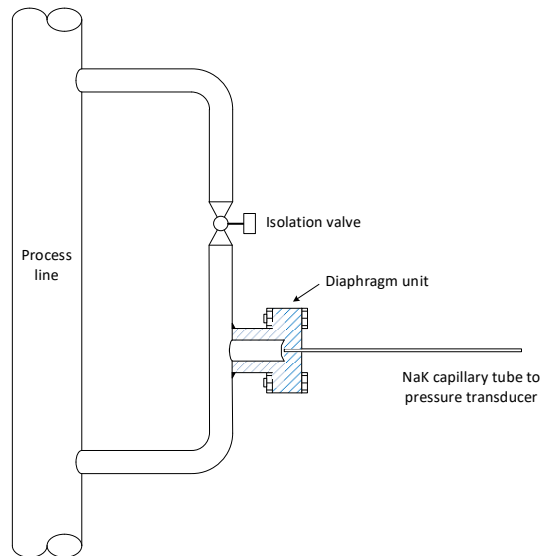


Figure 6-4 Pressure Transmitter in a Vertical Process Line

An example of an installation in a horizontal process line is shown in Figure 6-5. The dimensions of the standoff line are selected such that a diaphragm temperature of 275 °C is maintained, even if the temperatures in the process line reach or exceed 500 °C. During normal operation, the isolation valve is open. If it is necessary to perform maintenance on the diaphragm unit, the process line is drained and the valve is closed.

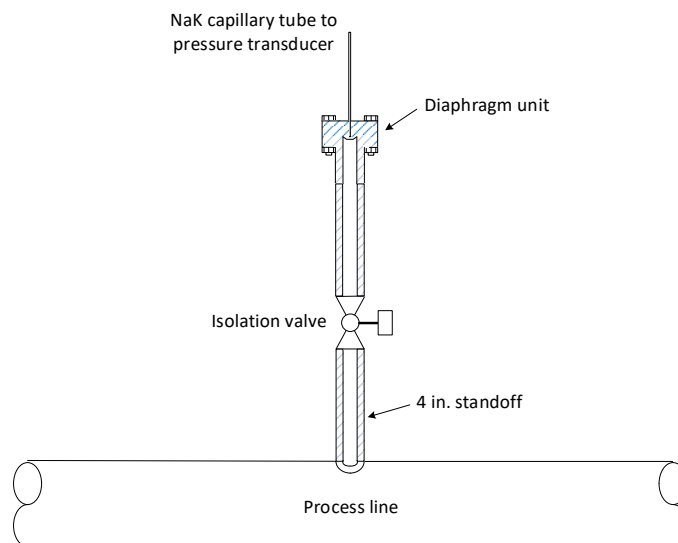


Figure 6-5 Pressure Transmitter in a Horizontal Process Line

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In both the vertical and the horizontal installations, compressed air, rather than salt, is likely to be in contact with the diaphragm. Rapid changes in the pressure in the salt are likely to be damped by the trapped air. However, for process control, the installation geometries have proven satisfactory.

6.8.4 Level Instruments

Bubbler level gauges may be used in both cold and hot tanks operating at atmospheric pressure, and in pressure vessels operating at elevated pressures.

For vessels operating at pressures greater than about 7 bar (i.e., the inlet vessel of a receiver), a supplemental air compressor, operating at pressures in the range of 30 to 50 bar, is required to supply air at the required pressure to the bubbler controller.

Radar level detector are suitable for use in both cold salt and hot salt tanks operating at atmospheric pressure.

6.9 Heat Tracing

6.9.1 System Description

The electric heat trace system thermally conditions equipment prior to initiating salt flow, and provides freeze protection on all salt equipment.

Although not specifically discussed in the Design Basis Document, electric heat trace systems may also be used for the overnight temperature control of large steam valves and the steam turbine casing.

The heat trace system is integrated, to varying degrees, with the distributed control system. The heat trace system is linked with other process control functions such that thermal conditioning can be partially or fully automated.

There are three approaches for the control of the heat trace system:

1. The heat trace vendor supplies a dedicated controller to the heat trace system. The controller, and the software, are typically proprietary to the vendor. The controller receives inputs from the control thermocouples, and sends on/off control signals to the electronic contactors that provide electric power to the heat trace circuits. The duty of the heat trace circuit is controlled by pulse width modulation of the output signal
2. Inputs from the circuit control thermocouples are received by the distributed control system, the distributed control system calculates the required pulse width of the heat trace circuit, and then sends the on/off control signals to the circuit contactors

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3. A hybrid approach, in which 1) the standard controller is provided by the vendor to control the heat trace circuits, and 2) a communication link is provided between the vendor controller and the distributed control system such that distributed control system can monitor all aspects of the heat trace system.

The first approach is the most common. However, the controller typically has a dedicated station for the operator, which can make tailoring the operation of the heat trace system to the plant operating mode a tedious process. Further, the temperature signals from the heat trace control thermocouples are not available to the data historian, which can make system diagnosis a more complicated process.

The second approach 1) makes available to the distributed control system all of the information from the heat trace system, and 2) allows the operation of the heat trace circuits to be automatically adjusted to the plant operating mode. However, the number of input signals to, and output signals from, the distributed control system is a large number, and this will increase the cost of the distributed control system.

This third approach can combine the benefits, and minimize the liabilities, of the first and the second approaches. However, it will require the cooperation of the heat trace vendor regarding project access to the vendor's proprietary hardware and software.

The approach selected in a commercial project will depend on the preferences of the Owner, plus the willingness of the heat trace vendor to allow the project access to vendor equipment.

6.9.2 Scope of Supply

The electric heat trace equipment will be purchased as a system that encompasses the zone definitions, heat transfer calculations, cable arrangements, sensor locations, cable fabrication, installation, and acceptance testing. Zone definitions, and locations of the temperature sensors, will be developed in cooperation with the project engineering contractor. The scope of supply will encompass all materials and hardware, including the heat trace cables, splice kits, power supply junction boxes, temperature sensors, sensor junction boxes, and all installation hardware.

Fabrication and assembly of each heat trace cable will be performed by the supplier in the field. This is to ensure the cables are cut to match the as-built configuration of the equipment. Normally, fabrication of the cables at the vendor's shop is preferred, as the quality of the hot-to-cold junctions can be more closely controlled. However, the project schedule does not allow the preparation of factory cables after the salt equipment has been installed. The vendor will be required to supply cables, manufactured in the field, to have the same quality and reliability as cables manufactured at the vendor's shop.

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The heat trace system design and supply should include the solid state circuit contactors. However, the electric power distribution, installation of the solid state contractors, and connections of the temperature sensors to the instrument junction boxes should be provided by the site electrical contractor.

6.9.3 Electric Heat Tracing System Design Basis

Zone Definitions

The boundaries of the heat trace zones will be defined by the equipment geometries, the system operating modes, and liquid locations within the equipment. Also, zone temperature set points will vary by state and transition. The definition of the zones will be determined largely by the project engineering contractor. However, the contractor will collaborate with the heat trace vendor to ensure that zone requirements (cable length, total duty, number and location of thermocouples) are consistent with commercial practice. As an example, transitions between operating modes can result in certain pipe segments changing from flowing salt to stagnant salt. The zone boundaries, and the location of the temperature sensors, must be selected such that the full lengths of the stagnant zones, which change with the operating mode, are always protected. In addition, the temperature sensors must monitor the temperatures only in the stagnant zones, not in the flowing zones. The design process can be aided by three-dimensional piping diagrams, which illustrate the status of each pipe segment during each operating mode.

In general, zone lengths for piping should be as long a possible. Plant availability is improved by 1) reducing in the number of cables, contactors, controllers, and communication lines; 2) reducing the number of alarms monitored by the operators; and 3) reducing the number of potential operating modes which are not used correctly. The division of a piping circuit into multiple zones tends to decrease the plant availability, as the loss of one zone is equivalent to the loss of the complete circuit. The potential availability penalty associated with lengthy outages to remove and replaced failed cables in long heat trace zones is offset by increasing the number of installed spare cables.

Any significant change in the mass per unit length in a zone, such a valve in a pipe, requires a separate zone.

Any significant change in heat loss per unit length in a zone, such as a valve bonnet, requires a separate zone.

Any pipe section or instrument location, in which a zone of stagnant salt is intentionally formed for temperature control, requires a separate zone. Examples are illustrated in Figure 6-4 Pressure Transmitter in a Vertical Process Line, and in Figure 6-5 Pressure Transmitter in a Horizontal Process Line.

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Thermal losses at pipe and equipment supports will be much higher than in the adjacent pipe or equipment. The geometry of the supports will need to be analyzed by the engineering contractor to determine if separate zones are required at the supports.

If different operating modes subject a pipe or a vessel to different static fluid heights, then the boundary between the zones must end above the liquid level. For example, recirculation from a salt pump at the solar project produced a static height of about 9 m in a riser. If the heat trace zone in the lower section of the riser had ended below an elevation of 9 m, the stagnant salt above the top of the zone would have frozen in a matter of a few hours. Further, if the zone spans a region which is not exposed to salt at the top, but is flooded below, a heat transfer analysis must be conducted to confirm that the temperature in the upper portion of the zone does not exceed safe levels.

Zone Thermal Capacities

The thermal capacity in each zone is determined by the following factors:

- An economic analysis, comparing the capital cost of the heat trace capacity with the system startup time. One example is the riser and the downcomer. At the end of an operating day, both are normally drained. Prior to startup the following day, the temperatures of the piping must be increased to the salt fill temperature. If both lines have high heat trace capacities, then the preheat time will be short, the preheat energy will be low, but the capital cost will be high. (The reverse also applies.) As such, there is an optimum heat trace capacity that minimizes the sum of (the equivalent capital cost of the preheat energy) and (the capital cost of the heat tracing).
- An Owner's assessment of the degree to which the heat trace capacity must accommodate an expected degradation in the cable heating capacity or an increase in the heat losses through the equipment insulation
- An Owner's assessment of the reliability of the heat trace cables, and the requirement for spare installed cables.

A representative set of design parameters might include the following:

- Empty salt pipes, empty valve bodies, and valve bonnets can be preheated from ambient temperature to 275 °C in 12 hours
- Empty salt heat exchangers can be preheated from ambient temperature to 275 °C in 36 hours. Preheating energy will be provided by a combination of electric heat trace cables on the shells and the channels, and an electric recirculation water heater.

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The calculations will be performed using the design minimum ambient temperature and a concurrent wind speed of 14 m/sec.

Allowances of 5 percent will be added to the zone capacities for simple geometries (pipe sections) and 30 percent for complex geometries (valve bodies, valve bonnets, pipe supports, guides, and anchors).

Control and Set Point Temperatures

The minimum system set point temperatures, for both cold and hot salt equipment, will be 275 °C, with the following exceptions:

- The cold tank/hot tank diversion valve will have a set point temperature of 475 °C to reduce the transient thermal stresses in the valve body
- In a proposed pumped heat storage project, the design cold salt temperature is 270 °C. This, in turn, implies minimum set point temperatures in the range of 260 to 265 °C.

The controller temperature dead band is selected to limit contactor cycling; i.e., -10 °C to +10 °C.

Component Redundancy

The required number of heat trace cables to satisfy the preheat requirements will be calculated as described above. The number of installed cables will be equal to the number of required cables, plus the following allowances for spare cables:

- 100 percent redundancy (i.e., 1 spare cable for each active cable) for the cables 1) on the hot salt piping from the receiver, and 2) the hot salt piping to the steam generator. These segments of piping operate under daily thermal cycles, and the failure rates of the heat trace cables are expected to be 'moderate to high'
- 50 percent redundancy (i.e., 1 spare cable for every 2 active cables) for the cables on 1) the cold salt piping to the receiver, 2) the cold salt piping from the steam generator, and 3) the vent and drain lines for the steam generator heat exchangers. These segments of piping operate under essentially steady state temperatures, and the failure rates of the heat trace cables are expected to be 'low'.

The spare cables will accommodate failures of the original cables without the need to remove the insulation. The spare cables will be labeled as such, but not connected to the electric power supply.

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To the extent possible, large valves will have spare heat trace cables installed. For small valves, and for the bonnets on both large and small valves, the installation of spare cables is not practical. As such, spare cables, fabricated to the dimensions of the original cables, will be assembled and kept in the warehouse for future maintenance.

For each temperature sensor installed on a pipe, valve body, valve bonnet, or instrument, an installed spare shall be provided. Dual element thermocouples satisfy the redundancy requirement.

Component Requirements

The recommended cable type is mineral insulated, with dual conductor heating elements, a welded sheath, and magnesium oxide insulation. The cables can be supplied with either an Inconel or a 300-series stainless steel sheath. The latter is less expensive than the former, and offers a similar resistance to corrosion should the cables be exposed to salt following a leak.

Two cable diameters are recommended: $\frac{5}{16}$ -inch (nominal 8 mm) for 600 V service for long piping zones; and $\frac{3}{16}$ -inch (nominal 5 mm) for 120 V service on valves and line devices. The conductor resistance will vary, depending upon the length of the cable. Cable power density should be limited to 160 W/m to reduce the potential for corrosion of the sheath in the event of a salt leak.

The cables are secured to the pipe using stainless steel tie wires or straps. A typical installation, showing the end of a zone, is illustrated in Figure 6-6. A stainless steel foil layer is installed over the cables, followed by a flexible blanket insulation approximately 25 mm thick. The foil prevents the flexible blanket from potentially lodging underneath the cables, which would cause the cables to overheat. A rigid block insulation is installed over the blanket. The blanket conforms to the uneven surfaces produced by the cables, the ties, and the corrugated tubes at the end of the zone. This, in turn, reduces the potential for convection heat flows under the rigid insulation.

For zone temperature measurement, either thermocouples or resistance temperature detectors are acceptable. Whichever type is selected, it will be standard across the entire facility.

Valve Zones

The heat trace for a valve will be independent of the heat trace for the adjacent pipe, for three reasons:

- The unit mass, in kg/m, of a valve body is much higher than the unit mass of the adjacent pipe. As such, if a time is specified to preheat the piping system to 275 °C, the heat input per meter for a valve body must be much higher than the heat input per meter for the pipe

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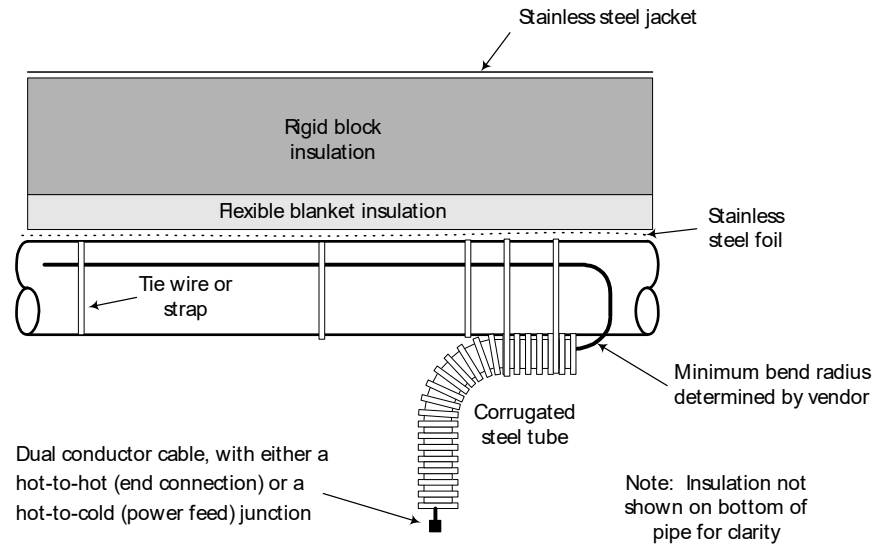


Figure 6-6 Heat Trace Arrangement on Pipe at End of Zone

- The type of insulation on a valve body is different than the type of insulation on the adjacent pipe. As such, under steady state conditions, the heat loss per meter of valve body will be different than the heat loss per meter of the pipe
- The unit heat losses, in W/m^2 , from the valve bonnet are much higher than the unit heat losses from the valve body. Also, the temperature of the bonnet must be maintained within a narrow range of $275\text{ }^{\circ}\text{C}$ to $300\text{ }^{\circ}\text{C}$ if a stem packing is used. The lower value ensures the salt will not freeze in the bonnet, and the upper value prevents the Teflon in the auxiliary stem packing from decomposing.

A typical cable installation on a valve is illustrated in Figure 6-7. The power rating for cable on the valve body is selected based on the required system preheat time. If the valve body is large enough, redundant cables can be installed. The power rating for the cable on the valve bonnet is selected to maintain metal temperatures in the range noted above, under the worst combination of wind speed and ambient temperature. In general, limited space is available on valve bonnets, and only one cable can be provided. Since the heat trace circuit for the valve body operates independently of the heat trace circuit for the valve bonnet, separate thermocouples are required for the two zones.

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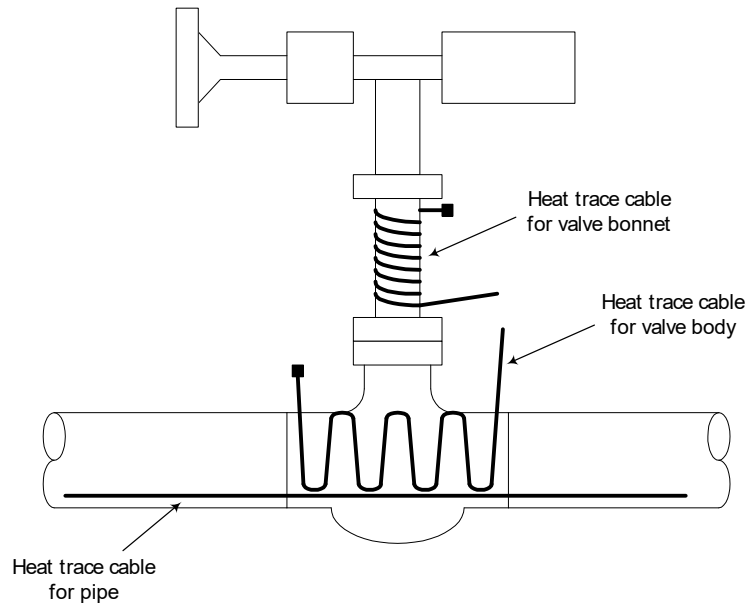


Figure 6-7 Typical Heat Trace Cable Arrangement on Valves

Check valves do not have extended bonnets. If the insulation thickness on the check valve is the same as the insulation thickness on the adjacent pipe, then the unit heat loss for the valve should be essentially the same as the unit heat loss for the pipe. Consequently, treating the pipe and a check valve as a contiguous zone is an acceptable design practice.

Temperature Sensor Installation

Sensor elements are to be welded or banded to the pipe or equipment, and then covered with a stainless steel tent to reduce convection heat losses from the sensor.

The number of sensors in a zone will depend on the geometry of the zone, as follows:

- For zones of limited dimensions, such as instrument stubs, one dual-element thermocouple is adequate
- For long pipes, operating in cyclic service, as many as 6 to 8 dual-element thermocouples may be needed to understand the temperature distribution throughout the zone. The heat trace circuits would be controlled based on a low select feature

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- For pipes operating with salt flowing in some modes, but switching to static salt in other modes, a minimum of 2, and as many as 6, dual-element thermocouples may be needed to maintain set point temperatures during all of the modes.

6.9.4 Equipment Heat Tracing

Heat tracing requirements unique to major equipment items are described below.

Pressure Safety Valves

All pressure relief valves in salt service will have electric heat tracing. However, the springs are subject to relaxation, even those fabricated from high temperature materials such as Inconel. As such, the heat trace cable locations and the valve insulation must be designed to 1) maintain a valve body temperature of at least 275 °C, and 2) maintain the spring temperature as low as practical. In general, separate zones will be provided for the valve body, the inlet piping, and the outlet piping.

The discharge line from the valve must have heat tracing over the entire length of the pipe.

A weep hole at the low point in the valve discharge line should be provided to indicate the valve is leaking.

Vortex Shedding Flow Meters

Vortex shedding flow meters reside in fittings, which are generally the same diameter as the pipe. As such, the meter and the adjacent pipe can be treated as a contiguous zone. Nonetheless, the meter has a small boss which houses the vibration sensor, and the boss extends through the pipe insulation. The boss is not insulated due to temperature limits on the piezoelectric sensor. Therefore, the boss acts like a fin and cools the top of the fitting.

At one solar demonstration project, a loop in the shape of an “S” with a length of 30 cm, was added to each cable on the 150 mm flow meters to compensate for the convection losses from the boss. With a unit rating of 130 Watts per meter, and 2 active cables, the loops increased the heat input to the meter by 80 Watts over that which would have been provided by the cables on the adjoining pipe.

Tank Level Gauges

Air supply lines to bubbler level gauges will have electric heat tracing. The heat tracing is required to prevent salt from freezing in the air lines should the flow of air stops, and 1) liquid salt flows backwards into the line, or 2) salt vapors enter and condense in the line.

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Capillary Lines for Diaphragm Pressure Transmitters

The capillary fluid is the eutectic NaK mixture, which has a freezing point is about -12 °C. As such, no electric heat trace is required for the capillary line.

Salt Pumps

The salt pumps are supported on a platform above the storage tanks, and draw suction from near the bottom of the tanks. An elevation view of the general arrangement is shown in Figure 6-8. The gap between the top of the tank and the bottom of the support structure is spanned with a metal bellows, as shown by the arrow in Figure 6-9. Electric heat trace will be installed on the bellows.

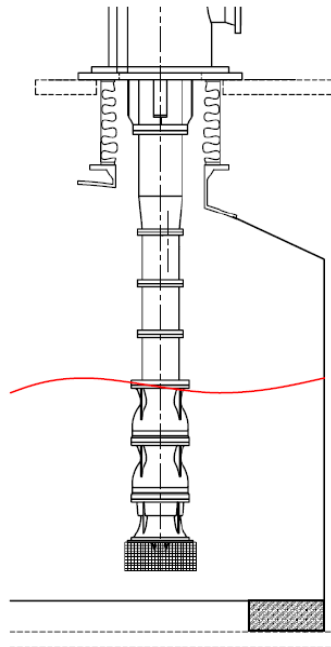


Figure 6-8 Elevation View of Salt Pump

Depending on the pump geometry, it may also be possible to install heat tracing in the region above the bellows and adjacent to the discharge flange. If not, this portion of the pump must be thoroughly insulated to prevent salt from freezing on the shaft, and, depending on the location of the bearings, lock the rotor to the housing.

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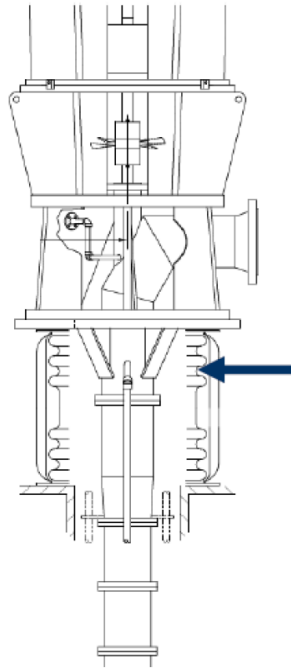


Figure 6-9 Detail of Salt Pump Discharge Head and Bellows

Essentially all salt pumps use a stuffing box near the top of the pump shaft. The stuffing box reduces the vertical migration of salt up the shaft due to surface tension effects. The stuffing box is provided with either service or instrument air to 1) develop a slight positive pressure near the top of the pump, and 2) cool the packing in the box. The air lines to the box must have electric heat tracing to prevent salt vapors from condensing in the lines. (When the pump is idle, the air flow to the stuffing box is stopped to limit cooling at the top of the shaft.)

6.10 Salt Pumps

6.10.1 Configuration

The pumps are a vertical turbine design, with the bearings lubricated by the salt. The pumps are located on structures above the salt tanks, and draw suction from a point near the bottom of the tank. Overall shaft lengths of 14 to 16 m are expected.

Representative fabrication materials for the cold salt and the hot salt pumps are shown in Table 6-1.

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Table 6-1 Salt Pump Fabrication Materials

	Cold Salt	Hot Salt
Column	Carbon steel	Stainless steel
Shaft	Alloy steel	Stainless or nickel alloy steel
Bowls	Carbon steel	Stainless steel
Impellers	Stainless steel	Stainless steel
Bearings	Cast iron	Cobalt chromium alloy (Stellite)

An insulated bellows, illustrated in Figure 6-8, 1) bridges the space between the top of the pump and the bottom of the support structure, and 2) allows for differential movement between the tank and the support structure.

The pump discharge may be either a ring type joint or a Reflange R-Con connector. The ring gasket or hub ring shall be Type 321H or 347H stainless steel.

The pumps use variable speed motor drives. The pump will be capable of operating from minimum speed through rated speed with no restrictions on speed.

6.10.2 Installed Redundancy

Essentially all commercial projects provide some level of installed pump redundancy. A typical arrangement for the pumps in the steam generation system include three 50-percent capacity hot salt pumps, and two 100-percent capacity cold salt attemperation pumps. The pump arrangement is based on two 50-percent heat exchanger trains supplied from 1) the 3 hot salt pumps, which are on a common header, and 2) the 2 attemperation pumps, which are on a common, but separate, header.

A typical arrangement for the receiver system uses four 33-percent capacity cold salt pumps supplying a common riser.

Steam Generator Pump Redundancy

As discussed in Section 6.6.1 of Volume 3 - Narrative, there is a range of benefits associated with independent steam generator trains. Since the trains are hydraulically independent, three options are available for supplying salt to the heat exchangers:

- Option 1) One 100-percent capacity installed hot salt pump, with a spare pump in the warehouse, and one 100-percent capacity installed attemperation pump, with a spare pump in the

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warehouse. For both spare pumps, the spare consists of only the pump; a spare motor and a spare variable frequency drive are not provided

- Option 2) Two 100-percent capacity installed hot salt pumps, and two 100-percent capacity installed attemperation pumps
- Option 3) Two 50-percent capacity installed hot salt pumps, with a spare pump in the warehouse, and two 50-percent capacity installed attemperation pumps, with a spare pump in the warehouse. For both spare pumps, the spare consists of only the pump; a spare motor and a spare variable frequency drive are not provided.

Option 1

Option 1 should provide the best combination of availability and capital cost. On the order of 150 salt pumps with extended shafts are in commercial service, and the reliability has been excellent. Further, the time required to remove and replace a pump can, in principle, be as short as 24 to 48 hours, as discussed in Section 7.2.5. An outage period of 1 to 2 days is comparable to the outage periods discussed below for Option 2. As such, Option 1 does not necessarily impose an inherent availability penalty relative to Option 2.

Option 2

In theory, Option 2 could offer an improvement in availability over Option 1. However, the problems with salt pumps are generally not associated with the pump; the problems are typically associated with the ancillary equipment for the pump. These problems include:

- Sticking or erratic movement of the discharge control and the discharge isolation valves
- Leakage of salt from the stem seals of the discharge control or the discharge isolation valves. The salt contaminates the insulation on the valves and the adjacent piping, which can lead to piping temperatures in stagnant lines falling below the freezing point of the salt. Once the salt in a line freezes, the heat trace is no longer capable of thawing the line. To restore the system, the insulation will need to be removed, the stem seals on the valves replaced, the insulation replaced, and the heat trace returned to service. The time required to thaw a line is a function of the pipe diameter, and can range from hours for a small line to days for a large line.
- Sticking or erratic movements of the vent valves which remove air from the pump column
- Incorrect or erratic readings from the flow and pressure instruments at the pump discharge, leading to unnecessary trip signals or incorrect judgement calls on the part of the operator

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- Incorrect timing of the supply of instrument air to the shaft packing, leading to salt freezing in the gland and binding of the pump
- Problems with the heat tracing on each of the above items, leading to low equipment temperatures (salt freezing) or high equipment temperatures (corrosion of carbon steel components).

Although counterintuitive, two 100-percent capacity pumps can result in a decrease in system availability relative to one 100-percent capacity pump because redundant pumps require twice the ancillary equipment. Stated another way, there are a finite number of maintenance hours available to maintain the equipment in working order. If the number of ancillary items is reduced by a factor of 2, then it is more likely that the equipment will receive the required maintenance to reach the projected levels of availability.

Lastly, Option 1 requires four pumps, two motors, and two variable frequency drives, while Option 2 requires four pumps, four motors, and four variable frequency drives. As such, Option 1 should be slightly less expensive than Option 2.

Option 3

As noted in the introductory paragraph, a typical steam generator pump arrangement includes three 50-percent hot salt pumps, and two 100-percent attemperation pumps, with the 5 pumps supplying two 50-percent steam generator trains. As such, for each train, there is one 100-percent hot salt pump and one 200-percent capacity attemperation pump. In contrast, Option 1 refines this approach by providing one 100-percent capacity hot salt pump and one 100-percent capacity attemperation pump, which should be the preferred arrangement.

In addition, Option 1 requires four pumps, two motors, and two variable frequency drives, while Option 3 requires six pumps, four motors, and four variable frequency drives. As such, Option 1 should be somewhat less expensive than Option 3.

Receiver Pump Redundancy

As with the steam generation system, the majority of the problems with salt pumps are associated with the ancillary equipment. To this end, the availability of the receiver pump system can be improved by using three, rather than four, 33-percent capacity pumps.

It can be noted that some commercial projects will operate four 33-percent capacity pumps during those hours in which the solar radiation is particularly high. Nonetheless, to some extent, this operating approach is adopted because the pumps cannot meet their required combination of flow and head; i.e., a 33-percent capacity pump cannot provide one-third of the required flow. The solution to this problem is

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not to provide a margin on the flow and head characteristics of the three pumps by starting a fourth pump. The preferred approach is to specify the necessary margins as part of the procurement process, and obligate the vendor to correct any performance deficiencies identified in the water flow tests. The margins should reflect uncertainties in items such as the effect of corrosion on the friction factors in the riser and the absorber.

6.11 Receiver System

The state-of-the-art in salt receiver designs is discussed below in Section 7.1.

6.11.1 Design Codes

The receiver is not a fired heater; as such, the selection of ASME Section I as the design code is not a mandatory jurisdictional requirement. (Nonetheless, Boeing selected Section I as the design code for the salt receiver at the 10 MWe Solar Two demonstration project in California. Of the Code options, this approach was judged to be the most stringent, and Boeing wished to avoid potential problems with the local Authorized Inspector regarding the eventual selection of the design code.)

ASME Section VIII Division 2 is often selected as the design code, as it includes methods for estimating low cycle fatigue life in cyclic service. Division 2 also includes provisions for design by analysis, including methods for estimating effects such as combined creep-fatigue damage using visco-plastic analyses. However, Section VIII is applicable only for section thicknesses of 3.2 mm (1/8 in.) and above. In contrast, all commercial salt receivers use tubes with wall thicknesses in the range of 1.5 to 1.65 mm, and Section VIII is not strictly applicable. To avoid this limitation, SolarReserve adopted ASME B31.1, Power Piping, as the design code for the receiver at the Crescent Dunes project.

6.11.2 Material Selection

The materials of choice for the receiver in a commercial project include the following:

- Alloy 230 for the absorber tubes
- Alloy 230 or Type 347H stainless steel for the panel headers
- Type 347H stainless steel for the outlet vessel and the high temperature piping
- Carbon steel for the inlet vessel and the low temperature piping.

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Intergranular Stress Corrosion Cracking of Alloy 230

Alloy 230 is a high nickel alloy, and is generally viewed as immune to intergranular stress corrosion cracking. Nonetheless, leaks in the steam generator introduce water into the storage tanks. Steam released from the salt can travel up the receiver vent line, and condense in the receiver panels at night. At one commercial project, cracks developed in the Alloy 230 receiver tubes, and an examination of the tubes showed evidence of water condensation and intergranular stress corrosion cracking.

It is optimistic to expect that leaks will never develop in the steam generator. One method to prevent steam condensation in the receiver is to purge the receiver with dry, or with heated dry, air at night. The source of the dry air is the instrument air system. This approach was adopted at the 10 MWe Solar Two demonstration project. At this project, the receiver tubes were fabricated from Type 316H stainless steel. Intergranular stress corrosion cracking was judged to be largely inevitable, and supplying dry air to the receiver at night was done to extend, as much as possible, the life of the receiver.

Post Weld Heat Treatment of Type 347H Piping

Type 347H stainless steel is nominally Type 304H stainless steel with niobium added as an alloying element. The niobium bonds with the carbon to form niobium carbide. As such, free carbon, normally present in Type 304H, is not available to react with chromium to form chromium carbide. Since chromium carbide is a necessary precursor to intergranular stress corrosion cracking, Type 347H offers an improved resistance to intergranular stress corrosion cracking than either Type 304H or Type 316H.

Nonetheless, the improved corrosion resistance is degraded during welding. Metal temperatures during welding are high enough to dissolve the niobium carbide. As the weld cools, there are regions of the weld zone that favor the reformation of niobium carbide, but there are also regions in which the time-temperature progression favor the formation of chromium carbide. Typically, there are narrow zones, on either side of the heat affected zone, that are rich in chromium carbide, and these zones are susceptible to intergranular stress corrosion cracking.

One solution to this weld feature is to perform a post weld heat treatment. The heat treatment process consists of a series of steps in which 1) the temperature is raised to a point which dissolves the chromium carbide, 2) the temperature is raised to a new value which favors the reaction of the now-available carbon with niobium to form niobium carbide, and 3) the temperature is quickly reduced through the point which favors the reaction of the remaining carbon with chromium to form chromium carbide.

It can be noted that the theoretical benefits of a post weld heat treatment are not assured, for the following reasons:

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- The work is performed in the field, and there are inherent variations in the ambient weather conditions and the skills of the personnel
- If the required time-temperature profiles are not followed accurately, then the full transition from chromium carbide to niobium carbide may not be achieved. As a result, the weld region may still be susceptible to intergranular stress corrosion cracking
- Depending on the weld filler material and the time-temperature profile, there is the potential to form a range of intermetallic compounds, such as Fe-Cr, Fe-Mo, and $(\text{FeNi})_x(\text{CrMo})_y$. These compounds increase the strength, but reduce the ductility, of the weld region.

Different engineering contractors have different opinions as to the value of post weld heat treatment of Type 347H piping. At one commercial project, the vendor for the receiver performed a post weld heat treatment of all of the stainless steel piping in the receiver, while the engineering contractor for the balance of the plant (i.e., everything outside of the scope of the receiver) did not perform a post weld heat treatment of any of the stainless steel piping.

In general, the solar industry has yet to reach a consensus on the value of, and the need for, the post weld heat treatment of Type 347H piping. Early in the design phase, the project should consult with a materials specialist and a welding engineer to develop an approach to post weld heat treatment that will be applied on a consistent basis throughout the plant.

6.11.3 Tube Clip Design

Each tube is supported at the top by a panel, and is periodically guided over the length of tube by a series of independent clips. The clips maintain the tubes parallel to each other, and in the plane of the absorber, while allowing unrestricted vertical movement for thermal expansion and contraction.

The clips are designed to allow one tube to be removed from the panel for replacement.

The clip-to-tube weld procedure must limit the penetration of the tube wall to the minimum amount necessary for complete fusion. Full penetration welds are not permitted.

The tube clips must slide freely in the support frame guides, while allowing for unrestricted thermal growth of the tubes. Unrestricted is defined as the change in tube length due to a change in temperature from ambient to the melting point of the material.

The tube clip must provide a low cycle fatigue life of 30,000 cycles at a rate of temperature change of 360 °C/min.

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The tube clip acts as a heat sink. As such, the clip cannot be located at the edge of the oven box structure. Otherwise, the tube clip would not be properly heated by either the radiant heaters in the oven box or by the preheat flux from the heliostat field. Tube clips must be located outside of the oven box envelope over the full range of tube thermal expansions.

6.11.4 Panel Vent Design

Air must be vented from the absorber panels during the daily fill process. Two options are available for the venting arrangement:

- A vent line is connected to the top of the upper crossover line between two adjacent panels, and an orifice is installed in the vent line. The orifices discharge to a header which is common to the panels in that flow circuit. An isolation valve is installed at the end of each vent header, and the discharge from the two isolation valves connect to the side of the outlet vessel.
- A vent line is connected to the top of the upper crossover line between two adjacent panels, and an isolation valve is installed in the vent line. The valves discharge to a location on the side of the outlet vessel that is above the normal liquid level

The first approach reduces the number of isolation valves, and this generally offers a meaningful improvement in system availability. However, the static pressure in the crossover line between Panels 1 and 2 is higher than the static pressure in the crossover line between Panels 3 and 4. As such, there is a small flow of salt which passes from the 1-2 crossover line to the 3-4 crossover line. Since the salt temperature at the outlet of Panel 1 is lower than the salt temperature at the outlet of Panel 3, the bypass flow in the vent line reduces the salt temperature at the inlet of Panel 4. The attenuation effect occurs along the length of the circuit, and reduces the outlet temperature from the last panel by about 5 °C. As such, the set point for the receiver outlet temperature must be set to 570 °C to achieve a temperature in the downcomer of 565 °C. This has a small, but measurable, effect on the receiver efficiency and the low cycle fatigue life of the panels near the end of the flow circuit.

The second approach avoids the performance penalties, but increases the number of valves. The reliability of valves in salt service, including the reliability of the heat trace and the insulation on the valves, is such that commercial projects generally accept the loss in performance associated with the vent orifices.

6.11.5 Downcomer Design

There are two principal approaches to the design of the downcomer:

- Flooded, with the level in the receiver outlet vessel controlled by throttle valves at the base of the downcomer

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- Partially flooded, with the static head dissipated by orifice plates located along the length of the downcomer. Neither an outlet vessel, nor throttle valves, are required.

The most common approach is the flooded downcomer, as the hydraulic performance is fully predictable.

With a flooded downcomer, the outlet vessel must be vented to the atmosphere to provide a predictable level control signal to the downcomer throttle valves. However, the vented vessel can overflow if there are control upsets in the downcomer. To protect the receiver from damage should the outlet vessel overflow, a vent line connects the top of the outlet vessel with the top of the hot salt tank. The vent line must normally operate empty due to Code requirements for a vented vessel. As such, when salt enters the vent line, the velocity increases at a rate of 9.8 m/sec^2 . When the salt reaches the first elbow in the first expansion loop, some of the velocity head is converted to a momentum load on the elbow. Momentum loads are difficult to estimate, as the flow is a mixture of salt and air, and the relative proportions of the two fluids change at different locations in the vertical and the horizontal sections of the vent line. Nonetheless, the momentum loads are expected to be high, with values in the range of 75,000 to 150,000 N. A robust set of pipe guides and anchors will be required to transfer these loads from the piping to the tower. The principal design risk is underestimating the loads, and then having to retrofit larger guides and anchors in a location where access is difficult.

To some degree, the complexity in analyzing the flows in a partially flooded downcomer with orifice plates is nominally the same order of magnitude as analyzing the flows in a partially flooded vent line. These considerations may influence the choice between a flooded downcomer and a partially flooded downcomer.

6.11.6 Assembly Location

There are two locations options for the assembly of the receiver:

- At the top of the tower. Components are lifted by crane, and then assembled at an elevation of about 200 m.
- At grade. The receiver is assembled at grade, moved to a position inside the tower through a large opening on the side of the tower, and then lifted by jacks supported by the tower. The opening on the side of the tower is then closed and reinforced.

The former approach is perhaps the more common, but it can incur certain liabilities, as follows:

- A penalty in labor productivity due to a need for all of the craft workers to commute on an elevator with a finite capacity

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- Limited areas for the staging of materials and equipment
- Wind speeds are higher than at grade, which can reduce the number of hours suitable for welding.

The latter approaches reduces the above liabilities, but requires a tower design which can safely withstand wind and seismic loads with a large opening in the side.

The engineering contractor will need to conduct an economic analysis of the two options to determine the preferred construction approach.

6.11.7 Instruments

Instruments in salt service are typically problematic. There has been a wealth of experience gained over the past two decades at commercial projects; some of it good, and much of it bad. As such, the selection of the type, and the method for the installation, of the instruments should be prescriptive. Details of the recommended instruments are presented above in Section 6.8.

6.11.8 Non Destructive Examinations

All tube and header welds require non-destructive examination to verify weld quality and integrity. Each panel will be hydrotested to 1.5 times the operating pressures in accordance with ASME code requirements. In addition, each panel will be pressurized with helium to TBD bar, and the panel assembly helium leak tested, in accordance with ASME Section V. Molten nitrate salt is an excellent wetting agent and will penetrate porous surfaces and minute cracks that will not be apparent with a hydrotest.

6.12 Thermal Storage System

The state-of-the-art in thermal storage designs is discussed below in Section 7.3.

6.12.1 Introduction

Specifications for the storage tanks in commercial projects are a mix of functional and prescriptive requirements. The functional aspects deal with items such as thermal capacity, allowable heat losses, and maximum soil temperatures. The prescriptive aspects include the following:

- Type of tank; i.e., flat bottom with a self-supporting dome roof
- Design codes and standards

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- Materials
- Methods for emptying a tank.

However, at least 6 failures of the hot salt tanks in commercial projects have occurred in the past decade. Further, the number of hot tanks in commercial service (outside of China) is very limited: Gemasolar, Crescent Dunes, Noor III, and two at Cerro Dominador. This translates to a probability of failure of essentially 100 percent. As such, the format of the current set of functional and prescriptive specifications is not adequate for the needs of the industry. Outlined below are some recommendations regarding additional prescriptive requirements to the specification.

6.12.2 Design Codes

A specific design code for flat bottom tanks, operating at high temperatures and under cyclic thermal conditions, is under development but is not yet approved for public distribution. As such, the closest applicable code is American Petroleum Institute Standard 650, Welded Steel Tanks for Oil Storage. However, the Standard applies only to process temperatures of 260 °C and below, and does not consider low cycle fatigue conditions. To date, commercial projects have used the following codes, in combination with API 650, for the design of the tank:

- ASME Section II for allowable material stresses
- ASME Section VIII Division 2 for calculating the low cycle fatigue life associated with stresses during commissioning and operation.

Nonetheless, if the tanks, designed to API 650 / Section II / Section VIII Division 2, are failing, is this combination of design codes appropriate for the service conditions? Specifically, at the temperatures of interest in the hot tank (> 550 °C), stainless steel is subject to the effects of creep. Due to the cyclic nature of plant operation, in combination with multi-hour periods at high liquid inventories (high stresses), the hot tank should be designed to withstand the combined effects of fatigue and creep.

The topic is discussed in Section 5.14.1, Changes in the Tank Design Codes, of Volume 3 - Narrative. The recommended approach is to use Section III Division 1 Subsection NH, or alternately Section III Division 5, to develop values for the allowable stresses consistent with limits on accumulated damage due to creep and fatigue. In addition, residual welding stresses should be included in the assessment of creep damage, as discussed in Section 5.15.9, Code Interpretation of Residual Stresses, of Volume 3 - Narrative.

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6.12.3 Inputs to the Fatigue Analysis

The floors in the thermal storage tanks are constructed from an array of rectangular plates. The nominal plate dimensions are 2.5 m by 12.5 m. Representative plate thicknesses are in the range of 7 to 10 mm, and two weld passes are typically required.

The allowable tolerances on the dimensions of the plates are presented in ASTM A480/A480M-22a, Standard Specification for General Requirements for Flat-Rolled Stainless and Heat-Resisting Steel Plate, Sheet, and Strip. Since all of the plates are not exactly the same size and shape, the dimensions of the gaps between adjacent plates will vary. To provide consistent inter-plate welds, backing strips are welded to the backs of alternating plates prior to arranging the plates on the foundation.

The storage tanks are subjected to daily thermal cycles, which results in cyclic loads on the floor generated by friction and radial temperature gradients. For the tanks in parabolic trough projects, it's not clear if a low cycle fatigue analysis is required. However, if such an analysis is justified, then the fatigue properties of the weld are subject to a weld fatigue strength reduction factor. The topic is discussed in Section 5.13.2, Fatigue Strength Reduction Factors, of Volume 3 - Narrative.

To improve the calculated fatigue life of the floor, the reduction factors should be as low as practical. The smallest reduction factors (1.0) are achieved by 1) machining the weld surface, 2) confirming the weld quality through the use of ultrasonic tests (radiographic examinations are not practical on the floor), and 3) examining the weld surface through a combination of visual, magnetic particle, or dye penetrant tests.

The largest reduction factors (4.0) occur on the back sides of welds that cannot be observed or are otherwise non-definable. This characteristic applies to the bottom of the root pass in contact with the top of the backing strip.

The Owner must coordinate with the tank fabricator to reach an agreement on the welding procedures and the weld examinations to ensure a consistent treatment among the cost of the tank floor, the calculated fatigue life, and the expected fatigue life in commercial service.

6.12.4 Material Selection

Hot Tank

For design temperatures above 538 °C, the H grades of stainless steel, with a minimum carbon content of 0.04 percent, are required by the ASME Code. The carbon helps to pin the metal grains, which provides the required resistance to creep.

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Candidate materials include Type 304H, Type 316H, and Type 347H stainless steel. At the Solar Two demonstration project, the hot tank was fabricated from Type 304H. However, to avoid potential problems with intergranular stress corrosion cracking, the solar industry has adopted Type 347H, a stabilized stainless steel, in the current generation of projects.

Nonetheless, the choice of Type 347H as the stabilized stainless steel is not without a known set of risks. The principal problem is the use of niobium as the stabilizing element, which can lead to a phenomenon known as stress relaxation cracking. The topic is discussed in some detail in Section 5.12 of Volume 3 - Narrative.

In future commercial projects, the engineering contractor must perform a risk analysis between potential tank materials, neither of which offers a perfect solution:

- Type 304H and Type 316H essentially avoid the potential for stress relaxation cracking due to the absence of niobium. However, the materials are susceptible to intergranular stress corrosion cracking should the tank be exposed to liquid water in combination with chlorides. This can happen under the following conditions:
 - a. A pumped heat storage project is planned for a coastal location. The tank, after fabrication but prior to commissioning, can be exposed to salt spray from the ocean.
 - b. After the tank has been in commercial service, the tank is drained for maintenance and inspection. If the temperature of the metal drops below the dew point, then moisture can condense on either the inside or the outside of the tank. (A potential solution is to introduce dry, heated, air into the tank such that the metal temperatures never fall below the dew point. To what extent this can be guaranteed by the maintenance company is perhaps an open topic.)
- Type 347H reduces the potential for intergranular stress corrosion cracking, but introduces the potential for stress relaxation cracking. The cracking mechanism is a function of the metal thickness, the number of welding passes, the stress distribution, and the time / temperature history of the tank. If the development rate of the cracking can be predicted with the necessary accuracy, then it may be possible to fabricate a tank in Year 0 of the project, and then fabricate a replacement tank to be placed into service, for example, in Year 10 of the project. However, if the development rate of the cracking cannot be predicted with the required certainty, then the project may be obligated to build two tanks in Year 0 in preparation for an expected failure at some undefined point in the future.

Cold Tank

The cold tank is fabricated from a conventional carbon steel, such as A 516 Grade 60 or 70.

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6.12.5 Determination of the Floor Status Prior to the Start of Commercial Operation

Fabricating the floor, preheating the tank, and filling the tank for the first time has the potential to introduce a set of plastic deformations in the floor. The engineering contractor must define the three-dimensional shape of the floor at the time of initial filling. The shape of the floor can be determined by lidar scans from the roof of the tank, or by inelastic finite element calculations of the buckling patterns. The initial shape of the floor is a principal input to a subsequent stress analysis of the floor. The stress analysis will consider loads due to friction, radial and circumferential stresses due to radial temperature gradients, and transient stresses due to receiver startup, shutdown, and trips. The stress analysis, in turn, will define operating permissives, to be programmed in the distributed control system, regarding allowable combinations of inventory depth, downcomer temperature, and downcomer flow rate.

Deformations Produced in the Floor During Welding

The floor consists of a pattern of rectangular plates. Welding the plates produces compressive stresses at each edge. The stresses produce plastic deformations, which convert the flat plates into shallow domes.

The combination of high spots near the center of each plate, and low regions at the edges, results in a plate which has already buckled. The floor is, in essence, a large thin plate, and if the plate has already buckled, then it's resistance to further buckling is very low.

Deformations Produced During the Hydraulic Test

The floor, after welding, has developed something of a waffle pattern. As such, tracing the distance along either the top or the bottom of all of the plates in the floor results in a dimension which is greater than the diameter of the tank.

The radial stiffness of the bottom course of the tank is several orders of magnitude greater than the radial stiffness of the floor. The floor plates are relatively thin (7 to 10 mm), and when the tank is filled with water during the hydraulic test, the slightly domed plates are pushed down into contact with the foundation. (The hydrostatic forces required to deform the plates are discussed in Section 5.8.1, Plate Stresses, in Volume 3 - Narrative.) This changes the dimension of the floor, from one which was slightly larger than the diameter of the bottom course to a new dimension which is equal to the diameter of the bottom course.

The change in dimension will appear as some combination of elastic and additional plastic deformation in the floor. When the water is removed from the tank, the floor will develop a new shape, based, in part, on the residual stresses associated with any new plastic deformations.

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An assessment of the plastic deformations made during the hydraulic test can be made by measuring the shape of the floor before and after the test.

Deformations Produced in the Floor During Preheating

During preheating, combustion gases from an air heater enter the tank through a nozzle suspended from the roof. The gases move the tank in a somewhat uncontrolled manner, and leave the tank through a manway in the roof opposite the inlet nozzle.

The temperature distribution in the tank is a function of the gas flow pattern, the local heat transfer coefficients, heat transfer to the foundation, the thermal inertia of the foundation, heat transfer to the tank external insulation, and the thermal inertia of the wall, the roof, and the external insulation. Almost by definition, the metal temperatures will not increase at the same rate throughout the tank. This, in turn, will establish thermal stresses within the tank.

The magnitude of the stresses must be estimated to determine the effect on the fatigue life of the tank, and to determine if additional plastic deformations have occurred during preheating. For example, as discussed in Section 7.3.3, radial temperature gradients in the floor of 35 °C and higher have the potential to plastically deform the floor near the center of the tank.

Deformations Produced in the Floor During Initial Filling with Salt

During preheating, additional deformations, both elastic and plastic, can be produced in the floor due to temperature gradients within the tank. After preheating is complete, the floor is likely to retain something of a waffle shape. Some form of buckles may also be present, depending on the stresses associated with welding, hydraulic testing, and preheating.

As with the hydraulic test, the floor will be pushed down against the foundation when the hydrostatic pressure reaches a nominal value of 7 kPa. This occurs with a salt depth of about 0.4 m. However, the structural stiffness of the buckles, if present, is much higher than the bending stiffness of the plates. As such, the buckles, once formed, become a permanent feature of the floor.

Deforming the plates during filling will result in a new combination of stresses and deformations. The new combination becomes the starting point for the creep and fatigue characteristics of the tank associated with the daily cyclic operation of the plant.

6.12.6 Computational Fluid Dynamics Calculations

The salt flow enters the tank through a distribution header located just above the floor. Due to the large dimensions of commercial tanks (~ 40 m diameter), the temperature distribution in the inventory is necessarily non-uniform during transient conditions. Non-uniform temperature distributions produce

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thermal stresses, and the magnitude of the stresses need to be defined to calculate the low cycle fatigue life of the tank.

Cold Tank Thermal Cycles during Commercial Operation

The cold tank is subjected to a number of temperature cycles over the course of a year. The most common is the daily startup of the receiver. At the beginning of the startup, the temperature of the salt leaving the receiver is the cold salt temperature (290 °C), and the flow in the downcomer returns to the cold salt tank. Over the next 15 to 20 minutes, the temperature at the salt leaving the receiver increases from 290 °C to the design value of 565 °C. At some point in the progression, the flow in the downcomer switches from the cold tank to the hot tank. A typical crossover temperature is 520 °C, but the crossover value can change, depending on the initial levels and the inventory temperatures in the cold tank and the hot tank. As such, the cold tank experiences a slow cooling rate during the evening shutdown period, followed by a modest heating cycle during each receiver startup.

Similarly, during a weather outage period, the cold tank experiences a slow cooling cycle, followed by a modest heating cycle when the receiver is placed back in service. However, the inventory temperatures, and the rates of temperature change, are somewhat different than the daily receiver startup cycles due to the different hold times.

A summary of the annual temperature cycles expected for the cold tank are listed in Table 6-2.

Estimated rates of temperature change in the cold tank inventory, during the first 40 minutes of the heating phase in each cycle, are shown in Figure 6-10. The temperature profiles assume the inventory and the flow entering the tank are perfectly mixed. However, the mixing cannot occur instantly; thus, the temperature change rates of the tank floor, wall, and roof will be somewhat less than shown. The corresponding inventory temperatures during the first 40 minutes of the heating phase are shown in Figure 6-11.

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Table 6-2 Temperature Cycles for the Cold Salt Tank

<u>Type of cycle</u>	<u>Number in 30 years</u>	<u>Duration, hours</u>	<u>Temperature, °C</u>		<u>Rate of temperature change¹, °C / hr</u>
			<u>Beginning</u>	<u>Ending</u>	
Annual maintenance					
- Cooling phase	30	168	290	283	- 0.04 ²
- Heating phase	30	0.17	283	295	+ 105 ³
Weather outage					
- Cooling phase	300	48	290	288	- 0.04 ²
- Heating phase	300	0.17	288	299	+ 102 ⁴
Equipment outage					
- Cooling phase	1,500	24	290	289	- 0.04 ²
- Heating phase	1,500	0.17	289	300	+ 102 ⁶
Daily operation					
- Cooling phase	10,000	12	290	290	- 0.02 ⁵
Receiver start					
- Heating phase	10,000	0.17	290	296	+ 57 ⁷

Notes:

1. The '-' sign indicates the tank is cooling; the '+' sign indicates the tank is heating.
2. Cooling rate with active inventory level of 50 percent.
3. Peak heating rate with an initial active inventory level of 50 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 283 °C, and ending inlet salt temperature of 530 °C.
4. Peak heating rate with an initial active inventory level of 50 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 288 °C, and ending inlet salt temperature of 518 °C.
5. Cooling rate with an active inventory level of 100 percent.
6. Peak heating rate with an initial active inventory level of 50 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 289 °C, and ending inlet salt temperature of 513 °C.
7. Peak heating rate with an initial active inventory level of 100 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 290 °C, and ending inlet salt temperature of 540 °C.

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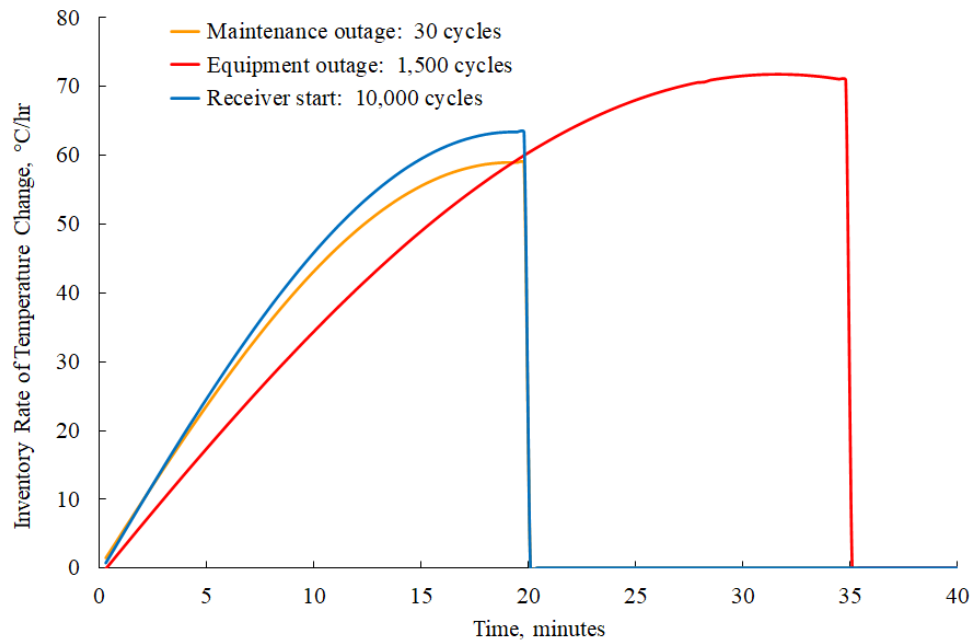


Figure 6-10 Rates of Change in the Cold Tank Inventory Temperature

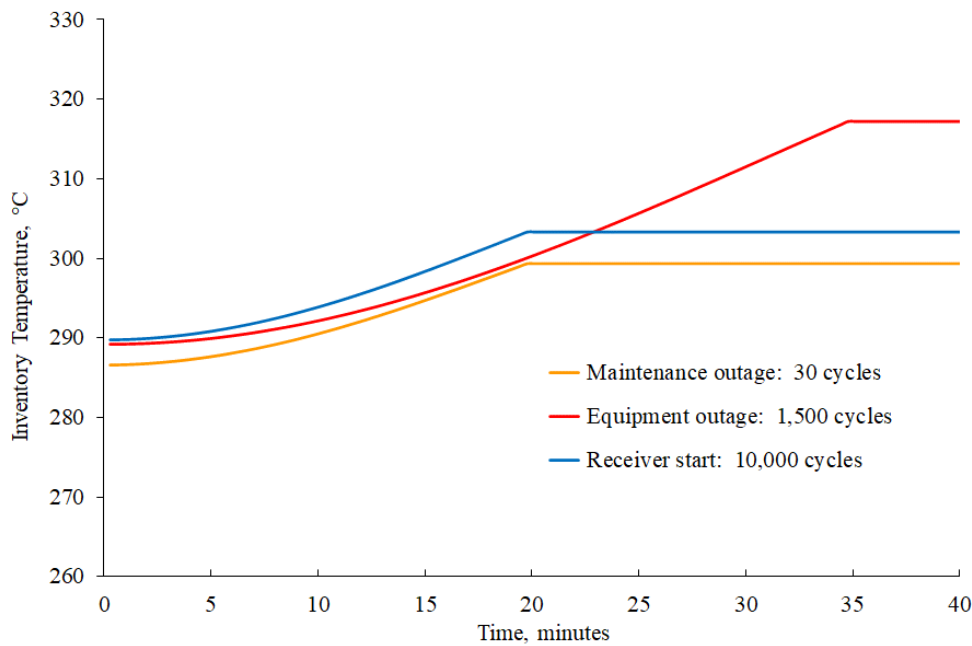


Figure 6-11 Cold Tank Inventory Temperatures versus Time

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Hot Tank Thermal Cycles during Commercial Operation

The hot tank is also subjected to a number of temperature cycles over the course of a year. The most common is the daily startup of the receiver. At a predefined crossover temperature ($\sim 520^{\circ}\text{C}$), the flow in the downcomer switches from the cold tank to the hot tank. The crossover temperature is generally lower than the inventory temperature in the hot tank ($\sim 550^{\circ}\text{C}$ during the overnight hold period) to capture as much thermal energy from the receiver as practical. During the period in which the temperature in the downcomer is in the range of 520 to 550°C , the hot tank undergoes a modest cooling cycle. Once the temperature in the downcomer reaches 550°C , the hot tank undergoes a slow heating cycle.

Similarly, during a weather outage period, the hot tank experiences a slow cooling cycle, followed by a modest heating cycle when the receiver is placed back in service. However, the inventory temperatures, and the rates of temperature change, are somewhat different than the daily receiver startup cycles due to the different hold times.

A summary of the annual temperature cycles expected for the hot tank are listed in Table 6-3.

Table 6-3 Temperature Cycles for the Hot Salt Tank

<u>Type of cycle</u>	<u>Number in 30 years</u>	<u>Duration, hours</u>	<u>Temperature, $^{\circ}\text{C}$</u>		<u>Rate of temperature change¹, $^{\circ}\text{C/hr}$</u>
			<u>Beginning</u>	<u>Ending</u>	
Annual maintenance					
- Cooling phase	30	168	565	540	- 0.40^2
- Heating phase	30	6	540	565	-9 to + 21^3
Weather outage					
- Cooling phase	300	48	565	558	- 0.40^2
- Heating phase	300	6	558	565	- 40 to + 8^4
Equipment outage					
- Cooling phase	1,500	24	565	563	- 0.10^5
- Heating phase	1,500	6	563	565	- 31 to + 2^6
Evening operation					
- Cooling phase	10,000	12	565	560	- 0.40^2
Receiver start					
- Heating phase	10,000	6	560	565	- 49 to + 12^7

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Notes:

1. The '-' sign indicates the tank is cooling; the '+' sign indicates the tank is heating.
2. Cooling rate with active inventory level of 0 percent (i.e., minimum stagnant heel).
3. Peak cooling rate and peak heating rate, with an initial active inventory level of 0 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 540 °C, and ending inlet salt temperature of 565 °C. The tank cools for the first 2 minutes, then starts to heat. The inventory temperature continues to rise until a value of 565 °C is reached.
4. Peak cooling rate and peak heating rate, with an initial active inventory level of 0 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 540 °C, and ending inlet salt temperature of 565 °C. The tank cools for the first 6 minutes, then starts to heat. The inventory temperature continues to rise until a value of 565 °C is reached.
5. Cooling rate with an active inventory level of 35 percent.
6. Peak cooling rate and peak heating rate, with an initial active inventory level of 35 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 513 °C, and ending inlet salt temperature of 565 °C. The tank cools for the first 7 minutes, then starts to heat. The inventory temperature continues to rise until a value of 565 °C is reached.
7. Peak cooling rate and peak heating rate, with an initial active inventory level of 0 percent, inlet salt flow rate of 1,177 kg/sec, beginning inlet salt temperature of 540 °C, and ending inlet salt temperature of 565 °C. The tank cools for the first 5 minutes, then starts to heat. The inventory temperature continues to rise until a value of 565 °C is reached.

Estimated rates of temperature change in the cold tank inventory, during the first 30 minutes of the cooling / heating phase in each cycle, are shown in Figure 6-12. The temperature profiles assume the inventory and the flow entering the tank are perfectly mixed. The corresponding inventory temperatures during the first 30 minutes of the cooling / heating phase are shown in Figure 6-13.

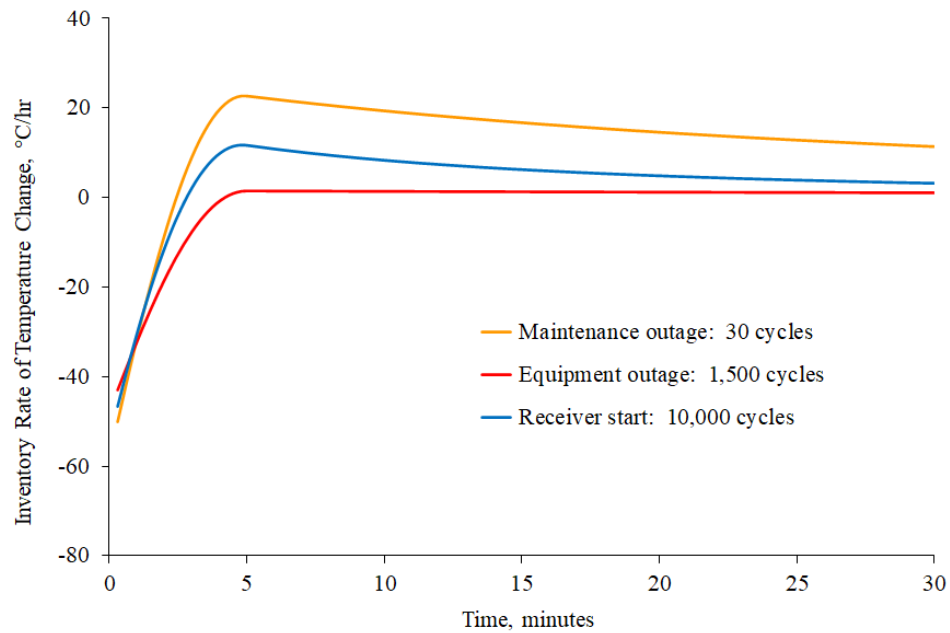


Figure 6-12 Rates of Change in the Hot Tank Inventory Temperature

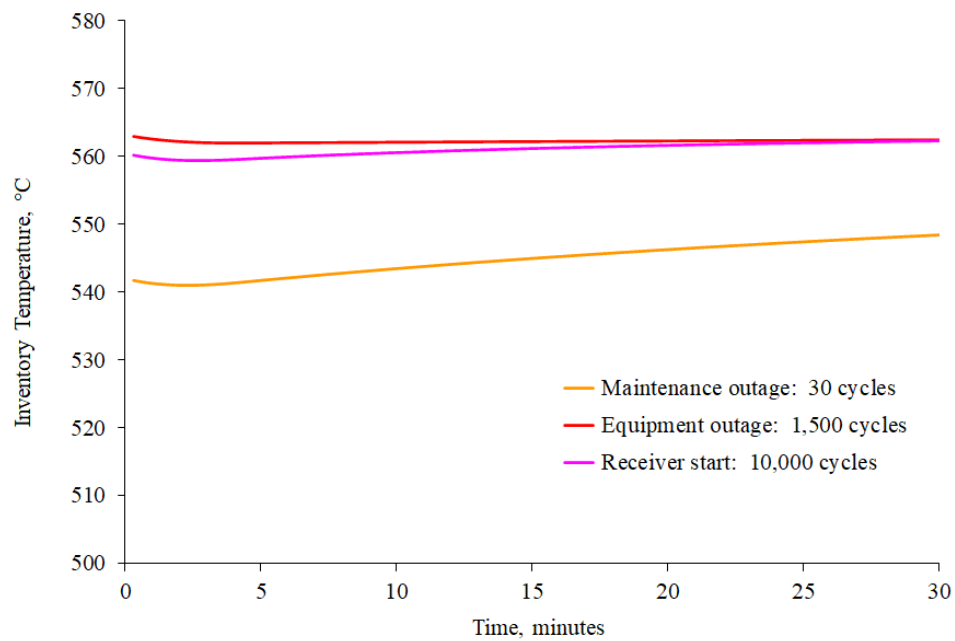


Figure 6-13 Hot Tank Inventory Temperatures versus Time

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Cold and Hot Tank Status During Outage Periods

Friction forces between the floor and the foundation develop, in part, due to changes in the inventory temperature during hold and outage periods (Section 5.10.4 in Volume 3 - Narrative). If the inventory is at, or near, the minimum level, then floor stresses remain well within allowable values for all changes in the inventory temperature. However, if the inventory is at, or near, the design level, then changes in the inventory temperature must be limited to values in the range of 2 to 3 °C to maintain floor stresses within acceptable values.

During extended outages, the tank temperatures can change, as follows:

- The temperature in the cold tank can decay by as much as 30 °C, which is followed by an increase of 30 °C during a return to commercial service
- The temperature in the hot tank can decay by as much as 290 °C, which is followed by an increase of 290 °C during a return to commercial service.

Changes in the inventory temperature of 30 to 290 °C are large enough to establish the full potential friction forces on the floor. As such, to reduce the friction forces during, and at the end of, the extended outage period, the inventory levels in both the cold and the hot tank should be set to, and maintained at, 50 percent.

Atypical Transient Conditions

The rates of change in the inventory temperatures listed in Table 6-2 and Table 6-3 represent the plant under normal operating conditions. However, a range of atypical conditions can arise, which will subject the tanks to conditions outside of those listed in the Tables. Two examples include the following:

Receiver Trip A possible, and not unreasonable, scenario is one in which the hot tank is close to the minimum inventory, the receiver trips, and then the following single failure occurs: the downcomer diversion valve to the cold tank fails to open. The receiver outlet temperature decays at the projected rate of 360 °C/min, and the entire inventory of the receiver inlet vessel is directed to the hot tank. The estimated rate of change in the tank inventory temperature is shown in Figure 6-14.

The peak rate of temperature change in the bulk inventory is -360 °C/hr, which is approximately 6 times the typical rate allowed by the tank supplier. Depending on the effectiveness of the distribution ring, local rates of temperature change at the floor could exceed -360 °C/hr, which has the potential for permanently damaging the floor.

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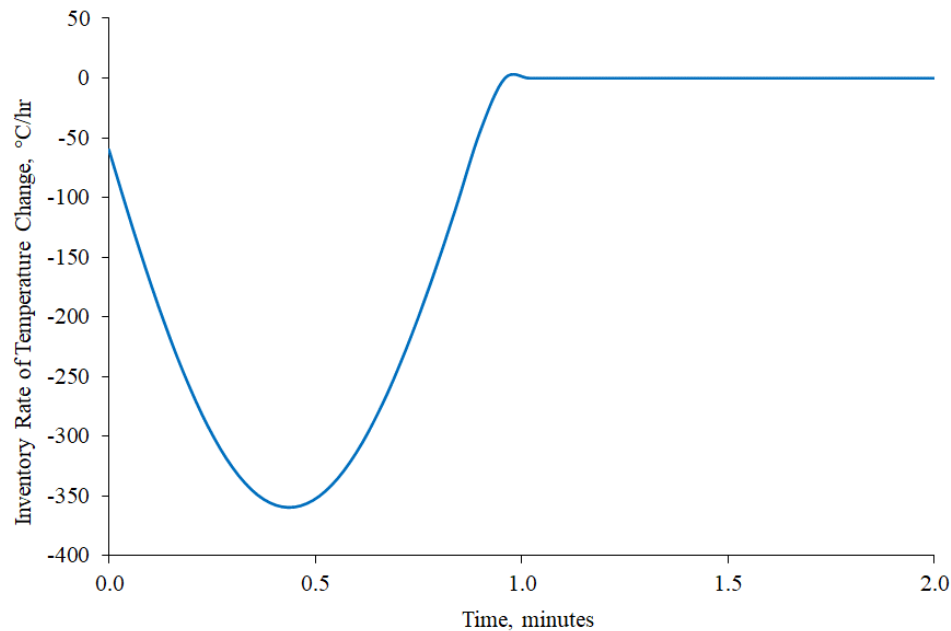


Figure 6-14 Rate of Change in the Hot Tank Inventory Temperature Following a Receiver Trip

Selected Crossover Temperature As discussed in Section 6.12.6, the downcomer crossover temperature can be reset by the operator at any time. In the interests of capturing as much energy from the receiver as possible for electric power production, the operator may select a crossover temperature that is considerably lower than the inventory temperature in the hot tank. The effect of selecting a crossover temperature that is 90 °C lower than the inventory temperature is illustrated in Figure 6-15.

The result is a peak rate of temperature change of -230 °C/hr, which is about 4 times the typical allowable rate.

Each of these atypical transient conditions has the potential to significantly reduce the low cycle fatigue life of, or causing immediate damage to, the tank. Since both of these conditions are possible, and perhaps probable, the project must identify approaches to avoid damage to the tank. Potential approaches include the following:

- Develop an inlet distribution system that isolates, as much as possible, the floor from the rapid changes in the inlet temperature
- Train the operating staff in the basic principles of transient thermal stresses and low cycle fatigue
- Mandate limits on acceptable crossover temperatures as a function of the tank level.

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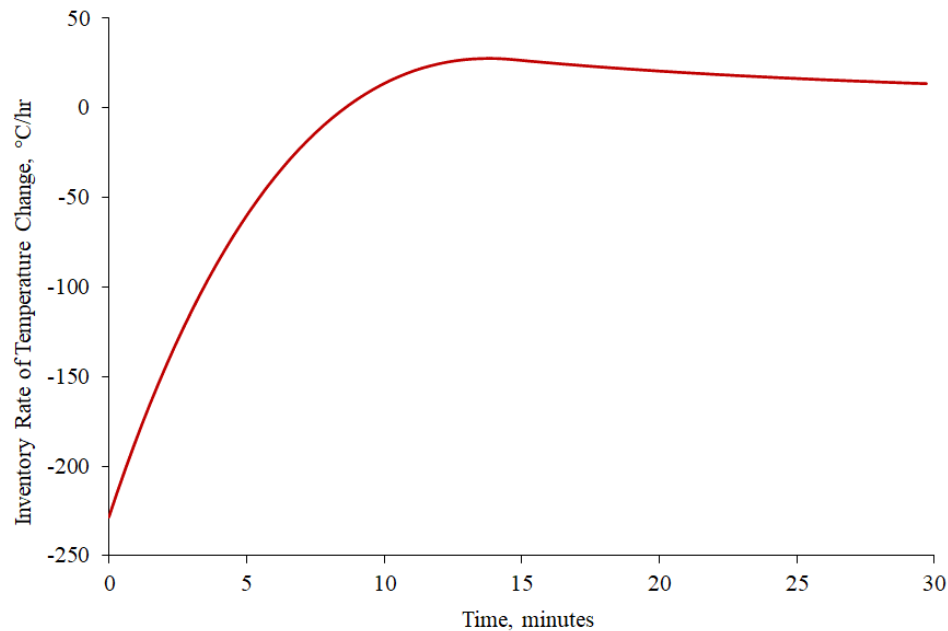


Figure 6-15 Effect of a Crossover Temperature 90 °C Lower than the Inventory Temperature

6.12.7 Combined CFD / FEA Calculations

The engineering contractor, having defined the initial stress distributions and the deformations in the floor after filling with salt, must determine the stress distributions in the tank expected during daily operating sequences. The sequences include both normal operation and emergency conditions, such as a receiver trip.

The calculations must include the effects of the following:

- Friction between the floor and the foundation, including the non-uniform friction effects due to the weld backing plates attached to the bottom of the floor
- Deformations of the floor, including the potential for continued yielding and ratcheting.

The engineering contractor must demonstrate to the Owner that the selected design approach of a flat bottom tank has a fatigue life at least equal to, and preferably longer than, the duration of the project.

6.12.8 Tank Venting

Nitrate salt, as delivered to the project, is approximately 99 percent nitrate (NO_3^-). Once the project has been in commercial operation for perhaps 1 to 2 years, the nitrate ion reaches an equilibrium condition

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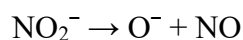
with the nitrite ion (NO_2^-) and oxygen in the ullage gas in the storage tanks based on the following reaction:



The equilibrium concentration of the nitrite ion is on the order of 3 to 5 percent, depending on the operating characteristics of the receiver and the storage volume.

Cover Gas

As such, air is the preferred cover gas. Specifically, using an alternate cover gas, such as nitrogen, would cause the above the equation to always shift to the right. This, over time, would increase the nitrite ion concentration at the expense of the nitrate ion concentration, and promote the following non-equilibrium reaction:



The non-equilibrium reaction is undesirable, for two reasons:

- The oxide ions (O^-) remain in solution, and are aggressive corrosive agents
- The NO is released from the liquid in the form of a gas, and produces various forms of NO_x in contact with the ambient.

Inter-tank Vent Line

During a discharge cycle, salt moves from the hot tank to the cold tank. If there is an inter-tank vent line, then ullage gas, at a temperature of 295 °C, moves from the cold tank to the hot tank. The cold gas, in contact with the inner surfaces of the hot tank, is heated. To prevent the ullage pressure in the hot tank from exceeding the design pressure, a portion of the ullage gas moving from the cold tank to the hot tank must be vented to the atmosphere.

The process is reversed during a charge cycle, in which ambient air must be added to the cold tank to prevent the ullage pressure in the cold tank from falling below the design vacuum rating.

An inter-tank vent reduces the mass of air which must be vented from the hot tank, and added to the cold tank, over the course of a charge / discharge cycle. This, in turn, reduces one source of thermal losses from the storage system. However, the use of an inter-tank vent line incurs two liabilities:

- The line must be continuously heat traced to prevent the condensation of salt vapors in the line. A calculation shows that the annual energy required to maintain the vent line at a minimum

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temperature of 275 °C is greater than the marginal energy lost due to simply venting the tanks to the ambient

- If the heat tracing and the insulation on the vent line are not in perfect working order, then it is possible to condense enough salt vapors from the storage tanks to completely plug the line. This is not a theoretical problem, as this condition developed at the Solar Two demonstration project. The defect was a section of the vent line, approximately 200 mm long, between the end of one heat trace zone and the start of the next heat trace zone. Over the course of about 2 years, the DN250 line became fully blocked. With the vent line blocked, and with the pressure/vacuum relief valve on the hot tank partially blocked due to a small defect in the valve insulation, filling the hot tank during the morning startup of the receiver forced the tank floor into a convex shape, which lifted the tank from the foundation. The relief valve (eventually) lifted, which dropped the tank back down onto the foundation. This situation had the potential to rupture the tank.

The preferred approach to venting the tanks is to provide, on each tank, an open goose neck which is vented directly to the atmosphere. The full length of the goose neck is heat traced and insulated to prevent the condensation of salt vapors. The goose neck is installed in parallel with the tank pressure / vacuum relief valves required by the API 650 Standard.

6.12.9 Foundations

Floor Insulation

The vertical loads on the foundation are a function of the radial position in the foundation. Near the perimeter of the tank, the principal vertical load is the dead weight of the wall and the roof. These loads are distributed into the foundation by an annular plate, approximately 1 m wide, located just below the wall. Inside the inner radius of the annular plate, the principal vertical load on the foundation is the static head of the salt inventory.

The principal insulating material under the floor of the tank is either an expanded clay (Utelite) or an expanded glass (FoamGlas). Design information for both expanded clay and Utelite show allowable compressive stresses that are higher than the anticipated foundation loads, both at the perimeter of the tank and at the center of the tank. As a result, a number of commercial projects have adopted expanded clay as the sole foundation material due to its low cost, low thermal conductivity, and acceptable strength. However, in all of these projects, the expanded clay has not performed as expected. In particular, the particles of clay are prone to shifting and movement as the tank undergoes cyclic expansion and contraction cycles. The movement of the clay is due to two effects:

- At the perimeter of the tank is an annular plate, which is thicker than the floor. The plate 1) transmits the concentrated weight of the roof, the wall, and the tank insulation down into the foundation, and 2) ensure that the displacements in bending are at acceptable levels. The annular

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plate also carries some of the static hydraulic loads from the salt inventory. However, the compressive loads just inside the perimeter of the plate must be matched by tensile loads just outside the perimeter of the plate. Expanded clay has only a limited capacity to transmit tensile loads. Further, periodic fill and drain cycles in the tank establish periodic tensile loads in the expanded clay. These two conditions cause the clay to move out, up, and away from the perimeter of the foundation.

- Daily thermal cycles of the tank cause the annular plate to push the clay out from the perimeter as the tank expands, but the clay does not return to its original position as the tank contracts.

As a result, the outer edge of the annular plate loses its vertical support. The plate has only a limited capability for accommodating bending loads, and the concentrated vertical load from the wall and the roof plastically deforms the plate in a downward arc. This is a dangerous condition, as plastic deformation of the wall-to-floor welds seriously compromises the low cycle fatigue life of the tank.

To prevent this situation, the foundation materials directly beneath the wall must be sufficiently rigid to prevent any angular deflections of the annular plate. Two design options have been adopted for this purpose:

- A refractory material, either in the form of bricks or a large casting, is placed in the space between the top of the concrete base mat and the bottom of the tank.
- Metal slip plates are located on top of the expanded clay to prevent direct contact between the floor and the expanded clay.

The first option was successfully used at the 10 MWe Solar Two demonstration project, it was adopted and is in use at one commercial project, and it has been adopted as the design basis for a plant under construction. To date, no problems have been reported from the commercial project in operation.

The second option has been adopted at two commercial central receiver projects, and at least one commercial parabolic trough project. However, the results are somewhat mixed for the hot tanks in the central receiver projects. Specifically, the annular plate and the slip plate must necessarily be stainless steel. However, stainless steels in sliding contact are prone to galling, and this can lead to effective coefficients of friction which are greater than 1. This in turn, can produce two undesirable characteristics:

- When the tank expands, high compressive forces can be transferred from the annular ring to the floor, and the floor already has a poor resistance to buckling

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- When the tank either expands or contracts, high radial forces are developed on the annular plate, and some form of an anchor must be provided for the annular plate. However, the anchor is necessarily located beneath the floor, and there is essentially no access for inspection, maintenance, or repair.

In general, the use of a refractory material to support the weight of the wall and the roof represents, perhaps, the smallest sum of liabilities.

It can be noted that refractory materials carry their own set of design liabilities, as follows:

- The materials, either as brick or cast, are not inexpensive
- Some of the refractories can react with nitrate salt, leading to a loss of strength. As such, the refractories must be isolated from potential leaks, as discussed below
- There is a significant degree of uncertainty in the coefficient of friction between refractories and stainless steel at temperatures of 500 °C and above. Preliminary experimental data shows coefficients to be greater than 0.3, and perhaps as high as 1.0. This may, in effect, obligate the use of a solid lubricant between the annular ring and the refractory.

Solid Lubricant

The most common solid lubricant is sand, as it is inexpensive and thermally stable. Nonetheless, the coefficient of friction between stainless steel and sand is poorly defined. The industry, for better or worse, has assumed nominal coefficients to be in the range of 0.3 to 0.4, but there are little data to corroborate this assumption.

Ideally, the project Owner would commission an experiment, early in the project development phase, to determine a defensible value. This value would then be used by the engineering contractor in the analysis of floor friction loads.

Drip Pan

Nitrate salt is an excellent wetting agent. Should a leak develop in either the wall or in the floor, salt will permeate the materials in the foundation. As discussed in Section 5.3 of Volume 3 - Narrative, contamination of the foundation with salt will lead to an increase in thermal losses, the potential to produce NO_x emissions, and the potential to decrease the strength of refractory materials. To limit these consequences, a continuous drip pan must be installed under the layer of solid lubricant. The pan will direct any leakage to safe areas beyond the edge of the foundation.

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6.12.10 Foundation Cooling

Soil temperatures beneath the concrete base mat must be maintained at values less than about 75 °C. These temperatures limit the desiccation of the soil, and to limit the oxidation of organic materials in the soil. The goal is limit the shrinkage and the settlement of the soil to values which are consistent with the normal allowances for the settlement of the tank.

To control the soil temperatures, heat is removed from the base mat by means of active cooling. Two cooling options are available: forced air cooling, and water circulation. The solar industry has universally adopted forced air cooling, for the following reasons:

- The heat flux into the foundation is perhaps 60 to 70 W/m². This is a modest value, and the flux can be readily transferred to a flow of air with an acceptable temperature rise along the diameter of the foundation.
- Unlike a water circuit, leakage in the air duct does not materially affect the cooling capacity.

6.12.11 Methods for Emptying a Tank

The salt pumps are located on structures above the roofs of the tanks. As such, the top of the pump is located in a fixed position. However, the location of the bottom of the pump is a function of the thermal expansion of the pump column. In the hot tank at one commercial project, the distance between the bottom of the pump and the top of the floor is 430 mm with the equipment at ambient temperature; the distance is reduced to 230 mm with the equipment at the design temperature.

At this particular project, the minimum submergence of the pump is 400 mm. As such, if the tank must be emptied, the pumps can only draw suction down to inventory levels between 630 mm (inventory at the design temperature) and 730 mm (inventory at the minimum salt temperature). After the pumps could no longer draw suction, this would leave something on the order of 1,700 metric tons of salt in the tank.

Pump Tail Pieces

To remove a portion of the remaining salt, one approach is to remove one of the pumps from the tank, install a tail piece on the suction bell, reinstall the pump, and pump down to a level that is as low as the pump can safely operate without vibration. Nonetheless, this approach carries two liabilities:

- All of the salt cannot be removed from the tank due to the flow characteristics of the salt at the entrance to the suction bell; i.e., some minimum submergence will always be required to prevent two-phase flow at the pump inlet

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- The time required to withdraw, and reinstall, the pump is at least 16 hours, as discussed in Section 7.2.5.

Ring Header with Eductors

One possible approach to remove essentially all of the salt is to install a dedicated ring header on the floor at the point of the lowest elevation; i.e., the tank perimeter. The ring header will have a series of eductors, which draw suction from points essentially on the floor.

The motive salt for the ring header in the cold tank is provided by salt from one of the hot salt pumps. The piping connection between the discharge of the hot salt pump and the inlet to the ring header is by a dedicated line, which has blind flanges at each end. The blind flanges are removed only when it is necessary to drain the tank. The discharge of the ring header returns to the hot salt tank.

Similarly, the motive salt for the ring header in the hot tank is provided by one of the attemperation pumps. The discharge from the ring header returns to the cold salt tank.

The motive salt for the eductors in the cold tank is hot salt. However, the hot salt flows only through the venturi; the high temperature salt does not come into contact with the cold tank wall or floor. A similar condition applies to the cold salt supplied to the eductors in the hot tank.

It can be noted that the proposed approach is only conceptual; it has yet to be demonstrated in a commercial project.

Temporary Drain Pump

One approach that has been adopted in 4 commercial projects to remove the last 0.5 m of salt is the following:

- A DN100 hole is drilled in the side of the tank. A pipe is inserted through the hole, and the dimensions of the pipe are such that the end of the pipe reaches the floor
- A temporary cantilever salt pump and sump are installed outside of the tank, at an elevation somewhat below the elevation of the floor
- A second section of temporary pipe is installed, which reaches the bottom of the sump
- The sump is filled with salt, and the pump is started. A partial vacuum is drawn in the temporary line, which establishes a siphon flow in the line

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- The discharge from the temporary pump is directed to a catch basin, where the salt is allowed to freeze.

The approach is effective in removing essentially all of the salt from the tank.

It can be noted that installing temporary drain equipment in response to a leak requires the removal of a section of insulation on the tank wall. This results in 1) a wall temperature that is relatively high beneath the insulation to the left of the section where the insulation has been removed, 2) a wall temperature that is relatively low where the insulation has been removed, and 3) a wall temperature that is relatively high beneath the insulation to the right of the section where the insulation has been removed. For a flat plate, subjected to a temperature gradient of high - low - high, the thermal stress can be estimated as follows:

A rectangular plate or strip $ABCD$ (Fig. 15.5) is heated along a transverse line FG uniformly throughout the thickness and across the width so that the temperature varies only along the length

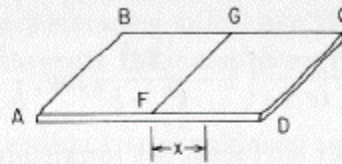


Figure 15.5

with x . At FG the temperature is T_1 ; the minimum temperature in the plate is T_0 . At any point along the edges of the strip where the temperature is T , a tensile stress $\sigma_x = E\gamma(T - T_0)$ is developed; this stress has its maximum value at F and G , where it becomes $E\gamma(T_1 - T_0)$. Halfway between F and G , a compressive stress σ_y of equal intensity is developed along line FG (Ref. 7).

Using representative properties for Type 347H stainless steel, a temperature gradient of 27 °C results in a peak stress equal to the ASME allowable value, and a gradient of 58 °C results in a peak stress equal to the yield stress. These gradients are quite moderate; as such, the potential for damaging the tank wall by removing the insulation is fairly high.

As an alternate approach, the equipment required to drain the last 0.5 m of the inventory can be classified as permanent equipment, which is installed when the storage system is constructed. In this way, the equipment is 1) pre-installed to limit the exposure of plant personnel to unexpected hazards, and 2) insulated to control thermal stresses.

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6.13 *Steam Generator*

The state-of-the-art in steam generator designs is discussed below in Section 7.4.

6.13.1 Introduction

The specification for the steam generator is largely a functional one. Requirements are set for a broad range of items, such as live steam pressure-temperature-flow, reheat steam pressure-temperature-flow, feedwater pressure-temperature-flow, and steam quality. The details of the design, such as heat transfer areas and pressure drops, are then selected by the vendor.

6.13.2 Design Codes

The steam generator is not considered a fired heater; as such, the requirements of ASME Section I are not considered mandatory. The most common design code is ASME Section VIII. Both Division 1 and Division 2 of Section VIII can apply, depending on the locations in the heat exchanger which are subjected to relatively high levels of transient stresses. Nonetheless, it can be noted that California and the city of Albuquerque, New Mexico, have determined that new steam generators, whether fired or unfired, must be designed to the requirements of Section I.

Allowable material stresses are derived from Section II of the ASME Code.

Depending on the process conditions and the dimensions of the heat exchangers, the requirements of the Tubular Equipment Manufacturers Association may also apply.

6.13.3 Process Design

The heat transfer areas, and the pressure drops, in each of the steam generator heat exchangers are selected based on the design point requirements. However, the steam generator necessarily undergoes a startup and a shutdown each day. At the low flow rates characteristic of startup and shut down, the pressure drops on both the shell- and the tube-sides are very low. Low pressure drops generally lead to non-uniform flow distributions. Non-uniform flow distributions produce non-uniform temperature distributions, which, in turn, can lead to unpredictable, and potentially high, local stresses. To a first order, the majority of the heat exchanger failures experienced in solar projects are due to high stresses, typically in the tubesheet, generated during the startup and the shutdown cycles.

In general, it is difficult to calculate, with certainty, the flow and the temperature distributions in the heat exchangers during transient conditions. As a result, it is difficult to predict, with any reasonable certainty, the low cycle fatigue life of the equipment. To prevent these problems, a process design must be developed which ensures that acceptable temperature distributions are known to be present in each of the heat exchangers at a continuous thermal duty equal to 1 percent of the design duty.

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An alternate approach is to establish a set of flows and temperatures in each heat exchanger that are not proven to be safe on a local level, but are known to be safe on a global level. One example is the preheater. Early in the startup process, feedwater flow rates are in the range of 1 to 10 percent of the design flow rate. In this range, it is unlikely that the flow will be distributed uniformly among the ~900 tubes in the heat exchanger. However, it is possible to artificially increase the feedwater volume flow rate by recirculating saturated water from the steam drum to a mixing point upstream of the cold end of the preheater. As an example, the volume flow rate can be increased to 25 percent of the design volume flow rate. The flow distribution will still not be known on a local basis. However, the volume flow rate satisfies the vendor's lower limit on the minimum flow rate (typically 16 percent). As such, the flow and the temperature distributions in the heat exchanger are known to be safe on a global level, and the preheater can safely operate, on a continuous basis, at feedwater flow rates in the range of 1 to 10 percent of the design value.

This design approach can be extended to each of the heat exchangers, as discussed in Section 6.5 of Volume 3 - Narrative.

6.13.4 Material Selection

Preheater

Due to a relatively low design temperature (400 °C), the preheater is typically fabricated from a carbon steel. The choice of the alloy for specific section of the heat exchanger, such as the tubes, will be made by the vendor based on availability, allowable stresses, and ease of fabrication.

Evaporator

The solar industry has generally defaulted to the use of Type 347H as the material for the tubes, the shell, the channel, and the tubesheet in the evaporator. At the design temperatures in the evaporator, Type 347H offers a favorable combination of high strength and good resistance to salt corrosion. Nonetheless, the material is not without potential liabilities. The steam generator is not a fired heater, and as such, does not need to comply with the requirements of Section I of the ASME Code. However, austenitic materials are prohibited in Section 1 for heat exchangers in boiling service due to the potential for chloride stress corrosion cracking.

A range of ferritic materials offer the required resistance to salt corrosion at the design temperature of the evaporator (510 °C). These include 5 Cr - 1 Mo and 9 Cr - 1 Mo, particularly if vanadium is included as an alloying element. The evaporator at the Solar Two project was fabricated using T91 tubes, and performed satisfactorily until the tube bundle ruptured as the result of an error in the process design. The ferritic materials do not use nickel as an alloying element, which makes the material essentially immune to chloride stress corrosion cracking. The ferritic materials are also much less susceptible to pitting corrosion should solid iron oxide materials be transported from the condensate

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system, and accumulate in the evaporator. Pitting corrosion in stainless steel evaporator tubes is a known problem in commercial plants.

Superheater and Reheater

The tubes, the shell, and the tubesheet are fabricated from stainless steel to provide the required resistance to corrosion from salt. The channel is not exposed to salt, but it is also typically fabricated from stainless steel. All of the components are fabricated from an 'H' grade of stainless steel, with a minimum carbon content of 0.04 percent, due to Code requirements for design temperatures above 538 °C.

In principle, the fabrication material could be a non-stabilized stainless steel, such Type 304H or Type 316H, or a stabilized stainless steel, such as Type 321H or Type 347H. However, the non-stabilized materials are subject to intergranular stress corrosion cracking should the heat exchangers, during a maintenance period, be exposed to liquid water. The most likely sources of water would be condensation of moisture in the air if the metal temperatures fell below the dew point, or a hydro test after a tube inspection or after inserting a tube plug. To avoid the potential for intergranular stress corrosion cracking, the solar industry has generally adopted a stabilized stainless steel for the tubes, the channel, the tubesheet, and the shell. The most common choice is Type 347H due to its availability in the shapes required.

It can be noted that the potential problems with stress relaxation cracking of Type 347H in hot salt tanks may not be as prevalent in a heat exchanger. Similar to the bottom course in a hot tank, the heat exchanger requires the welding of thick metal sections to one another. The thickest connection is the weld between the channel and the tubesheet. This requires the use of multiple welding passes, similar to the number of passes in the welds of a hot tank. In terms of stress relaxation cracking, both of these weld regions produce a high risk due to the large residual stresses. However, the heat exchanger, after fabrication, can be heat treated in an oven to reduce the residual stresses. Further, the rates of heating, and the hold times at the required temperatures, can be carefully controlled, which reduces the potential for hot reheat cracking.

6.13.5 Fouling Factors

Fouling factors on the water/steam side of the heat exchangers are generally well known, based on decades of experience with solar steam generators in commercial projects. Values recommended by the vendor are likely the best choice. However, there is little published information, either experimental or theoretical, on the recommended factor for the salt side. The industry has often adopted a default value of 0.000088 m²-°C/W (1/2000 Btu/hr-ft²-°F), but the value should be viewed as an allowance. In a commercial project, the engineering contractor should request the vendor to develop designs based on values of 0.00018, 0.000088, and 0.000059 m²-°C/W (1/1000, 1/2000, and 1/3000 Btu/hr-ft²-°F), and

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then perform a risk analysis to determine the best combination of long-term heat exchanger performance and capital cost.

On a related point, the fouling factors selected for the water side of the preheater and the water/steam side of the evaporator are likely to assume that the feedwater chemistry remains within the design values for pH and dissolved oxygen. However, a number of commercial projects have experienced problems with water chemistry due to operators errors, inadequate operator training, or improperly calibrated monitoring equipment. In a commercial project, the engineering contractor should evaluate the potential for a similar situation to occur, and conduct a risk assessment to determine the best combination of fouling factor, long-term heat exchanger performance, and capital cost.

6.13.6 Feedwater Preheating

The temperature of the feedwater at the cold end of the preheater must always be high enough to prevent salt from freezing in the heat exchanger. At the design point, the final feedwater temperature is always high enough ($\sim 240\text{ }^{\circ}\text{C}$) to satisfy this requirement. However, during turbine startup and low load operation, final feedwater temperatures are in the range of 180 to $230\text{ }^{\circ}\text{C}$, and freezing is likely to occur. To prevent this, one of two forms of feedwater preheating are used in commercial projects:

- A startup feedwater heater is located between the last extraction feedwater heater and the steam generator preheater. The heat source for the startup heater is typically live steam. The live steam is throttled and attemperated prior to delivery to the heater.
- Saturated water from the steam drum is recirculated to the cold end of the preheater. Direct contact heat exchange between the saturated water and the final feedwater raises the temperature of the mixed feedwater flow to the required values.

The first approach is the most common, as it effective and avoids the need for a recirculation pump. However, using live steam for feedwater heating is thermodynamically inefficient, and this form of feedwater heating has a measurable negative effect on the gross cycle efficiency at loads in the range of 60 to 90 percent.

The second approach incurs a smaller thermodynamic penalty than the first, and it avoids the need for an additional heat exchanger. Specifically, the startup heater is exposed to large temperature changes each day, and it is subjected to rapid temperature changes in response to a turbine trip. A leak in the startup heater, or a failure in the instruments associated with the startup heater, will likely lead to a forced outage. However, the second approach requires a recirculation pump, which is not needed in the first approach.

The decision between the two approaches will depend, to some degree, on the selection of the circulation concept for the evaporator. If the evaporator uses forced circulation, then the second

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approach requires only the addition of a set of preheater recirculation pumps in parallel with the evaporator circulation pumps. However, if the project or the vendor selects a natural circulation evaporator to avoid the need for circulation pumps, then the first approach to feedwater preheating is likely to be preferred.

6.13.7 Evaporator Circulation

Commercial projects have successfully used both natural and forced circulation for the evaporator. The decision is normally based on the preferences of the steam generator vendor. Nonetheless, there is an incentive to select a forced circulation design. The benefits are associated with actively controlling temperature profiles in the heat exchangers during startup and low load operation. The topic is discussed in detail in Section 6.5 of Volume 3 - Narrative.

The approach to temperature control discussed in Volume 3 relies, in part, on the recirculation of saturated water from the steam drum to the cold end of the preheater. As such, the second approach to feedwater preheating, discussed above in Section 6.13.6, would be the clear choice.

It can be noted that recirculating water from the steam drum the cold end of the preheater increases the volume flow rate through the preheater. The engineering contractor will need to evaluate each step in the startup of the Rankine cycle to determine the volume flow rate to the preheater at each combination of load, final feedwater temperature, feedwater flow rate, and recirculation flow rate. There may be a stage in the startup process that results in volume flow rate which is larger than the volume flow rate at the 100 percent load condition. If so, the design condition for the preheater may be during an intermediate turbine load, rather than at full load.

6.13.8 Fabrication Approaches

There are several methods available for distributing flow to the tubes in each heat exchanger:

- Conventional, with a flat tubesheet. The tubes are strength welded to the face of the tubesheet, and the tubes are then plastically expanded into the tubesheet in a rolling operation.
- Internal bore welding, with a flat tubesheet. Nozzles are machined from the face of the tubesheet. The ends of the tubes are butt welded to the ends of the nozzle using an internal bore welder.
- Header/coil, with a circular tubesheet. Holes are drilled in a section of pipe, nozzles are welded at each hole, and the ends of the tubes are welded to the ends of the nozzle.

The first approach is the most common, as it is typically the most economical. However, the steam generator necessarily operates through daily temperature cycles. The heat exchanger consists of metal

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sections which range from thin (1.65 mm for the tube wall) to thick (200 mm for the tubesheet). During transient conditions, temperature gradients are established, which lead to stress gradients. The stress gradients can result in flexing of the tubesheet, which causes relaxation of the friction connection between the tube and the tubesheet. If the friction bond is broken, then cyclic loads on the tube-to-tubesheet weld often cause the weld to fail, which leads to leakage from the channel (water/steam) into the shell (salt).

The integrity of the tube-to-tubesheet connection can be maintained if the heat exchangers are operated within the transient limits defined by the vendor. Representative limits include a maximum rate of temperature change of 10 °C/min and a maximum thermal shock of 100 °C. Thermal shock is defined as the difference in temperature between the fluid entering the heat exchanger and the metal temperature in the heat exchanger. The value can be either positive or negative. In general, one thermal shock per day is permitted.

It can be noted that these limits, although specified by the vendor, have been difficult to satisfy in many commercial projects. The topic is discussed in detail in Section 6.4.4 of Volume 3 - Narrative.

The internal bore welding and the header/coil approaches typically result in an increase the capital cost of the heat exchanger; perhaps on the order of 20 to 30 percent. However, each approach offers the following benefits regarding reliability:

- The connections between the tube and the tubesheet are welded, which should be a more robust connection than a friction connection
- The nozzles between the tubesheet and the tubes are typically tapered, and the transition in thickness between the tubesheet and the tube reduces the thermal stresses in this region during transient conditions.

Should a heat exchanger develop a leakage rate that requires a shutdown for tube plugging, the outage period to cool the heat exchangers, effect the repairs, and preheat the equipment in preparation for filling with salt is at least 1 week. The weekly revenue of a typical commercial project is on the order of \$1,000,000. If the selection of an all-welded fabrication approach reduces the number of plant shutdowns required for heat exchanger repair by even one during the 30-year life of the project, the marginal capital cost for all-welded construction is likely to be justified.

6.13.9 Commercially Acceptable Rates of Temperature Change

The allowable rate of temperature change is defined by the vendor, and is generally the result of a low cycle fatigue analysis of the heat exchanger geometry. Representative values range from 8 to 12 °C/min.

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As noted above, some commercial projects have had difficulty meeting a requirement of 10 °C/min. If the heat exchangers have a limit which is lower than 10 °C/min, this will compound the problem. In general, an allowable rate of 8 °C/min is probably too low for commercial consideration.

The engineering contractor must develop a process design that can satisfy, with some margin, the vendor limits on rate of temperature change. To the extent that the engineering contractor has confidence in the control authority of the proposed process design, this may determine the minimum allowable rate specified the procurement package.

6.14 Steam Bypass System and Air Cooled Condenser Capacities

During normal operation, the air cooled condenser rejects heat from the Rankine cycle to the environment. To a first order, a plant with a 125 MWe turbine, operating with a gross cycle efficiency of 0.43, requires an air cooled condenser capacity of 165 MWt.

As discussed in Section 7.4.6, tripping the steam generator, in response to a turbine trip, should be avoided if possible. However, if the steam generator were to remain in operation at full load following a turbine trip, then the duty of the condenser would increase to 290 MWt.

Clearly the project would like to avoid a situation in which the design capacity of the condenser must be 75 percent greater than the normal operating duty.

There is also a second-order problem with the capacity of the steam bypass system. Specifically, the bypass system consists of a high pressure throttle valve, a high pressure water spray attemperator, a low pressure throttle valve, and a low pressure water spray attemperator. The latent heat transfer in the high pressure attemperator mimics the live steam enthalpy drop across the high pressure turbine, and the latent heat transfer in the low pressure attemperator mimics the hot reheat steam enthalpy drop across the intermediate / low pressure turbine. Spray water for the high pressure attemperator is provided by the feedwater pump, and spray water for the low pressure attemperator is provided by the condensate pump.

The latent heat duty of the low temperature attemperator is such that required spray water flow rate plus the condensate flow rate from the condenser is about 130 percent of the condensate flow rate with the turbine operating at the design point. The condensate pump can be sized for the 130 percent condition. However, the pump would then be operating at approximately the 75 percent flow condition throughout the life of the project.

To avoid this situation, the capacity of the steam bypass system can be selected such that the required spray water flow rate to the low pressure attemperation plus the condensate flow rate is equal to the condensate flow rate when the turbine is operating at the design point. This can be accomplished if the live steam flow rate following a turbine trip is reduced to about 70 percent of the design flow rate.

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The live steam flow rate can be reduced to 70 percent of the design value by reducing the flow rate of hot salt to the steam generator by 30 percent; i.e., to 70 percent of the design value. The salt flow rate can be changed in manner of seconds, as the temperature profile along the steam generator flow path remains unchanged. However, the allowable rate at which the salt flow rate can be reduced is likely governed by level stability considerations for the steam drum.

If the live steam flow rate is reduced to 70 percent of the design value, then the condenser duty will be on the order of 200 MWt. This is about 20 percent greater than the design duty. However, by raising the saturation pressure in the condenser somewhat above the design value, the log mean temperature difference with the ambient increases, which increases the duty of the heat exchanger.

It can be noted that a steam bypass flow capacity equal to 70 percent of the design live steam flow rate is only a notional value. The actual value will be determined by the following:

- Transient level characteristics in the steam drum
- The potential availability of control valves, acting as pressure relief valves, on the steam drum to accommodate short term (on the order of minutes) increases in the drum pressure
- The ability of the air cooled condenser to accommodate saturation pressures above the design value.

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7. Current State of the Art

A summary of the current state of the art in nitrate salt central receiver technology is outlined below. The principal components described include receivers, salt pumps, thermal storage tanks, tank foundations, and steam generators. Topics include process designs, equipment designs, temperatures, rates of temperature change, efficiencies, materials, heat tracing, instruments, and valves. The discussion is, in essence, a compilation of functional and prescriptive specifications that incorporate lessons learned from successful commercial plants.

7.1 Receiver System

7.1.1 System Description

The receiver system consists of the following elements:

- Receiver circulation pumps, for supplying cold salt to the receiver inlet vessel. The vertical turbine pumps are mounted on top of the cold salt storage tank, and are driven by an electric motors with variable speed drives
- Receiver inlet vessel, for supplying a temporary flow to the receiver in the event of a loss of the receiver pump or site power
- Receiver absorber panels, consisting of a group of parallel tubes, upper and lower headers, support structure, upper and lower oven boxes, insulation, and temperature instruments. Multiple panels in series are required, generally arranged in 2 parallel flow paths
- Receiver outlet vessel, located above the top of the absorber panels. The vessel 1) collects the outlet flows from the two receiver circuits, 2) provides a level control signal to the throttle valves at the base of the downcomer, which, in turn, maintains the downcomer in a flooded condition, and 3) collects and stores the flow from the inlet vessel should the flow in the downcomer become blocked
- Internal receiver piping, including inter-panel piping, crossovers, valves, fill and drain lines, and inline instruments for flow, pressure, and temperature
- Receiver structural supports, ladders, and platforms
- Receiver tower crane to allow access to receiver panels for installation and replacement.

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A schematic equipment arrangement is shown in Figure 7-1, and details of the pump, vessel, and valve arrangement in a representative commercial project are shown in Figure 7-2.

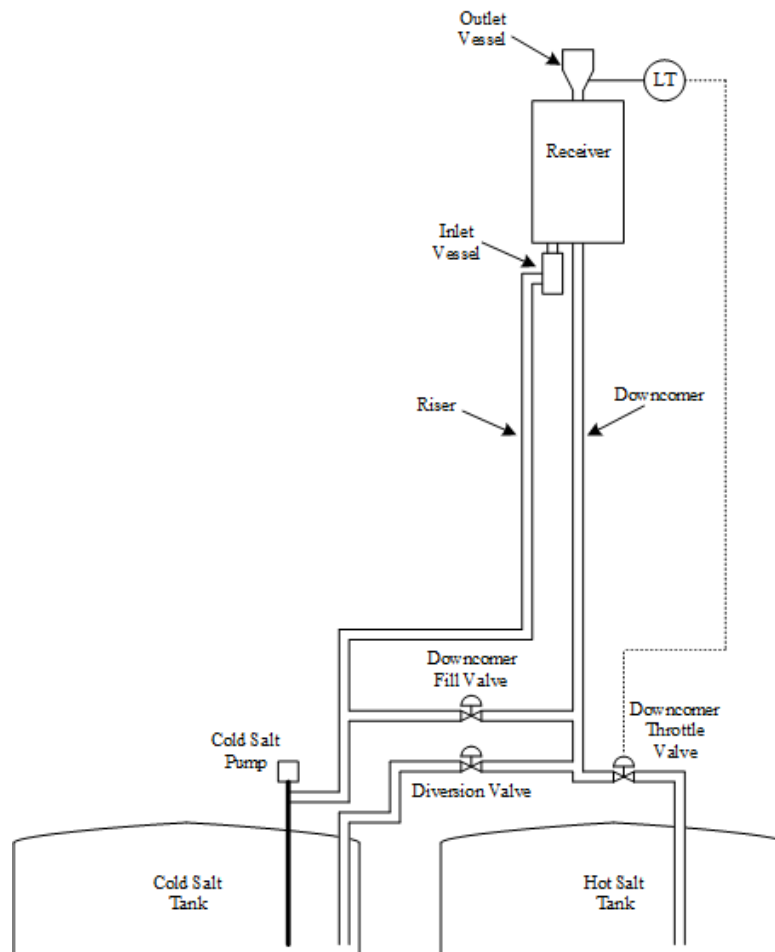


Figure 7-1 Receiver Outlet Vessel and Control Valve at the Solar Two Project

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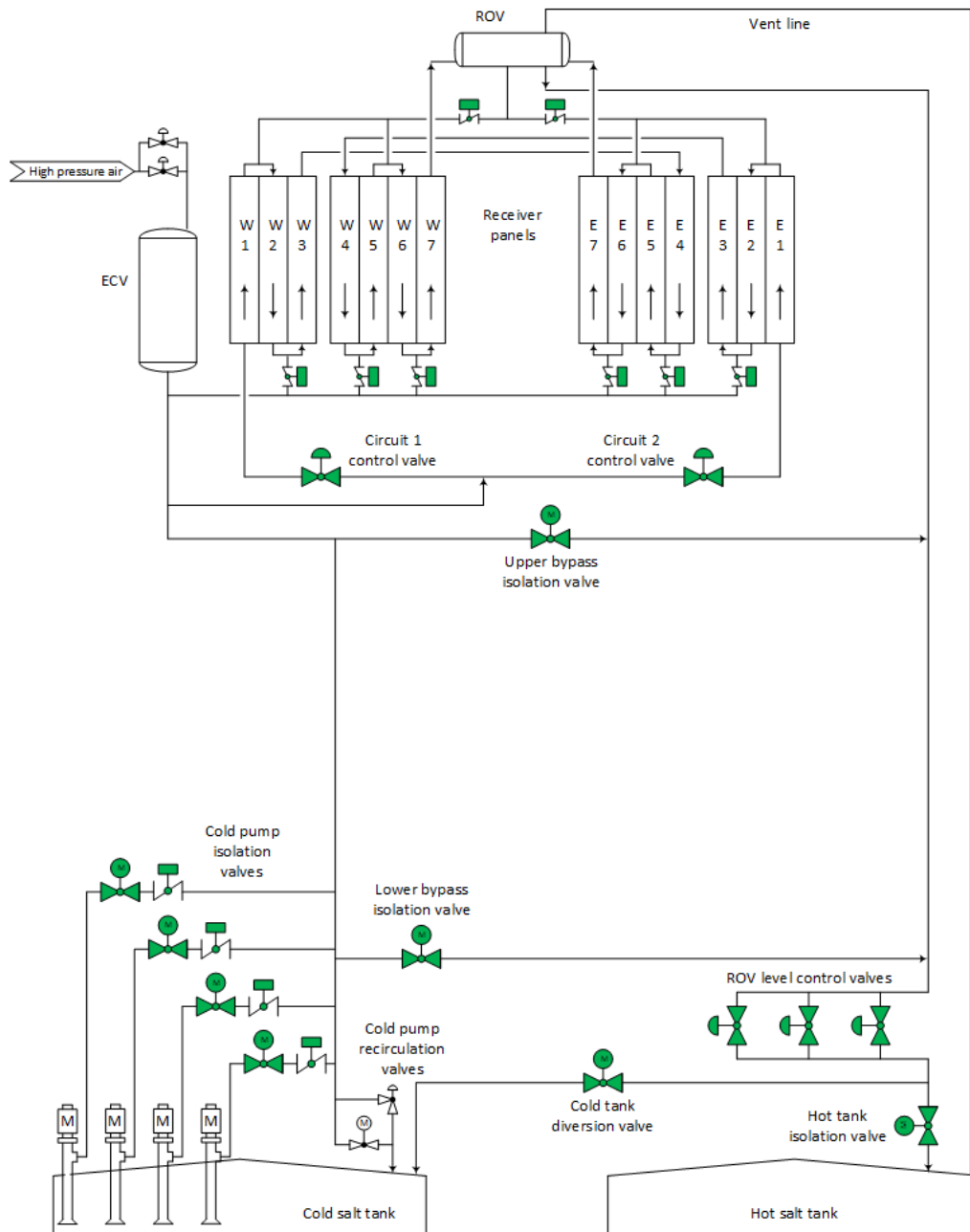


Figure 7-2 Receiver Equipment Arrangement in a Representative Commercial Project

An alternate approach, in which the flow in the downcomer is throttled at numerous locations, is illustrated in Figure 7-3. Throttling is provided by orifice plates, with a nominal vertical spacing of 6 m. At full flow rate, the liquid level above each plate approaches 6 m, providing the necessary head for the flow restriction. At lower flow rates, the liquid level above each plate is re-established at a lower height.

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There are no active flow control mechanisms in the downcomer; hence, there is no requirement for an outlet vessel.

The basic concept of periodic orifice plates in a downcomer was successfully demonstrated in an experiment at Sandia National Laboratories using a transparent plastic pipe, 50 m tall and 150 mm in diameter, and water. Stable liquid levels above the orifice plates were established over a wide range of flow rates, and pipe vibrations were well within acceptable values.

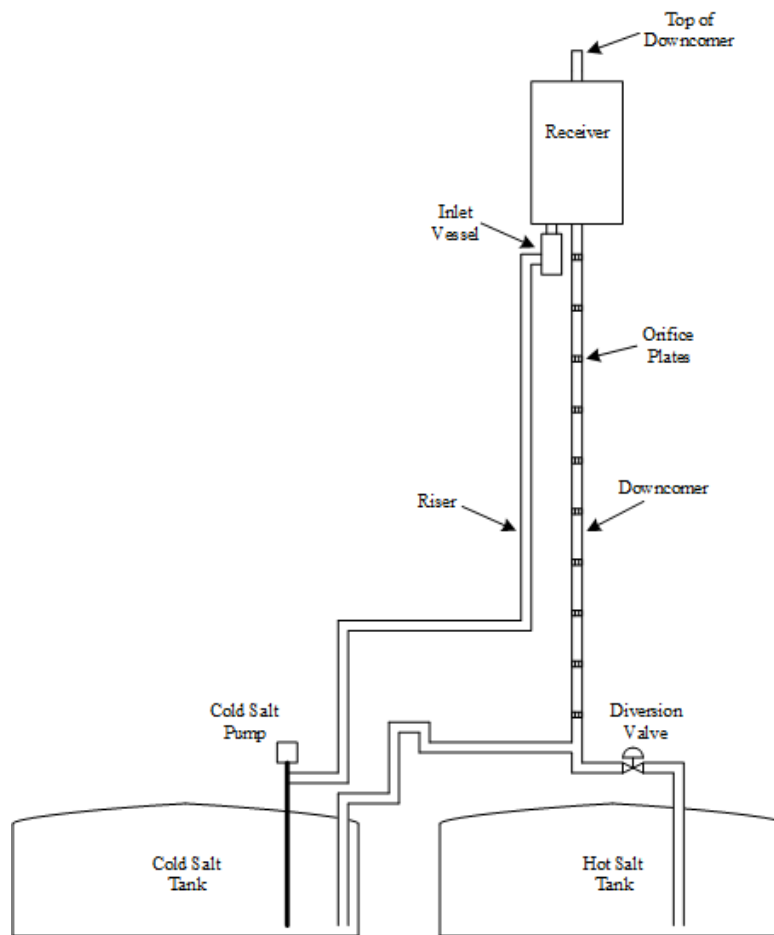


Figure 7-3 Downcomer with Periodic Orifice Plates for Throttling

Periodic orifice plates were selected for the downcomer design at the Crescent Dunes project. However, the downcomer supports and anchors were not designed to accommodate the momentum loads associated with salt. Rather than redesign and replace the support and anchors, throttle valves were

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added at the base of the downcomer, and the downcomer operated with a mix of two-phase flow in the upper section, and single-phase flow in the lower sections.

It can be noted that both types of receiver designs (outlet vessel or downcomer with orifice plates) require a vent line, which connects the top of the receiver with the top of the hot salt tank. The design requirements for the vent line, described below in Section 7.1.13, are rigorous. To a first order, the design difficulties and risks associated with a vent line are comparable to the design difficulties and risks associated with a downcomer with orifice plates. Stated another way, selecting a receiver design with an outlet vessel is no less complex than selecting a design with a downcomer using orifice plates.

7.1.2 Scope of Supply

The receiver procurement package includes the following items:

- Development of allowable flux as a function of the circumferential position on the receiver
- Selection of absorber dimensions, in conjunction with the optimization of the heliostat field layout
- Selection of the number of panels, number of tubes per panel, tube diameter, and tube wall thickness
- Calculation of thermal efficiency at various flux distributions
- System hydraulic calculations, both steady state and transient
- Development of process flow diagrams, and piping and instrument diagrams
- Analysis and design of inlet vessel
- Layout of inter-panel, crossover(s), fill lines, drain lines, valves, supports, and anchors
- Design of thermal insulation
- Design and layout of electric heat trace circuits, thermocouples, and radiant heaters
- Specification of instruments and control valves
- Structural design of panel supports, central structure, and receiver tower crane
- SAMA (Scientific Apparatus Makers Association) diagrams for all operating states, and transitions between states

Receiver panel fabrication package includes the support frames, tubes, headers, nozzles, tube clips, oven boxes, electric heat tracing, radiant heaters, thermal insulation, and instruments. The panel fabricator will be under contract to provide oversight during installation and testing.

Receiver installation package includes the receiver panels, inlet vessel, primary structural support, crane, piping, valves, instruments, programmable logic controller, electric power wiring, control wiring, thermal insulation, testing, and startup.

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7.1.3 Configuration

Two configurations of salt receivers have been tested and demonstrated: internal cavity; and external cylinder.

In either configuration, the receiver is drained at the end of the day to 1) limit thermal losses during the overnight hold period and 2) to prevent freezing due to convection heat transfer or degradation in the thermal insulation. As such, some means must be developed for preheating the receiver panels prior to filling with salt in the morning. Preheating can be accomplished by 1) focusing a limited number of heliostats on the receiver, or 2) activating a set of radiant electric heaters. The goal is to provide a nominally uniform panel temperature in the range of 300 to 350 °C. The minimum temperature of 300 °C ensures that salt does not freeze on the coldest portion of the tube; i.e., the inside of the tube at the back of the tube. The maximum temperature of 350 °C limits the thermal shock to the tube and limits the transverse temperature gradient across the panel. The transverse gradient must be controlled to limit bending loads on the panel header. If preheat flux, whether provided by the heliostat field or by the radiant heaters, is in the range of 30 to 60 kW/m². The range of fluxes is dictated by the wind speed and the wind direction.

It can be noted that a preheat flux of 30 kW/m² is only 2 to 5 percent of the incident design flux, depending on the position on the receiver. To provide incident fluxes in this range, and to simultaneously maintain the panel temperatures in the range of 300 to 350 °C, accurate control over heliostat image sizes and locations is required.

Internal Cavity Designs

Cavity designs typically use panels located at the back wall, the roof, the floor, and the sides. The back wall has the highest view factor to the heliostat field, and as such, receives the highest fluxes. The floor generally has the lowest view factor, and receives the lowest fluxes.

Due to the range of view factors, a preheat flux from the heliostat field will not be distributed uniformly among the panels. In addition, the heliostats which provide the most accurate flux distribution are the heliostats with the smallest images sizes, and these heliostats are those closest to the tower. However, the view factors between these heliostats and the floor of the cavity are very small. As such, it is not possible to select a subset of the heliostat field that can provide a preheat flux distribution that meets all of the requirements for filling.

To overcome this problem, cavity receivers typically provide a door at the aperture in combination with radiant heaters. The door is open for normal operation, and closed at night. To preheat the receiver, the door is closed and the radiant heaters are activated. The door reduces convection and radiation losses from the cavity to values that allow radiant heaters of reasonable capacity. When the cavity reaches a

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uniform temperature of, say, 300 °C, the receiver panels are filled and vented. Once stable flow conditions are reached, the door is opened and normal operation can begin.

As might be expected, preheating the receiver in a reasonable time with a reasonable radiant heater capacity requires control over convection heat losses from the cavity. This, in turn, requires an effective seal between the cavity structure and the door. However, as the size of the door increases, it becomes more difficult to control the shape of the door. Specifically, the door will be exposed to a range of spillage fluxes, including the following:

- Normal spillage fluxes associated with design values for the heliostat pointing errors and the mirror slope errors
- Higher than normal spillage fluxes associated with 1) heliostat optical errors that exceed design values, 2) transient fluxes following a trip of the heliostat field, and 3) transient fluxes following a loss of electric power to the heliostat field.

For small receivers (< 25 MWt), the width and height of the doors are a few meters, and the door has sufficient structural stiffness to provide a consistent seal. However, once a receiver crosses a threshold in size (perhaps 50 MWt), the dimensions of the door reach values that the shape of the door can be altered by the spillage fluxes and an effective seal can no longer be assured.

The market for commercial central receiver projects has evolved to one in which has the following characteristics:

- The minimum plant size is on the order of 100 MWe to reduce the annual unit operation and maintenance cost, in \$/kWhe, to affordable levels
- A plant without thermal storage cannot compete with photovoltaic projects on a \$/kWhe basis. As such, all commercial plants include some form of thermal storage (at least 3 hours), which allows the plant to supply electric energy to the local utility under a different dispatch strategy than a photovoltaic project.

These characteristics lead to required receiver capacities in the range of 400 to 600 MWt. In principle, it might be possible to develop a 500 MWt receiver concept using multiple cavities. The cavities would be located at different circumferential positions on the tower to 1) reduce the distance from the furthest heliostat to the receiver, and 2) reduce the height of the tower. However, the physical complexity, and the associated effects on cost and reliability, have led to an alternate commercial approach based on an external cylinder.

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External Cylinder Designs

The basic arrangement of an external receiver is a vertical cylinder, with a height-to-diameter ratio of about 1.5, as shown in Figure 7-4.

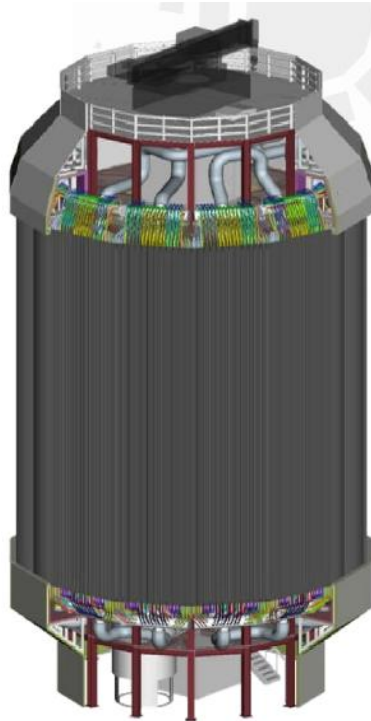


Figure 7-4 External Salt Receiver Arrangement

The heat transfer characteristics of nitrate salt are decidedly ordinary. To raise the temperature of the salt from the inlet value of 295 °C to the design outlet value of 565 °C, an overall heating length of 125 to 150 m is required. Fabricating a single receiver panel with this vertical dimension is impractical, for the following reasons:

- Receiver tubes can be fabricated in lengths up to perhaps 30 m. A panel length of 150 m would require 4 tube butt welds in the flux zone. Welding a tube changes both the metal chemistry and the grain size, both of which have detrimental effects on the low cycle fatigue life of the tube
- A single panel will have high spillage losses at the sides of the panel

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- A single panel would require that all of the heliostats be located North of the panel. For a receiver duty of 500 MWt, the optical efficiency of the most distant heliostats would fall below the point where the annual energy delivered to the receiver supported the capital investment in the heliostat.

To avoid these problems, the receiver is fabricated with 14 or 16 shorter panels, arranged side by side in a cylindrical arrangement. The hydraulic configuration uses two flow circuits in parallel, with each flow circuit consisting of 7 or 8 panels in series. A crossover is located at the mid-point of each circuit, as illustrated in Figure 7-5. The goal is to provide nominally equal thermal inputs to both flow circuits during the morning and the afternoon hours. (A note: The figure represents a flow arrangement for a plant located South of the equator. For plants located North of the equator, the flow enters at the North side, and exits at the South side.)

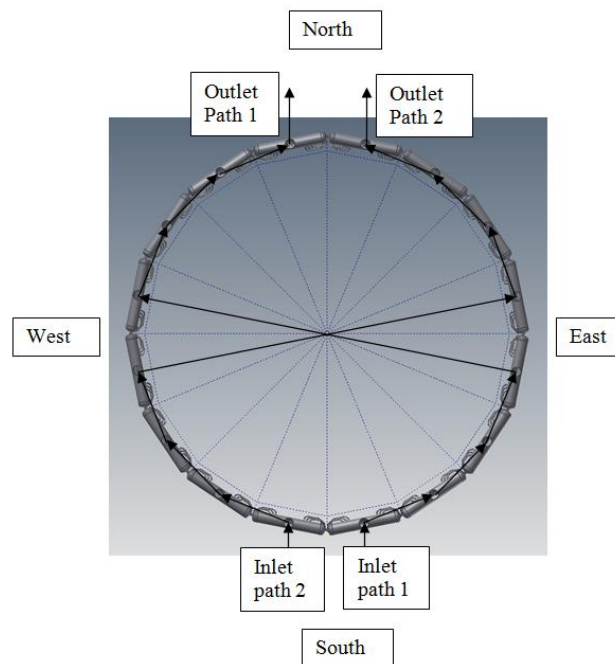


Figure 7-5 Circuit Flow Paths with Single Crossover Between Circuits

7.1.4 Flux Distribution

Circumferential Flux Distribution

As discussed below in Section 7.1.9, the tube is heated from one side. This produces a temperature difference between the front and the back of the tube, and generates a temperature gradient through the

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front (crown) of the tube. Since the tube material has a non-zero coefficient of thermal expansion, the temperature gradients establish strains around the circumference of the tube and strains through the crown of the tube. The ability of the material to accommodate the low cycle creep damage and fatigue damage generated by the strains is a function of the material properties and the metal temperature. In general, the lower the metal temperature, the lower the rate at which creep and fatigue damages are accumulated.

At the inlet to the receiver, the salt temperatures are the lowest. Accordingly, the ability of the tube material to accommodate creep and fatigue damage is the highest at this location. As such, the allowable strains are also the highest at this location, which means that the allowable incident fluxes are the highest at this location. In commercial plant designs, design fluxes near the equator of the receiver are in the range of 1,100 to 1,300 kW/m².

In contrast, at the outlet of the receiver, the salt temperatures are the highest. This places two limits on the design flux near the outlet of the receiver:

- The ability of the tube material to accommodate creep and fatigue damage is the lowest at this location. The allowable strains are also the lowest at this location, which means that the allowable incident fluxes are the lowest at this location
- The maximum allowable film temperature in the receiver tube is on the order of 600 °C. The temperature limit ensures that the steady-state oxide concentration in the salt does not reach values that would result in corrosion rates that are higher than commercially acceptable.

To comply with both of these requirements, design fluxes near the equator of the receiver are limited to values in the range of 600 to 700 kW/m².

Vertical Flux Distribution

Ideally, the vertical flux distribution along the height of a panel would consist of a single value, equal to the peak allowable flux at each circumferential position on the receiver. However, the reflected image from a heliostat has something of a Gaussian distribution. If a uniform vertical flux was applied to a panel, the spillage losses above and below the absorber would be in the range of 600 to 1,200 kW/m². Fluxes in this range, applied to uncooled surfaces, would result in structural damage to the panel ovens and the adjoining structures.

To prevent this situation, the incident fluxes decrease with distance from the equator. The flux limits near the top and the bottom of the absorber are defined by the following limits:

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- The incident flux must be high enough such that the incident power is as least as high as the reflection, reradiation, and convection losses; i.e., the ends of the absorber section must have a thermal efficiency greater than zero
- The spillage fluxes incident on the oven boxes must not exceed values which would cause thermal damage to uncooled surfaces.

Depending on the circumferential position on the receiver, the incident fluxes at the top and the bottom of the absorber typically fall in the range of 75 to 150 kW/m².

Flux Maps

The distribution of the incident flux on the panels near the inlet of the receiver flow circuit (Panels 1 and 2) is shown in Figure 7-6. The flux values are for noon on the summer solstice, and represent the design point conditions for the receiver. Similarly, the distribution of the incident flux on the panels near the outlet of the flow circuit (Panels 7 and 8) are shown in Figure 7-7.

INCIDENT HEAT FLUX MAP		Day of Year: 182				Time of Day: 12:00:00			
10/5/11 R4									
	PANEL 1E				PANEL 2E				
Radial Position (4)	0	7.46	15.04	22.50	22.50	29.96	37.54	45.00	
Elevation (3)									
17.983	150	150	151	151	151	151	148	145	
17.146	263	263	266	266	266	267	263	258	
16.310	412	412	419	421	421	423	419	412	
15.473	582	582	595	599	599	603	599	591	
14.637	748	748	768	774	774	781	780	770	
13.801	892	892	918	927	927	936	938	928	
12.964	1004	1004	1033	1044	1044	1056	1060	1051	
12.128	1085	1085	1114	1126	1126	1139	1146	1138	
11.291	1139	1139	1167	1180	1180	1194	1202	1194	
10.455	1175	1175	1201	1215	1215	1228	1237	1230	
9.619	1197	1197	1221	1235	1235	1249	1258	1250	
8.782	1206	1206	1229	1244	1244	1258	1266	1258	
7.946	1199	1199	1223	1237	1237	1251	1259	1250	
7.109	1170	1170	1195	1209	1209	1223	1229	1219	
6.273	1112	1112	1138	1151	1151	1164	1167	1156	
5.437	1015	1015	1042	1053	1053	1064	1064	1051	
4.600	880	880	904	913	913	922	916	904	
3.764	713	713	733	739	739	745	736	724	
2.927	535	535	549	552	552	555	544	533	
2.091	367	367	376	377	377	378	367	358	
1.255	231	231	235	235	235	235	225	220	
0.418	134	134	136	135	135	134	128	124	

Figure 7-6 Representative Flux Map for Panel 1 and Panel 2; Summer Solstice at Noon

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INCIDENT HEAT FLUX MAP Day of Year: **182** Time of Day: **12:00:00**

10/5/11 R4

	PANEL 7E				PANEL 8E			
Radial Position (4)	135.00	142.46	150.04	157.50	157.50	164.96	172.54	180.00
Elevation (3)								
17.983	100	97	91	89	89	86	84	84
17.146	177	172	160	155	155	150	143	143
16.310	286	278	258	249	249	240	228	228
15.473	417	405	377	363	363	350	331	331
14.637	555	539	501	482	482	463	437	437
13.801	681	661	613	589	589	565	532	532
12.964	781	758	702	674	674	646	607	607
12.128	852	826	764	733	733	703	661	661
11.291	896	868	802	770	770	738	695	695
10.455	919	890	822	789	789	756	713	713
9.619	925	896	827	793	793	759	716	716
8.782	917	887	818	784	784	750	706	706
7.946	896	866	796	762	762	728	685	685
7.109	860	830	762	729	729	695	653	653
6.273	807	779	715	682	682	650	609	609
5.437	730	707	651	621	621	592	553	553
4.600	626	609	566	541	541	517	482	482
3.764	497	487	460	442	442	425	397	397
2.927	359	355	342	331	331	321	301	301
2.091	234	232	228	223	223	218	206	206
1.255	137	136	135	133	133	132	127	127
0.418	74	73	73	72	72	72	70	70

Figure 7-7 Representative Flux Map for Panel 7 and Panel 8; Summer Solstice at Noon

Over the middle third of each panel, the incident fluxes are close to the peak value at the equator. The incident fluxes decay rapidly across the top third and the bottom third of the panel to meet the spillage limits.

7.1.5 Design Temperatures

Receiver design temperatures are listed below in Table 7-1.

Table 7-1 Receiver Design Temperatures

	Maximum Temperature, °C		
Operating Condition	Bulk	Film	Cumulative time
Steady state	580	600	Unlimited
Transient (5 minutes)	602	616	2,500 hours in 30 years
Trip	616	630	1 hour in 30 years

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The nominal receiver inlet and outlet temperatures are 298 °C and 565 °C, respectively.

Design temperatures for equipment and piping upstream and downstream of the receiver are as follows:

- Cold salt loop piping design temperature is 370 °C. This includes the cold salt pumps, the receiver inlet vessel, the supply piping to the receiver, and the upper bypass piping up to the isolation valve
- The minimum cold salt loop design temperature is 260 °C
- Hot salt loop piping design temperature is 593 °C. This includes the outlet piping from the receiver, the crossover piping, the downcomer, and bypass piping up to, and including, the isolation valve
- The minimum hot salt loop design temperature is 260 °C.

7.1.6 Materials

Unlike the heat collection elements in parabolic trough collectors, the tubes in salt receivers cannot take advantage of a vacuum enclosure to control convection losses. Essentially, the only mechanism to control thermal losses in a salt receiver is to make the absorber area as small as possible. This, in turn, requires the selection of incident fluxes that are as high as possible.

The receiver tube is heated only from the front. This establishes a circumferential temperature gradient which, in turn, establishes a circumferential stress gradient. The tube is also constrained to remain in the plane of the absorber. The goals of the tube restraint are to 1) limit incident fluxes which might pass in the gaps between tubes, and 2) prevent the tube from fluttering during windy conditions, which can lead to a failure from high cycle fatigue. The combination of the stress gradient and the constraint places the front of the tube in compression and the back of the tube in tension during normal operation.

If the tube stresses are limited to the values listed in Section II of the Code, then the incident fluxes are limited to values in the range of perhaps 400 to 500 kW/m². This, in turn, results in a relatively large absorber area, and a corresponding thermal efficiency of about 75 to 80 percent. Efficiency values in this range are too low for commercial consideration. However, if the tube stresses can approach, or exceed, the yield stress, then incident flux levels as high as 1,200 kW/m² can be accepted. This, in turn, reduces the absorber area such that the receiver efficiency increases to commercially-feasible values of 88 to 90 percent.

During the first cycle in which the receiver operates at the design flux, the front of the tube yields in compression, and the back of the tube yields in tension. During the first shutdown, the tube

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temperatures return to ambient and the stress distribution reverses. On the following day, the absorber is again exposed to flux from the heliostat field. However, at startup, the front of the tube begins in a state of tension due to the plastic compressive deformation carried over from the first day. This, in turn, reduces the peak compressive stress on the front of the tube when the flux from the field reaches the design value. This pattern is known as a shakedown cycle. On Day 3, the stress distribution from Day 2 repeats. The cyclic loading pattern is illustrated in Figure 7-8⁴.

The process is also illustrated in a Bree diagram, as shown in Figure 7-9. The abscissa is the primary mechanical stress, and the ordinate is the secondary thermal stress. The elastic shakedown step is the plastic deformation on the first day of operation. Ideally, receiver operation during the balance of the project life is shown as the low cycle fatigue region of the diagram. It is important to limit the stresses in the tubes to values which avoid the ratcheting portion of the diagram.

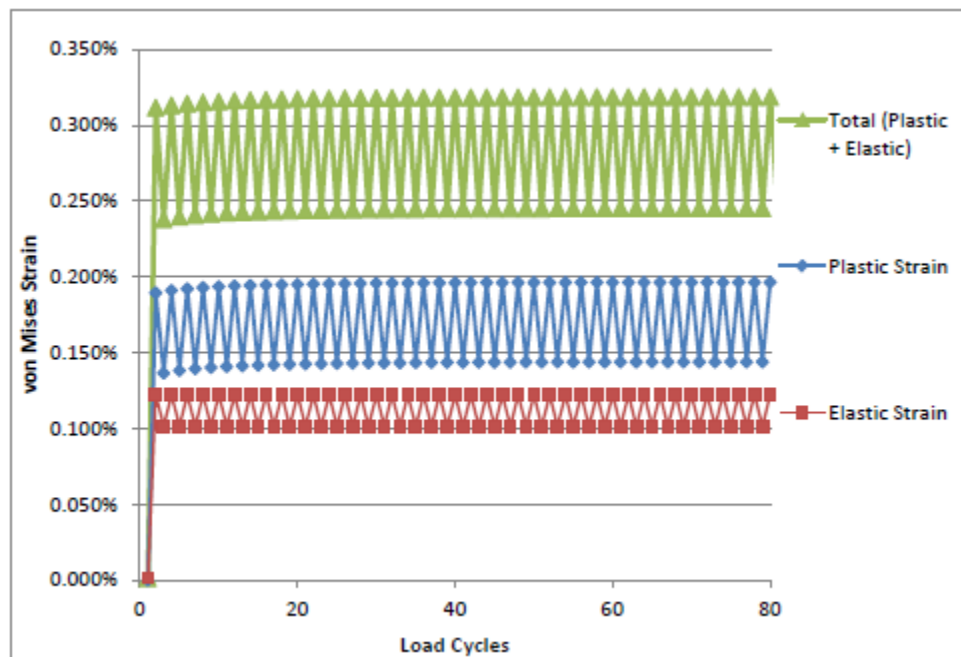


Figure 7-8 Cyclic Tube Strains on Successive Cycles

⁴ Greenhut, D., et. al., (Foster Wheeler North America Corporation, Clinton, New Jersey), "Baseload Concentrating Solar Power Generation Molten Salt Solar Receiver and Steam Generator Design, Analysis, and Cost Final Report", Work performed under DOE Cooperative Agreement DE-EE0003596, March 2012

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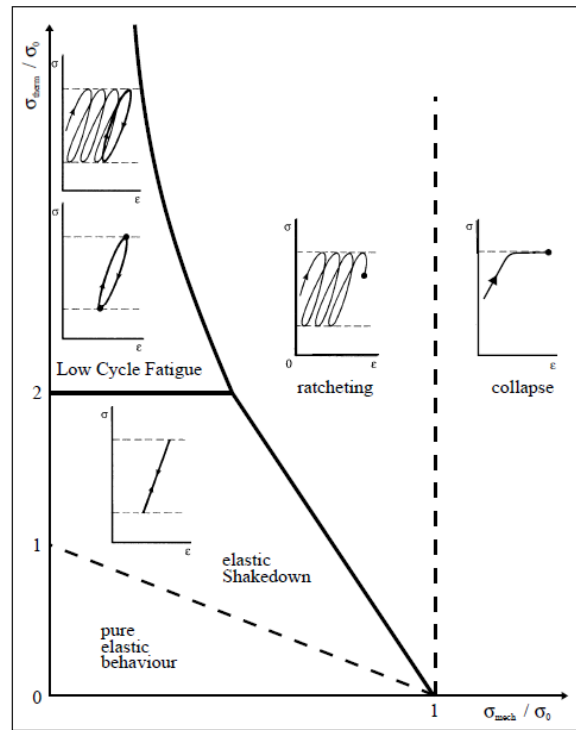


Figure 7-9 Bree Diagram of Primary Mechanical and Secondary Thermal Stresses

The only materials identified to date which provide the required damage tolerance are Alloy 230 and Inconel 625 LCF. Both are solution-strengthened nickel alloys, and both are available from a range of commercial suppliers,

The LCF in Inconel 625 LCF refers to low cycle fatigue. Inconel 625 and 625 LCF are identical in terms of alloy composition, but the levels of carbon, silicon, and nitrogen in the LCF version, together with the microstructure, are controlled to improve the fatigue life. Nonetheless, Alloy 230 is generally preferred for commercial receivers due to a higher thermal stability, and an increased resistance to forming intermetallic compounds, such Fe-Cr, Fe-Mo, and $(\text{FeNi})_x(\text{CrMo})_y$. These compounds increase the strength, but reduce the ductility, of the alloy, and the latter characteristic is more beneficial than the former.

7.1.7 Tube Design Parameters

A representative duty for a commercial receiver is 500 to 600 MWt. This translates to nominal tube diameters in the range of 50 to 60 mm. The tube wall thickness should be as thin as possible. This reduces the thermal resistance of the tube wall, which, in turn, reduces the through-the-wall temperature gradient and the compressive stresses at the tube crown. However, the tube wall cannot be so thin that the tube cannot be welded due to potential burn through, or so thin that the tube cannot be handled due

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to the potential for buckling. The industry has adopted a reference wall thickness of 1.5 to 1.65 mm to accommodate this range of requirements.

For equipment in hot salt service, a representative corrosion allowance is 0.7 mm. This applies to both stainless steels and nickel alloys. If these materials are operating at cold salt temperatures, the expected corrosion rate likely decreases by two orders of magnitude. If the receiver tube has a wall thickness of 1.5 mm, then the expected wall thickness at the end of the project is on the order of $1.5 \text{ mm} - 0.7 \text{ mm} = 0.8 \text{ mm}$. A 50 mm tube, with a wall thickness of 0.8 mm, can withstand pressures up to perhaps 25 bar before exceeding allowable Code stresses. However, the cold salt pumps, under certain conditions, can establish pressures at the inlet to the receiver which are greater than 25 bar. Nonetheless, the receiver avoids potential tube stress limitations, for the following reasons:

- The inlet to the receiver operates at the highest pressures, but at the lowest temperatures. Over the life of the project, the cumulative corrosion is expected to be well below 0.7 mm. As such, the wall thickness will remain close to the original value of 1.5 mm, and the tube stresses will remain within allowable limits under all of the potential salt pump operating conditions
- The outlet from the receiver operates at the highest temperatures, but at the lowest pressures. Over the life of the project, the cumulative corrosion is expected to be the full 0.7 mm. However, the fluid pressures near the outlet of the receiver are on the order of 2 bar (absolute), and the tube stresses will remain within allowable limits even with the reduced wall thickness.

7.1.8 Tube Fabrication

Seamless tubes are not commercially available with the above combination of diameter and wall thickness. As such, the tubes are fabricated by cutting strips from a flat sheet that is 1.5 to 1.65 mm thick. The width of the strip is $(\text{Tube diameter} * \pi + \text{Allowance for cold reduction in a subsequent process step})$, and the length of the strip is the distance from the between the upper and the lower panel headers plus an allowance for the lengths of the jumper tubes. The strips are rolled into cylinders, and the seam is joined in an autogenous weld using tungsten electrodes in an inert gas.

The welding process normally leads to an undesirable coarsening of the grains, which increases the strength but decreases the ductility. Welding can also lead to component segregation (liquation) due to alloy migration and variations in solubilities as the weld cools. Segregation can produce zones with high ductility / low strength adjacent to zones with low ductility / high strength, which can promote hot reheat cracking. To remove these detrimental effects, the tubes undergo the following post-welding steps:

- The tube is drawn to a smaller diameter, with a cold work of 15 to 25 percent, to mechanically reduce the size of the grains

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- The tube is solution annealed to remove the residual stresses, redistribute the alloy components, and recrystallize the grains in the desired size.

7.1.9 Combined Low Cycle Fatigue and Creep Damage

Low cycle fatigue data are available for a wide range of materials. However, the data are often collected with strain reversals occurring every 3 seconds; i.e., 20 cycles per minute. This frequency is adopted to collect fatigue data on the order of 100,000 cycles in a reasonable period of time (1 week). An example for Alloy 230, over a range of temperatures, is shown in Figure 7-10⁵.

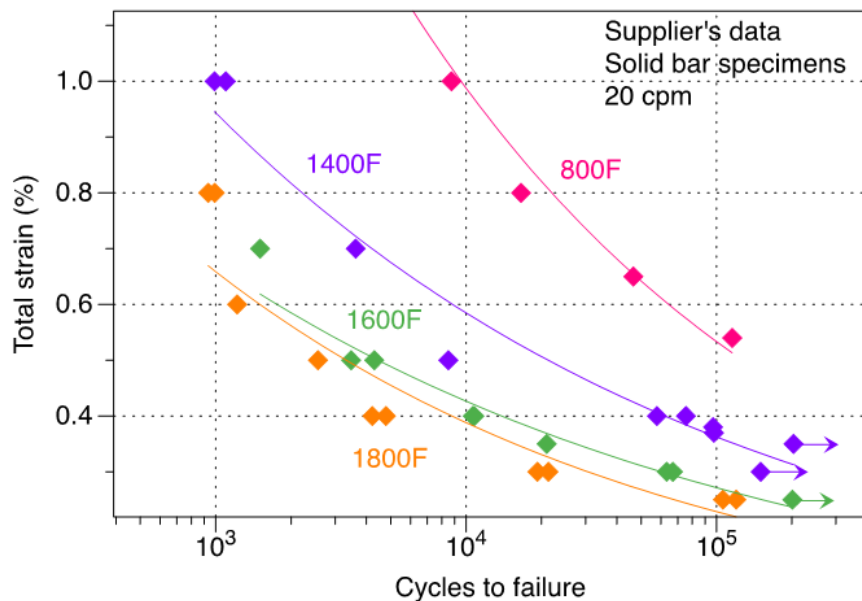


Figure 7-10 Low Cycle Fatigue Data for Alloy 230

It can be noted that adding a hold time to the strain tests has a significant influence on the fatigue life. Adding a hold time essentially adds a level of creep damage to the tests. One example is shown in Figure 7-11, in which a hold time of 2 minutes is added to the cycle time. The curves of interest are the ones labeled '1115F ORNL Welded pipe' and '1115F ORNL Welded pipe 2 min hold compression'. With a total strain of 0.4 percent, the former case has a low cycle fatigue life of 120,000 cycles, while the latter case has a fatigue life reduced by an order of magnitude to 12,000 cycles.

⁵ Keiser, J., et al, (Oak Ridge National Laboratory, Oak Ridge, Tennessee), 'Corrosion Fatigue Studies of High Nickel Tubular Samples Containing Molten Salt', TMS2013, 142nd Annual Meeting & Exhibition, San Antonio, Texas, March 2013

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In a commercial receiver, the hold times are on the order of several hours, rather than 2 minutes. The longer hold time is also expected to have a detrimental effect on the fatigue life. However, the fatigue data are very difficult to collect, as a test lasting 10,000 cycles would a test period of at least 6 years.

As such, the combined low cycle fatigue / creep life of a receiver tube in commercial service is poorly defined, and the current generation of receivers are, in effect, a set of novel experiments running in real time.

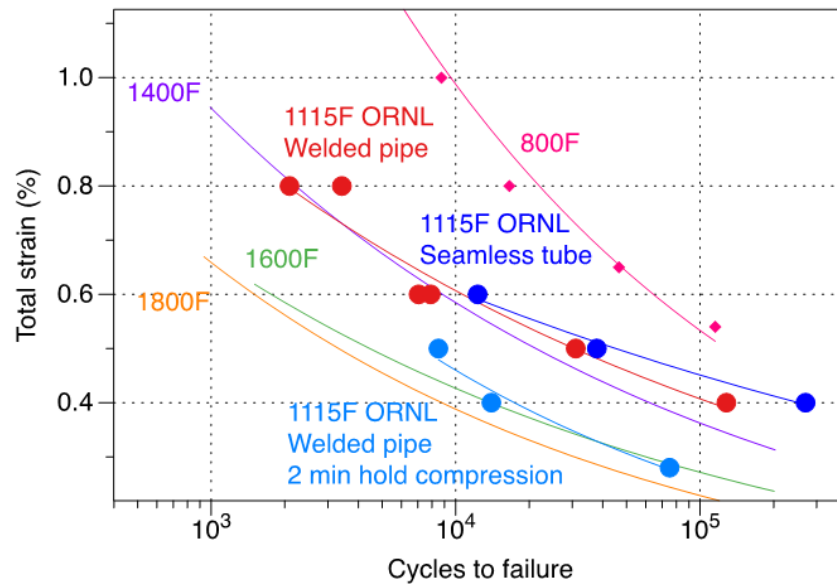


Figure 7-11 Low Cycle Fatigue Data for Alloy 230 with a 2 Minute Hold Time

7.1.10 Header Fabrication

Each panel consists of a series of tubes in parallel. The flow to the tubes is distributed by means of an inlet header, and the flow from the tubes is collected in an outlet header. The tubes are located side-by-side. To provide the space required to weld the tubes to the header, the tube-to-header connections are spaced around the circumference of the header, as shown in Figure 7-12. The portions of the tubes between the end of straight section and the connection to the header are known as jumper tubes.

The header is often fabricated from a Sch 20 pipe. The wall thickness of a Sch 20 pipe is at least an order of magnitude greater than the wall thickness of the tube. During a cloud transient, the temperature of the salt in the receiver can decrease at rates as high as 6 °C/sec. This rate of change is, in turn, imposed on the tube-to-header connection. To maintain the transient stresses in the connection at levels consistent with a fatigue life of 30 years, a tapered nozzle between the header and the tube is used. The tapered connection can take one of several forms:

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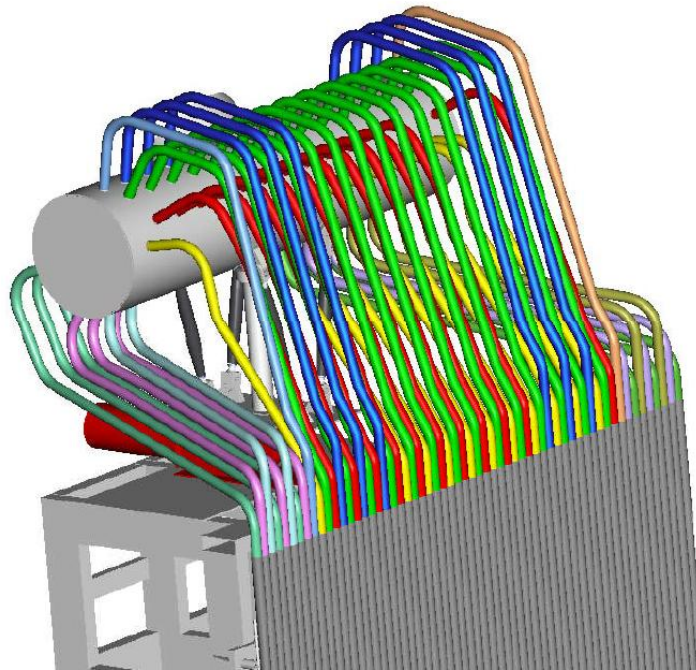


Figure 7-12 Tube-to-Header Connections

1. A hole is drilled in the header, and a machined nozzle is welded to the header.
2. In the lower header, a thermal sleeve can be used to trap a small volume of salt between the tube and the header.
3. A hole is drilled in the header. A collet is inserted in the hole, and a tapered nozzle is plastically pulled from the header.

7.1.11 Oven Boxes

Panel inlet and outlet headers, and the jumper sections between the headers and the tubes, are enclosed by upper and lower oven assemblies. The ovens are insulated, and contain radiant heaters for preheating the headers and jumper tubes. The ovens are removable to allow access to the headers for heater maintenance and tube replacement. Figure 7-13 is a conceptual view of the oven assembly.

The capacities of the radiant heaters are selected to preheat the header assemblies to a temperature of 300 °C in 2 hours, while exposed to the design wind speed and the design ambient temperature.

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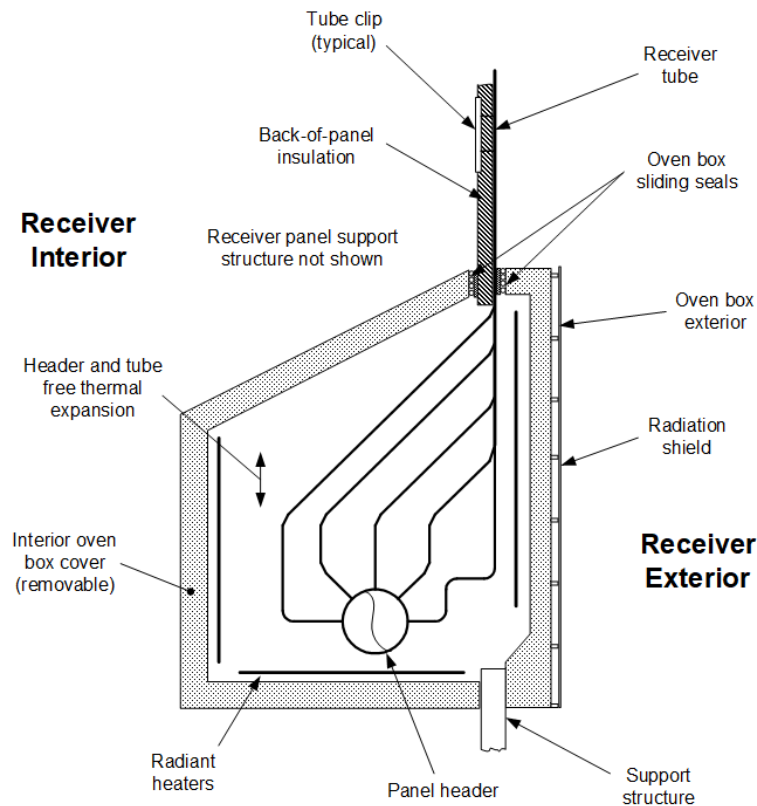


Figure 7-13 Cross Section of Receiver Lower Oven

The configuration of the oven box is such that it is difficult to completely seal the region where the tubes exit from, and enter, the oven. There are also gaps between tubes themselves. These openings can result in high convection heat losses, especially on windy days. To reduce the convection air flows, the tube entrance and exit regions will use either compressible seals, or spring loaded sealing surfaces. The region behind the tubes will be sealed with insulation.

Radiant electric heating elements are located on the oven box wall surrounding the header assembly. Depending on the effectiveness of the oven box-to-tube seals, additional electric heat tracing may also be required in the entrance / exit regions.

The oven boxes on the receiver at the Solar Two project located the inlet and outlet headers, and the associated jumper tube sections, outside of the plane of the absorber. However, it was very difficult to seal the gaps between the ovens in adjacent panels. On windy days, air entered in the gaps between the ovens on the windward side, flowed through consecutive ovens around the receiver, and exited on the leeward side. To avoid similar problems, the oven boxes in Figure 7-13 are located inside the plane of the absorber. The external surfaces of adjacent ovens then form continuous surfaces above and below the absorber, and preventing air infiltration is less complex undertaking.

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An exterior radiation shield is required to protect the oven boxes from spillage above and below the absorber. The shield requirements are as follows:

- Incident fluxes of 200 to 250 kW/m²
- Sections replaceable in a single shift of 8 hours
- Fabrication material of Type 304 or 316 stainless steel sheet, 1.6 to 2 mm thick, coated with Pyromark 1200 series (white) paint
- Coverage of 360° around the receiver.

7.1.12 Inlet Vessel Design

The inlet vessel performs the following functions:

- Stores a quantity of salt, which is adequate to supply the receiver for 60 seconds following a failure of the cold salt pump or site power. The potential energy for supplying the salt to the receiver is compressed air above the salt liquid level
- Provides a free surface for controlling the speed of the receiver pumps. Specifically, 1) the flow rate through the receiver is controlled by the two circuit control valves to satisfy the set point temperature of 565 °C at the outlet of each circuit, and 2) the speed of the receiver pumps are adjusted to maintain the level in the inlet vessel at the mid-height of the vessel.

The dimensions of the vessel are selected such that the salt volume and the ullage volume are large enough that the decay in ullage pressure does not cause the salt flow rate to drop below that required to protect the receiver. The inlet vessel is typically located inside of the primary support columns for the receiver. Since the distance between the columns is defined by the absorber dimensions, and since the absorber dimensions are defined by the optimization studies (i.e., as small as possible), the height-to-diameter ratio for the vessel is often a large value. A sketch of the inlet vessel at a representative commercial project is shown in Figure 7-14.

The vessel is designed and fabricated to the requirements of Section VIII Division 1 of the ASME Pressure Vessel Code, and is code stamped. The design pressure will be the shutoff head of the receiver pump, plus a nominal allowance of 10 percent, minus the static head between the pump and the vessel.

Since compressed air must be supplied to the inlet vessel, there is the possibility of air leaking from the vessel during normal operation, and reducing the mass of the ullage gas below the required values. As such, level instruments will be required to monitor the liquid level. At full flow conditions, the pressure

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loss through the receiver is a nominal 20 bar. Since instrument air is normally supplied at a pressure of 10 bar, a supplemental source of compressed air will be required to charge the vessel for normal operation and to compensate for air leakage during operation. The high pressure air is supplied by a supplemental compressor, which draws suction from the instrument air header.



Figure 7-14 Sketch of a Representative Inlet Vessel

7.1.13 Downcomer Design

Head Dissipation

A downcomer line connects the top of the receiver with the hot storage tank. In commercial projects, the elevation of the receiver is on the order of 200 to 230 m. This represents a static head of some

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35 bar, which must be dissipated before the salt enters the storage tank. There are two approaches for head dissipation:

- Throttle valves located at the base of the downcomer. The downcomer always operated in a flooded state. The control signal for the throttle valves is provided by an outlet vessel located at the top of the receiver.
- Orifice plates located along the length of the downcomer. The plate is a circular disc, perhaps 25 mm thick, through which is drilled a number of holes. The plates are spaced, for example, 10 m apart, and the pressure drop through each plate, at the design flow, is on the order of 9 m. The flow in the downcomer consists of flooded region above a plate, a void space immediately below this plate, followed by another flooded region above the next plate in the series. The height of the flooded region varies with the flow rate in the downcomer.

The principal advantage of the second approach is a passive approach to head dissipation; i.e., active throttle valves are not required. The design also avoids the need for an outlet vessel. Deleting the outlet vessel has a noticeable effect on the cost of the receiver system. The vessel, fabricated from stainless steel, must have a volume similar to that of the inlet vessel. Also, the vessel, when flooded, has a significant mass, which influences the weight and the moments carried by the tower and the foundation.

The principal liability of the second approach is the difficulty of predicting the flow patterns, and the corresponding momentum loads on the elbows in the expansion loops of the downcomer.

Outlet Vessel and Vent Line

To date, all but one commercial project has elected to use a flooded downcomer with an outlet vessel. Nonetheless, it can be noted that the selection of an outlet vessel is not a risk-free decision. The outlet vessels serves two purposes:

1. Provides a level control signal for the downcomer throttle valves
2. Collects the flow from the inlet vessel during an emergency in which the heliostat field must be defocused following a problem with the downcomer valves.

However, the capacity of the outlet vessel is not infinite. If a condition arises in which the level in the outlet vessel reaches the high-high value, but the heliostat field is not yet defocused, the operator must make a decision: 1) trip the receiver pumps to prevent the outlet vessel from overflowing, but risk damage to the receiver; or 2) protect the receiver by maintaining flow in the panels, but allow the outlet vessel to overflow. Either decision will lead to receiver damage, with repairs periods lasting days to months. To avoid this problem, a vent line can be used to connect the top of the outlet vessel with the top of the hot salt tank. The vent line must operate in the empty condition to satisfy the Code

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requirements regarding the operating pressure of the outlet vessel. However, should the outlet vessel overflow, salt will enter the vent line, and then fall by gravity to the hot tank. As one might imagine, the momentum loads imposed on the elbows at the expansion loops will be very high. Pipe guides and anchors with the required strength can be designed and fabricated. Nonetheless, the uncertainties regarding the design load criteria must be considered high, and some degree of conservatism in the design is warranted.

Actuators for the Downcomer Valves

A common approach to the selection of the throttle valves is a conventional globe valve, with a flow over arrangement.

During morning startup, the downcomer, the panels, and the outlet vessel are typically flood filled from below. During this phase, the throttle valves are set to an open position of approximately 10 percent. Once the normal level in the outlet vessel is established, the static head operating on the top of the valve plug can be as much as 250 m. To prevent the valve plug from moving down, and then oscillating between two different open positions, the actuator must be able to develop forces on the stem that can unambiguously maintain the valve position. Pneumatic actuators can perform this function. However, hydraulic actuators generally provide more control authority and are preferred by some engineering contractors.

7.1.14 Efficiency

The absorber tubes are painted with a selective surface coating, such as Pyromark. When new, the coating has a short wave length absorptivity of 96 percent. As such, for every 100 photons incident on the absorber, 4 photons are reflected back to the environment, and the balance of 96 are converted to thermal energy on the front of the receiver tube.

The selective surface has a long wave length emissivity of 0.80 to 0.85, depending on the surface temperature. The radiation losses from the absorber are proportional to the local emissivity, the distribution of the surface temperature on the front circumference of the tubes, and the view factor of the tube to the environment, including the view factors to the adjacent tubes.

The convection losses from the absorber are a combination of natural convection and forced convection. The net effect of the two convection coefficients are often combined using the following expression:

$$H_{Mixed} = \sqrt[3.2]{H_{Natural}^{3.2} + H_{Forced}^{3.2}}$$

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The convection losses are typically smaller than the radiation losses. Nonetheless, determining the convection coefficients is difficult, both experimentally and with CFD models, and the uncertainties in the mixed coefficient are judged to be high.

Lastly, there are also limited thermal losses to the support structure due to conduction heat transfer.

A plot of the thermal efficiency of the receiver, as a function of the input duty and the wind speed, is shown in Figure 7-15. The peak value is on the order of 90 percent.

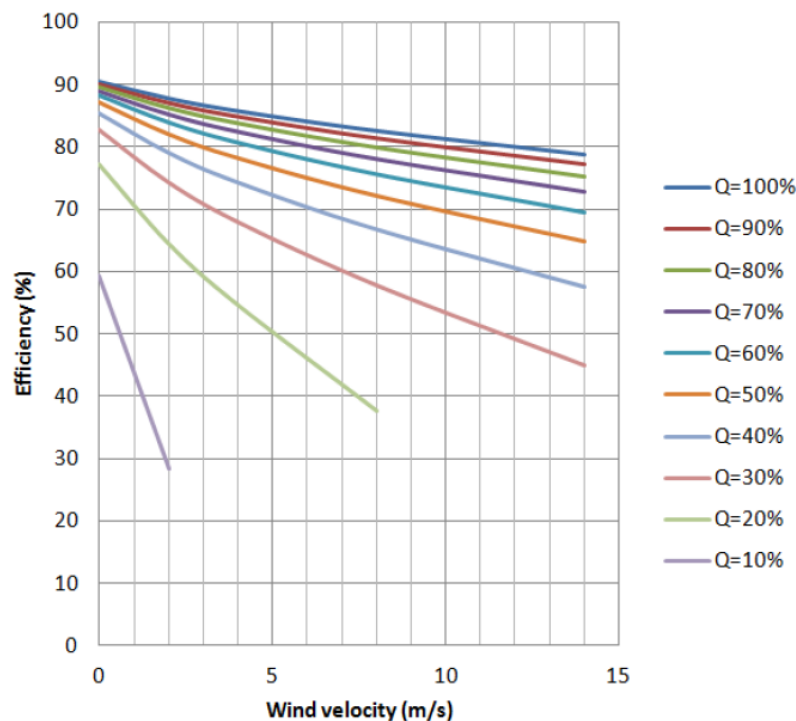


Figure 7-15 Receiver Efficiency as a Function of Input Duty and Wind Speed

7.1.15 Daily Operation Sequence

Fill

To prevent the salt from freezing in the panels during overnight shutdowns, the receiver is drained at the end of each operating day. This requires, in turn, that the receiver be filled each at the start of each operating day. The fill sequence involves the following steps:

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- Approximately 1 hour prior to sunrise, the radiant heaters on the panels headers are activated. The goal is to establish nominal metal temperatures of 300 to 350 °C at the start of the fill process.
- When the elevation of the sun reaches a value of perhaps 6 degrees, a select group of preheat heliostats is instructed to track the absorber, and to track the oven boxes above and below the absorber. The preheat flux falls in the range of 30 to 60 kW/m², depending on the wind speed and the wind direction. The goal is to establish nominal tube temperatures, measured at the back of the tubes, on the order of 350 °C.
- During the panel preheating process, the riser and the downcomer are flood filled from below. The fill process occurs up to the elevation of an upper riser-to-downcomer bypass line, which is located below the bottom of the receiver panels.
- When the panel headers and the absorber tubes have reached a safe fill temperature, the panels, the upper portion of the downcomer, and the outlet vessel are flood filled from below by increasing the speed, and therefore the head, of the cold salt pumps.
- When the panels start to fill, salt also starts to fill the inlet vessel. The vent on the inlet vessel is closed at this point in the process, and as the salt level in the inlet vessel increases, the pressure of the ullage gas in the inlet vessel also increases.
- When the outlet vessel reaches its normal operating level, the panel vent valves are closed and the panel drain valves are closed. This changes the flow path in the receiver from 1) parallel flow up in each of the panels to 2) series flow in the panels in each of the two flow circuits.
- The ullage gas pressure in the inlet vessel is increased to provide the potential energy needed for emergency cooling of the receiver.

Normal Operation

During normal operation, the positions of the control valves at the inlet to each of the two flow circuits are adjusted such that the circuit outlet temperature is equal to the set point temperature. With clear sky conditions, the set point is the design value of 565 °C. During cloudy weather, the set point value is typically reduced, as discussed below. The speed of the cold salt pumps are adjusted to maintain a level set point in the inlet vessel.

The temperature at the outlet of each circuit provides a feedback signal to the circuit flow control valves. Due to the long heating path in the receiver, the residence time of the salt in the receiver ranges from 1 minute at the design flow rate to perhaps 4 to 6 minutes at the minimum flow rate. If the control logic

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were to rely only on the circuit outlet temperature for flow control, the panels would be subjected to overheating during cloudy conditions in which the radiation on the solar collectors was rapidly changing. To reduce the potential for overheating, a feed forward signal to the circuit control valves can be provided by flux instruments measuring the incident power distribution on the absorber.

Instruments which measure the flux in the plane of the absorber are commercially available. However, due to the harsh operating conditions, the instruments must be actively cooled, and if there are problems with the cooling system, the instruments will soon fail. As such, most commercial projects rely on instruments which measure the short wave length radiation reflected from the absorber. The instruments can be located below the absorber, supported from the tower on extended platforms, or they can be located in the collector field. The former offers the advantages of high resolution, but must be able to survive high temperatures associated with spillage flux below the absorber.

The accuracies of flux instruments, both direct and indirect, are generally good but not outstanding. To prevent the receiver from overheating during cloudy weather, the set points for the circuit outlet temperatures are normally reduced to compensate for potential errors in the flux measurements. A typical set point value might be 550 °C, which allows the outlet temperature to overshoot by as much as 35 to 40 °C without challenging the high temperature trip point for the receiver.

Naturally, reducing the set point temperature below 565 °C results in a reduction of the temperature in the hot salt tank. This has two measurable effects:

- The thermal capacity of the storage system, in MWh, is reduced
- The steam generator is supplied with salt from the hot salt pumps. The pumps have a defined flow and head capacity. If the hot salt temperature falls below some threshold value, the hot salt pumps can no longer supply the design thermal duty to the steam generator. This results in reductions in the live steam temperature, the reheat steam temperature, the live steam flow rate, and the gross Rankine cycle efficiency.

Turndown Ratio and Clear Sky Flow Rate

The flow circuits are typically arranged in a sequence with the flow moving up in one panel, moving down in the next panel, up in the next, and so on.

The highest flux levels occur at the circuit inlet, and the lowest fluxes occur at the circuit outlet. As such, there is a lateral flux gradient present in each panel. Further, in the panels which flow down, buoyant forces are in opposition to pressure losses. At low flow rates, the pressure drop in each panel is a fraction of 1 bar. At some flow turndown ratio, a condition can occur in which the lateral flux gradient in combination with the buoyant forces cause the flow to either stall or reverse in one or more tubes.

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Should this happen, these tubes will quickly overheat, and likely rupture due to metal temperatures exceeding the melting point of the alloy.

To ensure that this condition never occurs, a maximum flow turndown ratio for the receiver is defined. A typical value is 6:1, but some commercial plants have selected conservative values as low as of 4:1.

When the theoretical turndown ratio falls below, for example, 5:1, the control logic changes. Outlet temperature control is abandoned, and the flow is set to a value corresponding to clear sky conditions. Under this scenario, the outlet temperature would be the design value of 565 °C if the skies were completely clear at that time of the day and on that day of the year. As such, the receiver panels cannot overheat as the flow rate is already set to a value which is high enough to ensure adequate cooling.

The consequence of switching from a flow rate of 20 percent to a flow rate of essentially 100 percent, with no change in the incident power on the absorber, is an immediate decrease in the circuit outlet temperature. This transient, which can produce rates of temperature change as high as 6 °C/sec, is imposed on the outlet vessel, the downcomer, and the throttle valves at the base of the downcomer. This equipment must be able to tolerate at least 10,000 cycles of these transient conditions.

Drain

At the end of an operating day, the receiver is shut down through the following process:

- Outlet temperature control is maintained until the maximum turndown ratio is reached
- The majority of the heliostat field is defocused, except for a select group of heliostats which continue to track the absorber to provide a post heat flux
- The majority of the ullage pressure is released from the inlet vessel. This results in a decay in the flow rate through the receiver
- The panel vent valves and the panel drain valves are then fully opened. This changes the flow path from a series arrangement to parallel flow down in each panel. Simultaneously, 1) the downcomer valves are opened to reduce the level in the downcomer to the elevation of the upper riser-to-downcomer line, and 2) the inlet vessel is vented to the atmosphere
- The riser and the downcomer are drained simultaneously
- The post heat flux is maintained on the receiver for as long as practical to reduce the amount of residual salt remaining in the panels.

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7.1.16 Control System

The basic control objective is to maintain tube strains within acceptable limits, while simultaneously regulating salt flow to provide an outlet temperature of 565 °C. A representative approach to receiver control includes the following elements:

- A feed-forward signal from two groups of photometers sets the demand signal for the control valve in each circuit. The photometers, mounted just below the receiver at the top of the beam characterization system targets, view a ‘representative’ reflected flux from each of the two flow circuits. The photometers consist of a filter, a collimator, and a photodiode. The incident flux on the photodiode produces a voltage proportional to the reflected flux from the receiver
- Two proportional-integral feedback temperature signals from thermocouples in the outlet of each flow circuit trim the flow rate signal
- Flow from the inlet vessel to the East circuit, and separately to the West circuit, are controlled by a valve at the inlet to each circuit. Flow to the inlet vessel is modulated by adjusting speed of the cold salt pumps to maintain a set point level in the inlet vessel
- Panel high temperature protection is provided by defocusing a nominal 30 percent of those heliostats focused on the panel in response to a high temperature alarm. 60 percent will be defocused on a high - high alarm, and all of the heliostats on a high - high - high alarm. Temperature signals are provided by thermocouples welded to the backs of the tubes, and by thermocouples located in the crossover lines between adjacent panels.

Four permanent infrared cameras, capable of viewing the entire receiver surface, are used by the plant operators to support the daily preheat, fill, and drain procedures. Cameras are installed in the heliostat field, preferably co-located with the beam characterization system cameras. The cameras, which will not be part of the control system, provide qualitative data to check for cold or hot spots on the absorber surface.

7.2 Salt Pumps

7.2.1 Configuration

To date, no mechanical seal facing material has been found to be compatible with nitrate salt. As such, an alternate approach has been adopted, in which gravity is used as the pump shaft seal. Specifically, all commercial salt pumps are a vertical turbine designs, with the shaft bearing lubricated by salt. Near the top of the pump shaft is a throttle bushing. Salt flow past the bushing is returned by gravity to the tank

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which supports the pump. A packing gland is located at the top of the shaft to help prevent salt from wicking from the top of the pump. The gland is cooled by a small flow of instrument air.

Pumps drives are exclusively electric motors, and normally use variable frequency drives to help control auxiliary power demands.

Pumps stages range from 2 for the pumps supplying the steam generator, to 5 for the pumps supplying the receiver. Pumps are available with flow capacities of at least 1,000 m³/hr, total developed heads of at least 280 m, and motor drive powers greater than 2.5 MWe.

7.2.2 Structural Support

To collect the salt flow from the throttle bushing, the pump requires some form of a sump. At the Solar Two project, a cold salt sump was provided for the receiver pumps and the steam generator attenuation pump, and a hot salt sump was provided for the steam generator hot pump. The two sumps were located below grade, and supplied with salt from the thermal storage tanks by means of salt lines and level control valves. The advantage of this arrangement was ability to use salt pumps of a relatively short length (~ 2 m), which simplified the harmonic analyses of the pump columns. The principal disadvantage was the reliance on a control system to maintain a set point value for the liquid levels in the sumps. Specifically, a failure in the level control system could cause the sumps to flood, and in a worst case scenario, the entire inventory in the storage tank could be spilled on the ground.

To avoid this potentially catastrophic failure condition, all commercial projects following the Solar Two project have used the thermal storage tanks as the pumps sumps. The roofs of the storage tanks do not have the necessary structural stiffness to support the pumps. Instead, a dedicated support structure is provided, which supports the pumps near the perimeter of the tank. A thermal storage tank in a commercial project has a representative height of 12 m. As such, the length of the salt pump must be on the order of 15 to 16 m to accommodate the sum of the height of the tank, the thickness of the insulation on the roof of the tank, and the vertical dimensions of the steel sections in the support structure. A photograph of a pump in a representative commercial project is shown in Figure 7-16.

A pump column with this length has a low bending stiffness; however, harmonic analyses of the columns have shown that the moments and thrust forces are acceptable. To date, at least 150 pumps with this type of extended shaft have been installed in commercial projects, and the reliability has been excellent.

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Figure 7-16 Salt Pump in a Commercial Project

7.2.3 Materials

The cold salt pumps are fabricated primarily from carbon steel, but a range of materials are used for the impellers, depending on the supplier. These materials include chrome-moly steel and martensitic stainless steel. A range of materials is also used for the shaft bearings, including cast iron, Type 420 stainless steel, and Stellite.

The hot salt pumps are fabricated primarily from a stabilized stainless steel; i.e., Type 347H. The impellers are typically a cast stabilized stainless steel, and the shaft bearings can be cast iron or Stellite.

7.2.4 Efficiency

Rated efficiencies range from 72 to 81 percent, with the efficiency, in general, inversely related to the drive power.

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7.2.5 Temperature Rate Limits

In a commercial project, the length of the hot salt pump, between ambient temperature and the design temperature, changes by distances in the range of 150 to 200 mm. The distance is much smaller than the vertical clearance between the impeller and the bowl. As such, the radial and longitudinal temperature gradients in the pump must be closely controlled to prevent the impeller from contacting the bowl and potentially binding the pump. In general, the allowable rate of temperature change for the pump is similar to the allowable rate of temperature change for the tank; i.e., 1 °C/min.

The temperature rate constraint also has implications for the design of the distribution header in the tank. Specifically, the salt pumps draw suction from a location just above the floor. During receiver startup, salt from the downcomer to the hot tank at temperatures as much as 50 °C below the bulk inventory temperature. When the relatively cold salt leaves the distribution ring, it can descend to the floor due to buoyancy effects. The density gradient will promote stratification, and it is possible for the relatively cold salt to move across the floor, and enter the suction of the hot pump. This, in turn, has the potential to thermally shock the pump, leading to interference, vibration, and wear.

On a related point, there are limits on the rate at which the pump can be withdrawn from the tank for maintenance purposes. This is to ensure that the temperature at the centerline of the pump does not become out of phase with the temperature on the outside surface of the column. Should the two temperatures become out of phase, differential thermal expansion between the outside of the column and the shaft has the potential to permanently damage the shaft, the impellers, or the bowls. Representative removal rates for a commercial pump are as follows:

- Raise 3 m, and hold for 15 minutes
- Raise an additional 3 m, and hold for 15 minutes
- Raise an additional 3 m, and hold for 15 minutes
- Raise an additional 2 m, and hold for 15 minutes
- Raise above the liquid level, and hold for 30 minutes or until no visible salt is draining from the pump.

A similar schedule is required to insert the pump into the tank, as follows:

- Lower 1 m into the salt inventory, and hold for 8 hours
- Lower an additional 3 m into the salt inventory, and hold for 2 hours

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- Lower an additional 3 m into the salt inventory, and for hold 2 hours
- Lower additional 3 m into the salt inventory, and hold for 2 hours
- Lower to the mounting flange.

7.3 *Thermal Storage System*

The thermal storage system consists of the hot tank, the cold tank, the foundations for each tank, the salt inventory, and inlet distribution piping in each tank.

7.3.1 **Representative Commercial Design**

Tank

A representative storage quantity for a commercial project is 30,000 to 40,000 metric tons of salt. Since the salt has an extremely low vapor pressure (< 1 kPa at $600\text{ }^{\circ}\text{C}$), the lowest cost storage approach is a vertical tank with a flat bottom and a self-supporting roof. Due to the chemical stability of the salt, the tank can be vented directly to the atmosphere.

Design standards for this type of tank, operating at temperatures up to $600\text{ }^{\circ}\text{C}$, have yet to be developed. In the interim, the closest design standard is American Petroleum Institute Standard 650, Welded Tanks for Oil Storage. However, the Standard only considers operating temperatures up to $260\text{ }^{\circ}\text{C}$, and low cycle fatigue conditions are not part of the Standard. As a result, a hybrid design approach has been adopted, as follows:

- For design temperatures above $260\text{ }^{\circ}\text{C}$, allowable material stresses are derived from Section II of the ASME Code
- For the evaluation of low cycle fatigue, the methods described in Section VIII, Division 2, of the Code are often adopted.

Materials

Hot Tank For design temperatures above $538\text{ }^{\circ}\text{C}$, the H grades of stainless steel, with a minimum carbon content of 0.04 percent, are required by the ASME Code.

Candidate materials include Type 304H, Type 316H, and Type 347H stainless steel. At the Solar Two demonstration project, the hot tank was fabricated from Type 304H. However, to avoid potential problems with intergranular stress corrosion cracking, the solar industry has adopted Type 347H, a

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stabilized stainless steel, in the current generation of projects. Nonetheless, Type 347H is not a problem-free material selection, and the relative merits of stabilized and non-stabilized stainless steels is discussed in Section 5.12 of Volume 3 - Narrative.

Cold Tank During receiver startup, the temperature of the salt in the downcomer increases from an initial value of 295 °C to a final value of 565 °C. To prevent a decay in the inventory temperature of the hot tank, salt at temperatures below a crossover set point is diverted to the cold tank. The crossover temperature is a function of the inventory level and the inventory temperature in the hot tank, with representative values in the range of 470 to 510 °C.

On a typical day, the inventory level in the cold tank is high during morning startup of the receiver. As such, the cold tank can accept relatively large quantities of salt, at temperatures as high as 510 °C, before the bulk inventory temperature in the cold tank rises as noticeable amount. To provide as much flexibility in starting the receiver, particularly during cloudy weather when it may be difficult to promptly reach a design receiver outlet temperature of 565 °C, a design temperature of 370 °C for the cold tank is often selected. This design value allows the use of carbon steel for the cold tank, and allows the highest allowable stresses in Section II of the Code to be used for the stress analyses.

Tank Dimensions

The storage capacity, in MWht, is proportional to the gross Rankine cycle power, the gross Rankine cycle efficiency, and the number of hours of full-load turbine operation from storage. The active storage mass is proportional to the storage capacity, in MWht, and the difference in between the enthalpy of the salt at the hot tank design temperature and the enthalpy of the salt at the cold tank design temperature. Once the active storage mass is defined, the active storage volume in the cold tank is inversely proportional to the density of the salt at the cold storage temperature. Similarly, the active storage volume in the hot tank is inversely proportional to the density of the salt at the hot storage temperature.

In addition to the active storage inventory, there is an inactive storage inventory. The latter consists of the following:

- The clearance between the floor and the suction bell of the salt pumps
- The minimum submergence of the bottom of the pump column.

The combination of the clearance and the submergence ensure that only salt (i.e., not a combination of salt and air) enters the pump suction. Representative distances for the pumps in a commercial project are shown in Figure 7-17. The distances are in mm.

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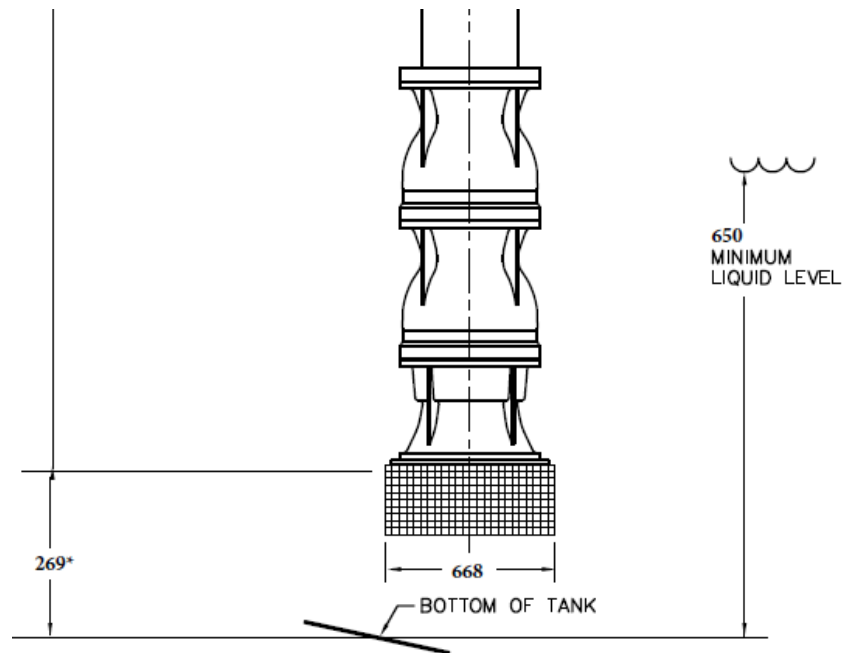


Figure 7-17 Salt Pump Clearance and Submergence

Additional considerations in terms of tank dimensions include the following:

- It is desirable that each tank can hold the entire salt inventory of the project. The inventory includes the inactive inventory of the other tank, plus the inventories in the receiver system and the steam generation system.
- During an earthquake, sloshing can occur in the tanks. The welded connection between the wall and the roof is not particularly strong, and should an internal wave in the tank reach this area, damage to the connection can occur. With the entire project inventory in a tank, a nominal freeboard allowance of 0.3 m is provided at the top of the wall to limit hydraulic loads on the wall-to-roof connection.

As discussed in the following section, the tank foundation is supported on a concrete base mat. As such, the allowable vertical load on the base mat is nominally equal to the allowable soil bearing load. For a representative commercial site, the allowable soil bearing load is 240 kPa (5,000 lb_f/ft²). This, in turn, limits the maximum allowable salt depth in a tank to nominal 12.5 m. The maximum salt depth includes the active inventory, the inactive inventory, the inventory of the receiver and the steam generation systems, and the inactive inventory from the other tank. This, in turn, translates to a net active inventory depth of about 10.5 to 11.0 m. Once the active inventory depth is defined, the diameter of the tank is then a function of the required storage volume.

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To a first order, the thicknesses of the various wall sections can be calculated from the hoop stress equation; i.e., Thickness, $m = (\text{Static height, m}) * (\text{Density, kg/m}^3) * (\text{Acceleration due to gravity, m/sec}^2) / (2 * \text{Allowable stress, kg/m-sec}^2)$. To this thickness is added a corrosion allowance. For the tanks in commercial projects, the thickness of the bottom course can range from 35 to 45 mm.

The cold tank is fabricated from carbon steel. If the plate thickness exceeds 38 mm (1.5 in.), then Section VIII of the Code requires that a post weld heat treatment be performed. However, post weld heat treatments are not without risks, as follows:

- The temperatures are high enough (600 to 650 °C) to deform the wall plates
- The temperatures are high enough for the iron in the steel to reduce the carbon dioxide in the atmosphere. This, in effect, carburizes the surfaces of the plates, which can lead to an increase in the strength, but a reduction in ductility, of the material.

If the engineering contractor is adverse to the use of a post weld heat treatment, then one solution is to reduce the wall thickness by using two 50-percent capacity tanks.

Foundation

The foundation consists of the following elements, moving from the top down:

- A layer of solid lubricant, the most common of which is sand
- The primary tank support and insulating layer under the floor. The most common material is an expanded clay.
- A concrete base mat below the expanded clay
- A concrete ring wall to retain the expanded clay underneath the tank.

In some commercial projects, the only material directly supporting the tank floor is expanded clay. The clay has a compressive strength adequate to support the concentrated loads at the tank perimeter from the weight of the wall and the roof. Nonetheless, the clay particles can shift due to the daily expansion and contraction of the tank. This leaves the annular plate at the perimeter of the floor only partially supported, which, in turn, produces bending loads in the annular plate that are high enough to plastically deform the plate. The wall-to-floor welds in this region are very thick (50 to 60 mm), and subjecting the welds to plastic deformations compromises the low cycle fatigue life of the tank and can accelerate stress relaxation cracking.

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To prevent this problem, a number of commercial projects use a rigid material, such as refractory bricks or cast refractory, at the foundation perimeter to ensure that the annular plate remains in the same plane as the floor. A cross section view of a representative tank foundation is shown in Figure 7-18.

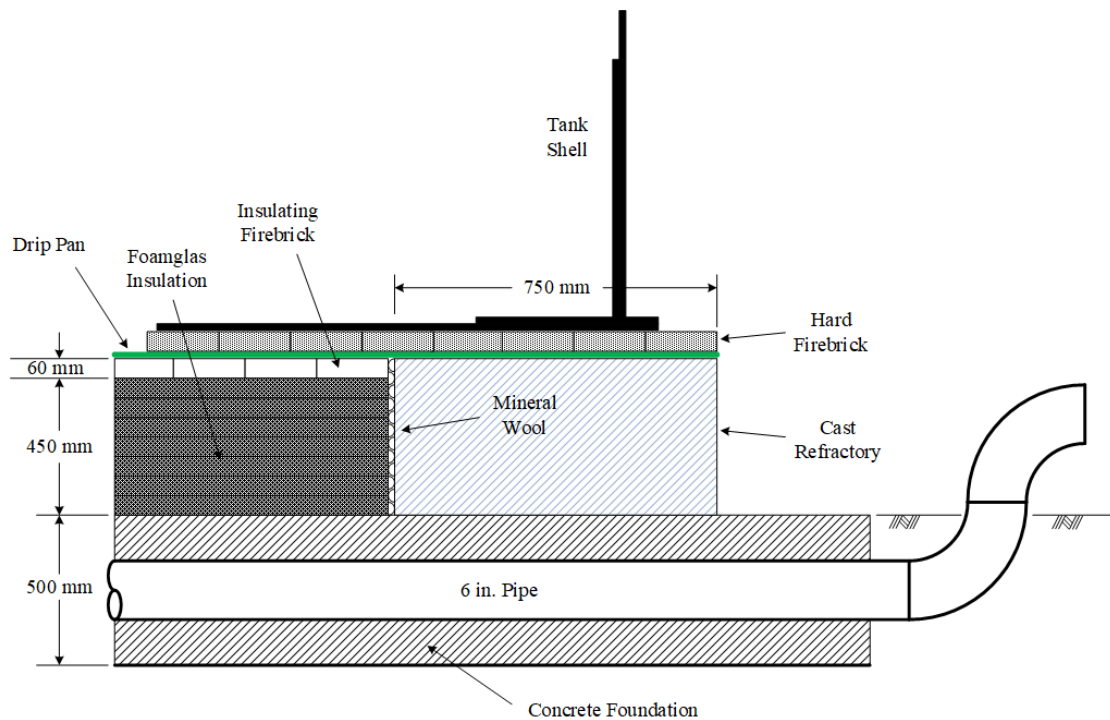


Figure 7-18 Representative Cross Section of Tank Foundation

Foundation Cooling

The foundation base mat has a series of cooling air pipe embedded in the concrete. A plan view of the cooling passages in a representative commercial project is shown in Figure 7-19.

The cooling air is forced through the passages by a group of fans. The flows in adjacent pipes move in opposite directions to provide a more uniform heat flux through the foundation.

The air flow rate is set to maintain the soil temperature, directly beneath the foundation base mat, at values of 75 °C or lower. The goal is to limit the desiccation of the soil, and the oxidation of organic materials, to values that prevent unpredictable settlements of the tank.

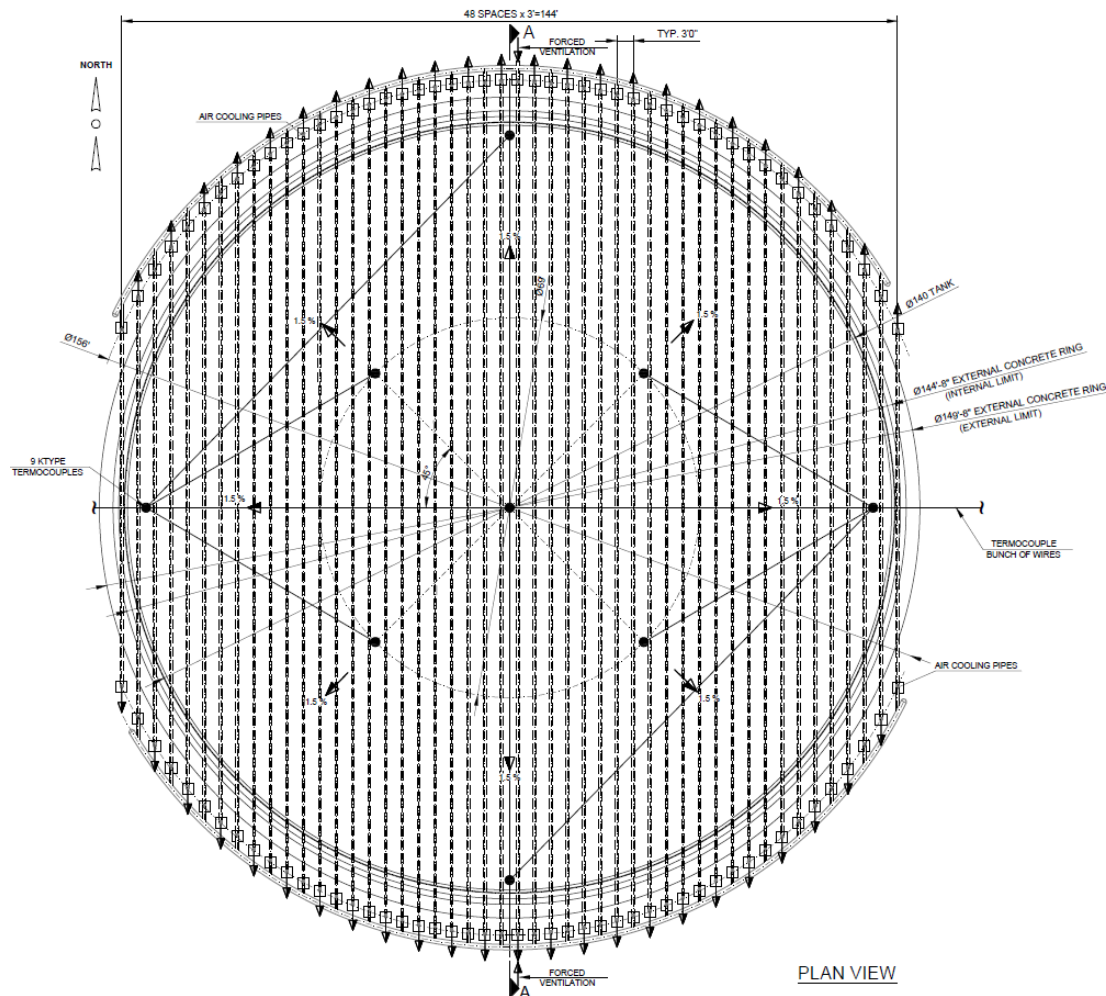


Figure 7-19 Foundation Cooling Air Passage Layout

Inlet Distribution Piping

The flow is introduced in the tank by means of a distribution ring header. The circumference of the ring is generally in the range of 40 to 60 percent of the circumference of the tank. The ring is located at a distance from the floor (0.5 to 1.0 m) that is within the depth of the minimum permanent inventory of the tank. The header has a series of either holes or eductors to promote mixing between the incoming flow and the bulk inventory.

It should be noted that a distribution header is not a perfect mixing device, and non-isotropic temperature distributions in the inventory during transient conditions are inevitable. This topic is additional detail in Section 5.9 of Volume 3 - Narrative.

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7.3.2 Tank Preheating

The tanks must be preheated to a nominal temperature of 300 °C prior to filling with salt. This is to prevent a thermal shock of the tank material, which would otherwise result in transient stresses exceeding the yield stress.

The most common approach to preheating the tank is to burn either propane or natural gas in an air heater, and then circulate the combustion gas through the tank. The combustion gas is introduced through a large nozzle located just inside the roof. The gases leave the tank through a manway in the roof opposite the nozzle. The temperature of the gas entering the tank is controlled to match the allowable rate of temperature change for the tank, and to control intra-tank differential temperatures to predefined values. Close control over the gas and the metal temperatures is needed to limit intra-tank thermal stresses to acceptable values.

As might be expected, the thermal inertia of the foundation is at least an order of magnitude greater than the thermal inertia of the tank metal and external insulation. The preheating process must occur slowly enough to allow heat to be transferred into the foundation by conduction. As such, the time required to preheat a tank is on the order of 10 days.

The combustion gas from the air heater contains a mixture of N₂, O₂, H₂O, CO₂, and NO_x. The H₂O and the CO₂ will react to form carbonic acid, H₂CO₃, and at temperatures below perhaps 80 to 85 °C, the acid vapors will condense on the inside surface of the tank. The acid presents no corrosion risk to the stainless steel in the hot tank, but it is extremely corrosive to the carbon steel in the cold tank. In commercial projects, corrosion allowances as high as 3 mm have been specified for the cold tank to accommodate potential losses in section thicknesses. To reduce the risk of potential damage to the cold tank, the preheating phase must move as quickly as possible through the temperature range in which carbonic acid will condense. Further, the preheating process must not be allowed to stall if the metal temperatures are lower than 100 °C.

7.3.3 Temperature Limits

Allowable Rate of Temperature Change

The tanks are large structures, with maximum section thicknesses on the order of 35 to 45 mm. In addition, there are sharp changes in section thicknesses; for example, where the bottom course of the wall (40 mm) meets the annular plate at the perimeter of the floor (15 mm). As such, the allowable rate of temperature change for the tank is typically limited to 1 °C/min to prevent the development of large, and potentially damaging, transient thermal stresses.

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Intra-tank Temperature Differences

Similarly, the tank will develop thermal stresses if there are temperature gradients in the tank. A common source of transient thermal gradients in both the cold tank and the hot tank is the daily startup and shutdown of the receiver.

For the tanks in a commercial project, representative allowable temperature differentials include the following:

- 15 °C between any point on the wall and any point on the floor (or the roof)
- 25 °C between any two points on the wall.

It should be noted that these temperature values are only representative. A finite element stress analysis, to be conducted by either the tank vendor or the engineering contractor, will define the allowable limits for the project.

Radial Temperature Gradients in the Floor

It is also possible to develop a combination of radial and tangential stresses in the floor due to steady state radial temperature gradients in the floor. The radial temperature gradients are produced due to differences in the conduction heat transfer distances. For example, heat transfer to the foundation near the center of the tank is predominantly straight down, and the thermal resistance through the foundation insulation between of the tank and the top of the cooling air passages in the concrete base mat is 'high'. In contrast, the heat transfer from the foundation near the perimeter of the tank is a combination of vertical (down) and radial (horizontal). The effective conduction distance to the environment is less than the effective distance near the center of the tank. As a consequence, the heat flux into the foundation at the tank perimeter is approximately double the heat flux into the foundation at the center. The radial distribution in heat flux translates to a radial distribution in the temperature of the floor.

For an unconstrained circular plate, subject to a radial temperature gradient, the radial stress and the tangential stress are given by the following formula in Roark ⁶:

⁶ Young, W. C., and Budynas, R. G., 'Roark's Formulas for Stress and Strain', Seventh Addition, 2002

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12. If the disk of case 10 is heated symmetrically about its center and uniformly throughout its thickness so that the temperature is a function of the distance r from the center only, the radial and tangential stresses at any point a distance r_1 from the center are

$$\sigma_{r_1} = \gamma E \left(\frac{1}{R^2} \int_0^R T r \, dr - \frac{1}{r_1^2} \int_0^{r_1} T r \, dr \right)$$

$$\sigma_{t_1} = \gamma E \left(-T + \frac{1}{R^2} \int_0^R T r \, dr + \frac{1}{r_1^2} \int_0^{r_1} T r \, dr \right)$$

where R is the radius of the disk and T is the temperature at any point a distance r from the center minus the temperature of the coldest part of the disk. [In the preceding expressions, the negative sign denotes compressive stress (Ref. 7).]

The radial and the tangential stresses are combined using the following formula:

$$\sigma_{Combined} = \sqrt{\sigma_{Radial}^2 - \sigma_{Radial} * \sigma_{Tangential} + \sigma_{Tangential}^2}$$

Conduction heat transfer from the bottom of the floor to the foundation is offset by convection heat transfer from the inventory to the top of the floor. However, the convection coefficients are 1) strong functions of the plant operating mode, 2) vary locally across the floor, and 3) for the most part, are generally unknown.

If the convection coefficient can be simulated by multiplying the thermal conductivity of the salt by a factor of 100, then the radial gradient in the floor is a modest 4 °C. However, the associated stress at the center of the tank is nominally equal to the allowable stress for Type 347H stainless steel at the design temperature of 565 °C.

If the convection coefficient is modeled by increasing the thermal conductivity of the salt by a factor of 10, then the radial gradient increases to 35 °C, and the floor stresses exceed the yield value at the center of the tank.

The limiting case is one in which the inventory is stagnant, and all of the heat transfer from the inventory to the floor is by conduction. This has the potential to develop radial temperature gradients in excess of 100 °C, and yielding of the floor is essentially assured. It can be noted that this is not a hypothetical case. If the hot tank is at a temperature of, for example, 500 °C and the plant enters a weather shutdown lasting 3 or 4 days, then the electric heaters for the tank will not start operation until the temperature of the inventory has decayed to the normal operating temperatures of the heaters; i.e.,

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275 to 290 °C. If the heaters are not in operation, the inventory will remain stagnant, and the potential for permanently damaging the tank floor is high.

Friction Forces Between the Floor and the Foundation

Friction forces due to thermal expansion of the tank are a combination of radial forces and tangential forces. The differential equation for estimating the stresses is as follows:

$$\frac{d\sigma_{rr}}{dr} + \frac{1}{r}(\sigma_{rr} - \sigma_{\theta\theta}) = -\frac{\mu \rho H g}{t_p}$$

where σ_{rr} is the radial stress, kg/m-sec² (i.e., Pa)

r is the radius, m

$\sigma_{\theta\theta}$ is the tangential stress, Pa

μ is the coefficient of friction, []

ρ is the density of the salt, kg/m³

H is the depth of the salt, m

g is the acceleration due to gravity, m/sec²

t_p is the thickness of the floor, m

Solving for the radial stresses yields the following:

$$\sigma_{rr} = \frac{E}{(1 - \nu^2)} \left(\frac{du}{dr} + \nu \frac{u}{r} \right)$$

where E is the modulus of elasticity, MPa

ν is Poisson's ratio, []

u is the local deflection, m

r is the local position on the tank radius, m

Solving for the tangential stresses yields the following:

$$\sigma_{\theta\theta} = \frac{E}{(1 - \nu^2)} \left(\nu \frac{du}{dr} + \frac{u}{r} \right)$$

Using representative values for a hot tank in a commercial project, the peak stresses in the floor occur near the center of the tank, and reach values in the range of 135 to 140 MPa, under the following conditions:

- The friction coefficient is 0.3

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- The temperature of the inventory changes by an amount ($\sim 15\text{ }^{\circ}\text{C}$) which is sufficient to develop the full friction load on the floor.

A stress of 135 MPa is essentially equal to the yield stress of stainless steel at a temperature of $560\text{ }^{\circ}\text{C}$. If the floor is exposed to cyclic stresses at, or approaching, the yield value, then the low cycle fatigue life of the floor will be significantly reduced.

It can be noted that the uncertainty regarding the coefficient of friction between the floor and the foundation is large. Some experimental work with refractory materials at high temperatures show coefficients to be in the range of 0.5 to above 1.0. If the coefficient is greater than 0.4, then the calculations show that plastic deformations in the floor near the center of the tank are likely.

7.4 *Steam Generation System*

7.4.1 System Description

The steam generator converts the thermal energy in the salt into main and reheat steam suitable for use in the turbine-generator. The primary system components include the following:

- Superheater
- Reheater
- Evaporator
- Steam drum
- Preheater
- Startup feedwater heater
- Hot salt circulation pump
- Cold salt attemperation pump
- Evaporator recirculation pump (forced recirculation evaporator)
- Preheater recirculation pump
- Startup electric water heater
- Startup cold reheat steam electric heater.

The salt components are located on a platform above the thermal storage tanks. This allows the equipment to drain back to the tanks for long-term freeze protection.

A schematic flow diagram on the water/steam side of the system is shown in Figure 7-20, and a schematic flow diagram on the salt side is shown in Figure 7-21.

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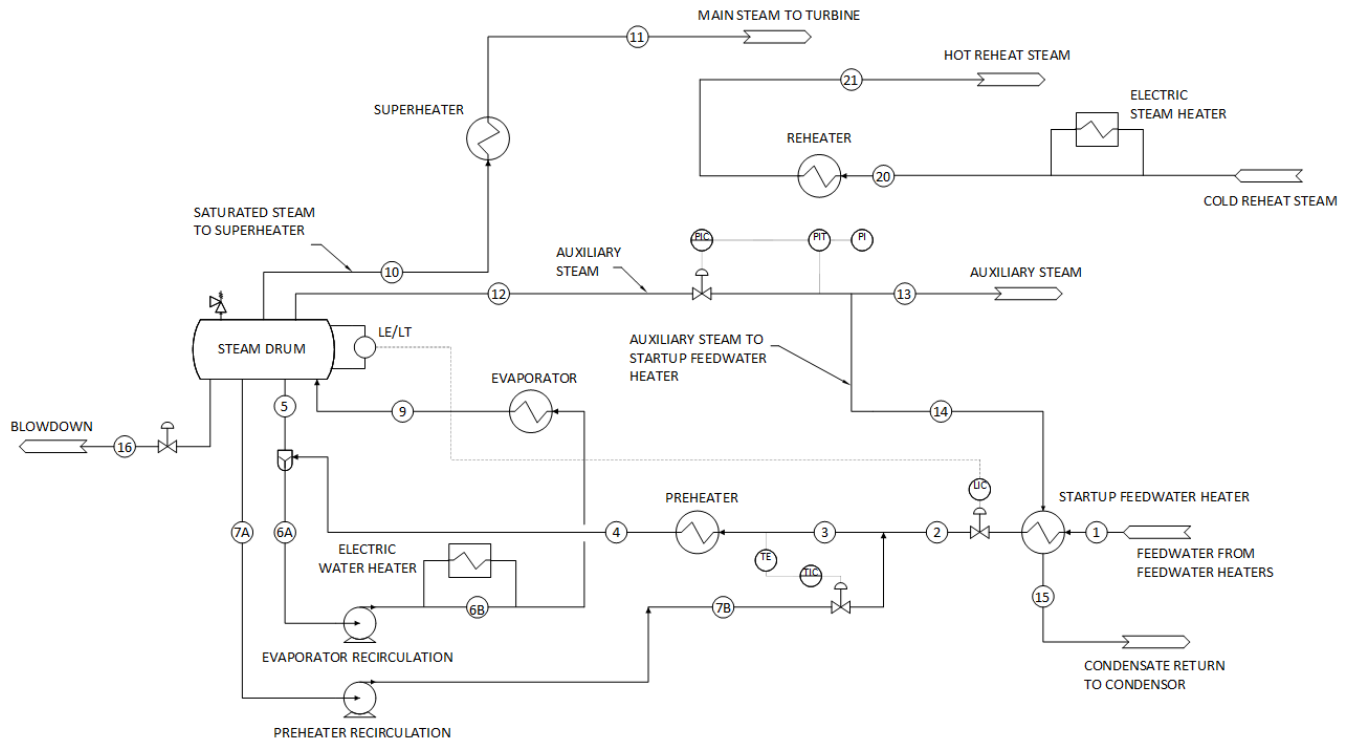


Figure 7-20 Flow Diagram on the Water/Steam Side of the Steam Generator

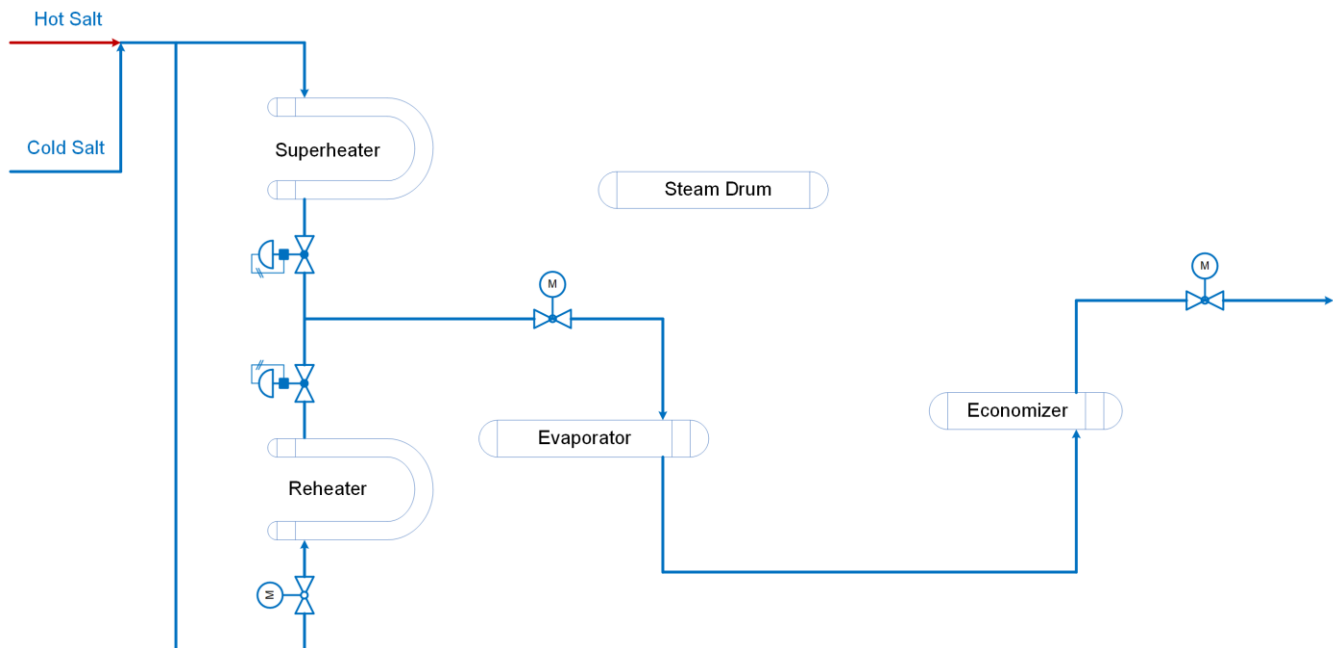


Figure 7-21 Flow Diagram on the Salt Side of the Steam Generator

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Several characteristics are illustrated by the flow diagrams, as follows:

- The hot salt entering the steam generator divides into 2 streams, with the majority of the flow passing through the superheater, and the balance of the flow passing through the reheater. The flow from the cold end of the superheater combines with the flow from the cold end of the reheater, and the total flow passes in series through the evaporator and the preheater. This is in contrast to the flow arrangement in the steam generator of a parabolic trough project. In the latter case, the majority HTF flow passes through the superheater / evaporator / preheater in series, and a minority HTF flow passes separately through the reheater only. The two flows combine downstream of the steam generator. The different flow arrangements are adopted for the following reasons:
 - In a trough project, the cold reheat steam temperature (210 °C) is lower than the final feedwater temperature (225 °C). As such, the HTF temperature at the cold end of the reheater is lower than the HTF temperature at the cold end of the preheater, and combining the two HTF flows downstream of the steam generator results in the minimum cold HTF temperature.
 - In a central receiver project, the cold reheat steam temperature (360 °C) is higher than the final feedwater temperature (245 °C). As such, the minimum salt temperature at the cold end of the preheater is achieved by taking the salt from the cold end of the reheater, and further cooling the salt by transferring heat to the water/steam sides of the evaporator and the preheater.
- During full load operation, the final feedwater temperature (245 °C) is high enough to prevent salt solidification at the cold end of the preheater. However, during turbine startup and low load operation, final feedwater temperatures are in the range of 180 to 240 °C, and salt solidification can no longer be avoided. To prevent this situation, the temperature of the feedwater at the cold end of the preheater is increased by one, or both, of the following methods:
 - A recirculation pump draws suction from the drum, and sends saturated water to the cold end of the preheater. Direct contact heat transfer raises the mixed water temperature to a nominal value of 250 °C.
 - Saturated steam from the drum, or throttled and attemperated steam from the hot end of the superheater, is supplied to the shell side of a startup feedwater heater. The startup heater is located downstream of the last extraction feedwater heater and upstream of the preheater. The temperature of the feedwater at the hot end of the startup heater is a nominal 250 °C.

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A discussion of the relative advantages and disadvantages of the two approaches is presented in Section 6.5.4 of Volume 3 - Narrative.

- The electric water heater is used to preheat the equipment prior to the filling with salt. The preheater, the evaporator, and the drum are preheated by direct contact with the circulating water. In contrast, the superheater and the reheater are preheated by sending saturated steam from the drum through the two heat exchangers in series. Condensate produced from the condensing steam is routed to the condenser through the live steam and the reheat steam line drains. Even though the capacity of the electric heaters is relatively large (5 to 6 MWe), the preheating process for the superheater and the reheater is somewhat lethargic, and preheat times from ambient conditions are on the order of 24 hours.
- During steam generator startup, and prior to turbine operation, live and reheat steam is sent directly to the condenser through a bypass system. The bypass system consists of a high pressure throttle valve, a high pressure steam attemperator, a low pressure throttle valve, and a low pressure steam attemperator. The high pressure throttle valve and attemperator are located between the hot end of the superheater and the cold end of the reheater. The low pressure throttle valve and attemperator are located between the hot end of the reheater and the condenser. The high pressure throttle valve and attemperator replicate the enthalpy drop across the high pressure turbine, and the low pressure throttle valve and attemperator replicate the enthalpy drop across the intermediate pressure / low pressure turbine.

Early in the startup process, the temperature of steam leaving the high pressure throttle valve is low enough (220 °C) that salt solidification at the cold end of the reheater is possible. To prevent this condition, the temperature of the cold reheat steam is increased to a value of at least 265 °C by means of an electric steam heater. In a commercial project, the capacity of the electric heater is in the range of 1.5 to 2.5 MWe.

7.4.2 Heat Exchanger Configuration

The Rankine cycles in essentially all solar projects are a single reheat design. Correspondingly, the steam generators in essentially all solar projects use a superheater, a reheater, an evaporator, and a preheater.

A common approach to the fabrication of the heat exchangers is a U-tube bundle in a straight shell, with 2 passes on the shell side. The fluid on the shell side is typically salt. This is done for two reasons:

1. The high pressure fluid is the water/steam, and accommodating a high pressure fluid in a tube is easier to accomplish than accommodating a high pressure fluid in a shell

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2. The salt in the heat exchanger can reasonably be expected to freeze at least once in the life of the project. If the salt is on the tube side, and freezes in the tubes, then the solid salt can form essentially immovable plugs. Thawing the tubes anywhere along the length of the frozen plug runs the risk of yielding the tube due to the nominal 4 percent volume change on thawing. If the freeze/thaw cycle is repeated, then there is the possibility of rupturing the tube. (This was the failure mechanism for the tube bundle in the kettle evaporator at the Solar Two demonstration project.) In contrast, if the salt is on the shell side, the thawing process begins by ensuring that the salt in the lines to and from the heat exchanger is liquid. Activating the electric heat tracing on the shell and the channel first melts the salt on the inner surface of the shell and the channel. Since the heat input from the heat tracing is typically a modest value, the initial mass of salt which melts is also a modest number. The volume change on thawing still occurs, but the liquid salt has an exit path through the salt lines. As such, the potential for plastically yielding the shell is low.

Most commercial heat exchangers use a flat tubesheet. During fabrication, the tubes are inserted into the tubesheet, the ends of the tubes are strength welded to the tubesheet, and the tubes are then rolled into the tubesheet.

An alternate commercial arrangement replaces the flat tubesheet with a section of pipe. The heat exchanger is fabricated as follows:

- A series of holes are drilled along the length, and around the circumference, of the pipe
- Short, tapered nozzles are welded to the pipe at each hole
- The ends of the heat exchanger tubes are welded to the ends of the short nozzles.

In some commercial projects, the evaporators use forced circulation, while other projects use natural circulation. The selection is largely the responsibility of the equipment supplier.

7.4.3 Cyclic Operation

In a solar project, the steam generator, almost by definition, undergoes a daily startup and shut down cycle.

During overnight hold, cold salt is circulated through the steam generator. The salt is supplied by the cold salt attemperation pump, and the salt leaving the steam generator is returned to the cold salt tank. The heat input from the cold salt compensates for the heat losses from the heat exchangers and associated piping. The heat input also heats the makeup water added to the steam drum to compensate for steam and condensate losses in the Rankine cycle.

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To start the steam generator, the hot salt pump is started, and hot salt is blended with cold salt from the attemperation pump at a point upstream of the superheater / reheater. Early in the startup process, the steam flows produced by the superheater and the reheater are not at temperatures high enough to admit to the turbine. The live steam and the hot reheat steam flows are attemperated and throttled, and the flow is then directed to the condenser through a turbine bypass system.

When the live steam, the reheat steam, or a combination of live and reheat steam reach temperatures which can be accepted by the turbine, a portion of the steam flow in the bypass system is sent to the turbine. The turbine is preheated based on a schedule defined by the vendor, synchronized, and loaded to a value that will prevent a trip on reverse power. The steam flow in the bypass system is eventually stopped, with all of the steam flow now directed to the turbine.

The relative proportions of hot salt and cold salt at the mixing point upstream of the superheater / reheater are adjusted until the flow of cold salt reaches zero. The output of the turbine is adjusted by changing the speed of the hot salt pump.

To shut down the steam generator, the above process is essentially reversed.

The blending process of adding hot salt to cold salt is consequence of the vendor limits on allowable rates of temperature change and thermal shock. Typical limits include a maximum rate of temperature change of 10 °C/min, and a maximum thermal shock (Fluid temperature - Metal temperature) of 60 °C. The limits are developed by the vendor to ensure that the heat exchangers have a low cycle fatigue life equal to the life of the project.

The vendor also specifies a minimum allowable flow rate on both the shell- and tube-sides. A typical value is 16 percent, which corresponds to a turndown ratio of 6:1. At low flow rates, the pressure drops on both the tube- and the shell-sides approach very low values. Local effects due to friction and buoyancy can then influence the flow distribution, which can produce stratified flow, channel flow, or both. If the flow distribution is not uniform, then the temperature distribution within the heat exchanger is also not uniform. Non-uniform temperature distributions can produce internal stresses which are non-uniform, unpredictable, and oscillating. These stresses may add to, or subtract from, the normal process stresses, which can lead to meaningful reductions in the low cycle fatigue life of the equipment.

7.4.4 Startup and Shutdown

A common steam generator arrangement in commercial projects consists of the following:

- Two 50-percent capacity steam generator trains
- Three 50-percent capacity hot salt pumps, discharging to a common header

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- A common minimum flow recirculation valve for the three hot salt pumps
- Two 100-percent capacity cold salt attenuation pumps, discharging to a common header
- A common minimum flow recirculation valve for the two attenuation pumps
- Salt control valves, at the cold ends of the superheater and the reheater, for distributing the relative flows between the two heat exchangers
- A startup feedwater heater, located downstream of the last high pressure feedwater heater and upstream of the steam generator preheater.

The salt pump and discharge valve arrangement is shown in Figure 7-22, and the intra-heat exchanger salt valve arrangement is shown in Figure 7-23.

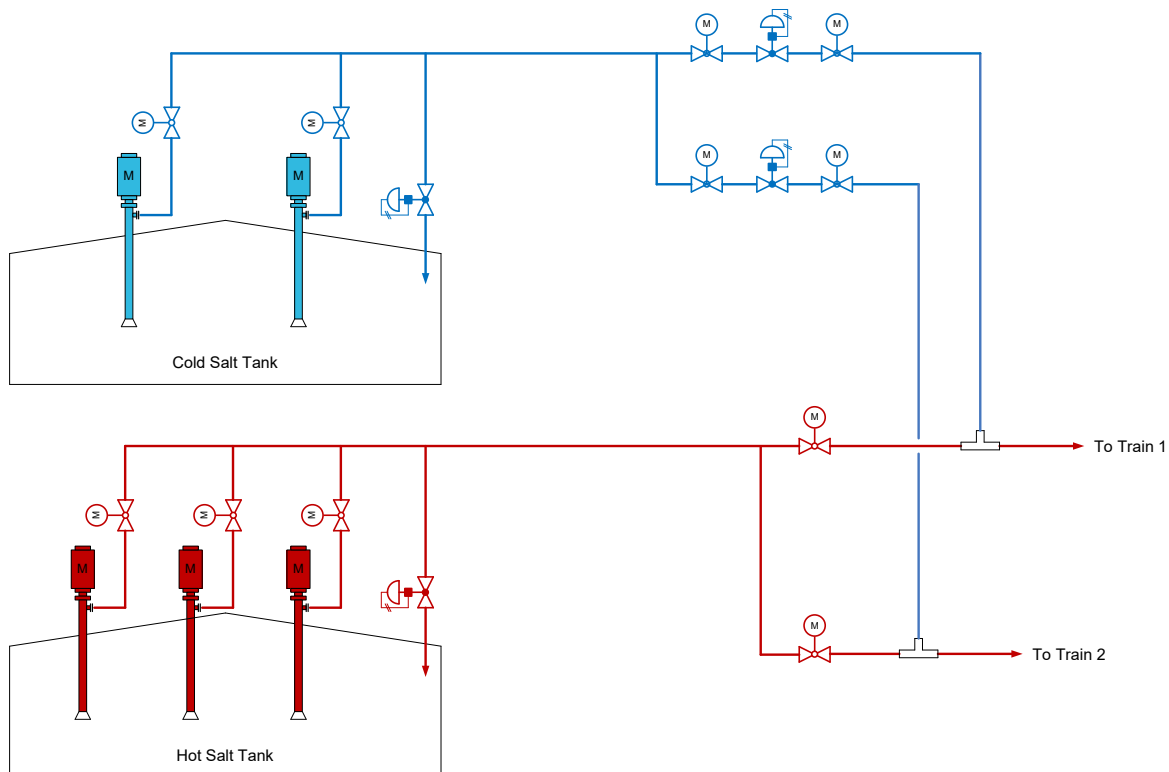


Figure 7-22 Salt Pump Arrangement in a Representative Commercial Project

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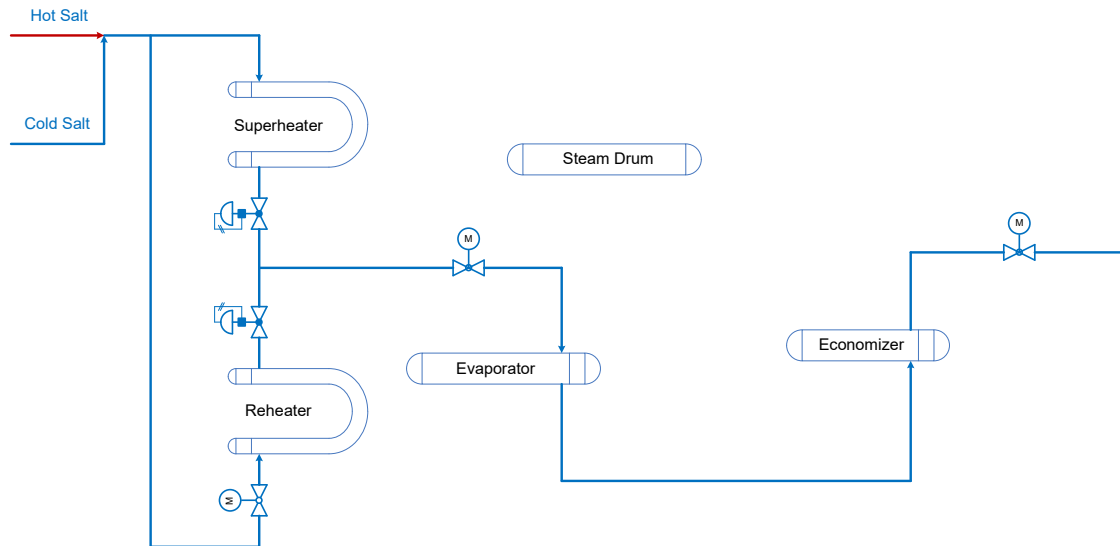


Figure 7-23 Intra-Heat Exchanger Salt Valve Arrangement in a Representative Commercial Project

This arrangement reduces the number of control valves to a minimum, and likely offers the lowest cost piping arrangement. However, the arrangement carries a range of liabilities, as discussed in Section 6.4.4 of Volume 3 - Narrative.

7.4.5 Steam Turbine Trips

When the steam turbine trips, the live steam and the reheat steam stop valves immediately close. To prevent the saturation pressure in the steam drum from increasing to a value which would open (and likely damage) the safety valves, two options are available:

- Trip the hot salt pumps to stop the energy addition to the steam generator
- Continue the flow of hot salt, and place the steam bypass system into service.

The first option is extremely effective. However, the metal and the salt temperatures in the evaporator are necessarily above the saturation temperature. As such, saturated steam production continues for some period of time (minutes) by converting the sensible heat in the metal and the salt to latent heat in the steam. The metal temperatures in the hot section of the evaporator tubesheet (450 °C) decay to the saturation temperature (325 °C) at a rate which is likely an order of magnitude greater than the allowable rate of 10 °C/min. To prevent damage to the evaporator, the second option is preferred.

If the turbine trip can be cleared in a matter of minutes, then the turbine can be immediately restarted as the steam generator is already in service at the normal design conditions. If the turbine trip cannot be

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cleared in a short period of time, then the steam generator should be shut down following the normal process to protect the heat exchangers.

7.4.6 Steam Generator Trips

At some point, it will be necessary to trip the steam generator. This involves tripping, or placing into recirculation, the following pumps: hot salt; evaporator recirculation; preheater recirculation; feedwater; and condensate.

The fluid flows on both the tube side and the shell side of the heat exchangers stop in unison. As such, the normal metal temperature profile along the steam generator train is preserved. The one exception is the evaporator, which reaches a uniform metal temperature equal to the saturation temperature within a few minutes.

This condition represents a conundrum, as it is not possible to restart either of the fluid flows without thermally shocking some portion of the steam generator train.

The safest course of action is to do nothing. Specifically, the operators would wait until conduction heat transfer within the heat exchanger, in combination with conduction / convection heat transfer to the environment, results in metal temperature which are within the thermal shock limit (100 °C) associated with resuming a flow of cold salt through the complete train. However, as one might imagine, the rate of temperature decay in large, well-insulated equipment is very low, and a waiting period of one to several days will be needed.

If the project is in a forced outage, waiting for the steam generator to cool to a safe restart temperature, then the loss in revenue will be in the range of \$100,000 to \$500,000. If the project is not willing to accept the loss in revenue, then the project can restart the steam generator at any time. This would likely be done by establishing a flow of cold salt from the attemperation pump, and quickly cooling the heat exchangers to the normal startup temperature of 295 °C. However, the project must also accept a potential, and largely undefined, reduction in the low cycle fatigue life associated with thermally shocking the heat exchangers in a non-standard shut down.

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Appendix A. Nitrate Salt Properties

The nitrate salt, which is a mixture of sodium nitrate and potassium nitrate, acts as the receiver coolant, the thermal storage medium, and the heat transport fluid in the steam generator. The salt has several thermophysical properties which make it suitable as a heat transport fluid and storage medium, including:

- High densities, in the range of 1,700 to 1,900 kg/m³
- Acceptable thermal conductivities, in the range of 0.50 to 0.56 W/m-°C
- Acceptable specific heats, in the range of 1.50 to 1.55 kJ/kg-°C
- Low absolute viscosities, in the range of 0.0010 to 0.0036 kg-m/sec
- Very low vapor pressures, on the order of several Pascals
- Low corrosion rates for carbon steels at temperatures up to 400 °C, and low corrosion rates with stainless steels at temperatures up to 600 °C.

The largest difficulty with nitrate salt is a freezing point of approximately 230 °C.

The freezing point of the salt mixture, together with its corrosion characteristics, effectively define an operating temperature range of 250 °C to 600 °C. To provide a safety margin on the freezing point, a lower temperature limit of approximately 275 °C is often used. Together with the characteristics of a subcritical Rankine cycle, the following design parameters are considered representative of a commercial project: 295 °C cold salt tank temperature; 125 bar live steam pressure; 540 °C live steam temperature; 540 °C hot reheat steam temperature; and 565 °C hot salt tank temperature.

The nitrate salt is a mixture of 64 mole percent sodium nitrate (NaNO₃), and 36 mole percent potassium nitrate (KNO₃), which is equivalent to 60 weight percent NaNO₃, and 40 weight percent KNO₃.

Note: The mixture is not the eutectic. The eutectic is a mixture of 50 mole percent NaNO₃, and 50 mole percent KNO₃. For solar applications, the fraction of NaNO₃ is increased to reduce the cost of the salt. Increasing the NaNO₃ fraction raises the melting point from 222 °C for the eutectic to a nominal 238 °C. However, the increase in the melting point can be safely accommodated through careful design of the steam generator.

Temperature range The salt mixture can be used over a temperature range of 260 °C to approximately 621 °C.

Freezing point As temperature decreases, the mixture starts to crystallize at 238 °C, and is completely solid at 221 °C.

Isotropic compressibility (NaNO₃) at the melting point $2 \times 10^{-10} \text{ m}^2 / \text{N}$.

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Heat of fusion (based on the average of heat of fusion of each component) $h_{sl} = 161 \text{ kJ/kg}$

Change in density upon melting $\Delta V / V_{\text{solid}} = 4.6\% \Rightarrow V_{\text{liquid}} = 1.046 V_{\text{solid}}$

A list of fluid properties, over a range of temperatures, is shown in Table A-1.

Table A-1 Nitrate Salt Properties Over a Range of Temperatures

Temperature, C	Density, kg/m ³	Specific heat, kJ/kg-°C	Absolute viscosity, kg/m ² -sec	Thermal conductivity, W/m-°C
220	1,950	1.481	0.00578	0.485
240	1,937	1.484	0.00501	0.489
260	1,925	1.488	0.00434	0.492
280	1,912	1.491	0.00376	0.496
300	1,899	1.495	0.00326	0.500
320	1,886	1.498	0.00284	0.504
340	1,874	1.501	0.00249	0.508
360	1,861	1.505	0.00220	0.511
380	1,848	1.508	0.00196	0.515
400	1,836	1.512	0.00178	0.519
420	1,823	1.515	0.00163	0.523
440	1,810	1.519	0.00152	0.527
460	1,797	1.522	0.00143	0.530
480	1,785	1.526	0.00137	0.534
500	1,772	1.529	0.00131	0.538
520	1,759	1.532	0.00127	0.542
540	1,747	1.536	0.00122	0.546
560	1,734	1.539	0.00116	0.549
580	1,721	1.543	0.00109	0.553
600	1,708	1.546	0.00099	0.557

The fluid properties of nitrate salt, each as functions of temperature between 250 °C and 600 °C, are described below. The properties are nominally independent of pressure.

Density, as a function of temperature:

$$\rho (\text{lb}_m / \text{ft}^3) = 131.2 - 0.02221 * T (^\circ\text{F})$$

$$\rho (\text{kg} / \text{m}^3) = 2090 - 0.636 * T (^\circ\text{C})$$

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Specific heat, as a function of temperature:

$$c_p \text{ (Btu / lb}_m\text{- }^\circ\text{F)} = 0.345 + (2.28 \times 10^{-5}) * T \text{ (}^\circ\text{F)}$$

$$c_p \text{ (J / kg - }^\circ\text{C)} = 1443 + 0.172 * T \text{ (}^\circ\text{C)}$$

Absolute viscosity, as a function of temperature:

$$\mu \text{ (lb}_m\text{ / ft - hr)} = 60.28440 - 0.17236 * T \text{ (}^\circ\text{F)} + (1.76176 \times 10^{-4}) * (T \text{ (}^\circ\text{F)})^2 - (6.11408 \times 10^{-8}) * (T \text{ (}^\circ\text{F)})^3$$

$$\mu \text{ (mPa - sec)} = 22.714 - 0.120 * T \text{ (}^\circ\text{C)} + (2.281 \times 10^{-4}) * (T \text{ (}^\circ\text{C)})^2 - (1.474 \times 10^{-7}) * (T \text{ (}^\circ\text{C)})^3$$

Thermal conductivity, as a function of temperature:

$$k \text{ (Btu / hr - ft - }^\circ\text{F)} = 0.253208 + 6.26984 \times 10^{-5} * T \text{ (}^\circ\text{F)}$$

$$k \text{ (W / m - }^\circ\text{C)} = 0.443 + 1.9 \times 10^{-4} * T \text{ (}^\circ\text{C)}$$

Properties of solid salt are as follows:

Density, ρ

NaNO ₃	2,260 kg / m ³ at ambient temperature
KNO ₃	2,190 kg / m ³ at ambient temperature

Heat capacity, c_p

NaNO ₃	37.0 cal / °C - mol = 1,820 J / kg - °C near the melting point
KNO ₃	28.0 cal / °C - mol = 1,160 J / kg - °C near the melting point

Thermal conductivity, k

KNO ₃	2.1 W / m - °C

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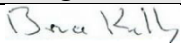
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1. Introduction, Report Format, Acknowledgements, and Disclaimer

1.1 Introduction

The work proposed concentrates on a Design Basis Documents / Owner's Technical Specification for parabolic trough and central receiver power plants using nitrate salt as the heat transport fluid and the thermal storage medium. The goals are to 1) distill the successful experience with nitrate salt systems from as many commercial projects as possible, 2) provide technical bases for equipment design/selection that an owner can impose on an EPC contractor, 3) compile this information in one location to provide guidance on salt systems that is as broadly applicable to as many projects as possible, and 4) work toward an industry consensus on a Design Basis Document.

1.2 Project Goal

The principal goal of the project is to develop a Design Basis Document for nitrate salt systems in CSP projects that reflects the lessons learned from earlier commercial projects. The document allows an owner to provide design guidance, and to impose a minimum set of requirements, above and beyond those in the normal Codes and Standards, on an EPC contractor in an effort to avoid a repeat of past mistakes. If successful, this would allow salt systems to achieve the same levels of reliability and availability as commercial parabolic trough plants using diphenyl oxide / biphenyl (i.e., Dowtherm A, Therminol VP-1) as the heat transfer fluid.

1.3 Report Format

The report consists of 3 volumes:

- Volume 1 - Specifications for Parabolic Trough Projects
- Volume 2 - Specifications for Central Receiver Projects
- Volume 3 - Narrative

Volumes 1 and 2 include discussions of the following topics:

- Plant functional requirements for the salt systems
- Plant operating states, and transitions between states, for the salt systems
- Risk analysis of the principal salt components

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- Plant requirements for the salt systems to meet the functional and the operating requirements
- Type of specification for the principal salt components: functional; or prescriptive
- Current state of the art for salt systems.

Volume 3 discusses a number of cases in which the salt equipment at commercial parabolic trough and central receiver projects has not met the projected levels of reliability and availability. Possible reasons for the sources of the problems are discussed, as is a range of possible alternate designs that could avoid the known problems.

1.4 Acknowledgements

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1.5 Disclaimer

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

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2. Inorganic Heat Transport Fluids in Parabolic Trough Projects

2.1 *Fluid Selection*

A range of potential inorganic fluids can be considered as the working fluid, including the following:

- Binary, ternary, and quaternary nitrate salt mixtures
- Ternary nitrate / nitrite salt mixtures
- Quaternary nitrate / nitrite salt mixtures.

A review of the benefits and the liabilities of 6 candidate mixtures is presented in Section 10.3 of Volume 1 - Parabolic Trough Specifications.

The ideal salt would have 1) a high upper temperature limit in terms of thermal stability, 2) a low freezing point, 3) a high specific heat, and 4) be inexpensive. As one might expect, none of the candidate salts fulfills all of the ideal requirements.

As discussed in Section 10.3 of Volume 1, the mixture which best meets all of the requirements is the binary mixture of 60 weight percent sodium nitrate and 40 percent weight percent potassium nitrate. Although it has the highest melting point (a nominal 235 °C, depending on the mixture fractions and the nitrite concentration) of the candidate salts, the benefits of a high thermal stability and a low material cost typically offset the liability of a high cost for freeze protection system in the collector field.

2.2 *Demonstration Facility*

The German Aerospace Center and the University of Évora in Portugal operate a test facility for molten salt technologies near the city of Évora ¹. The platform has a parabolic trough solar field, a salt storage system, and a steam generator. The working fluid is the binary mixture of sodium nitrate and potassium nitrate.

In the solar field, HelioTrough 2.0 trough collectors form a circuit, that operates at outlet temperatures up to 565 °C. The collectors, with a total length of 684 meters, have a maximum thermal output of around 3.5 MWt.

The salt system comprises a hot and a cold storage tank, an underground dewatering tank, a salt melting system, in combination with pumps and valves to control the various operating modes.

¹ <https://www.dlr.de/en/sf/research-and-transfer/research-infrastructure/evora-molten-salt-platform>

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The steam generator has a capacity of 1.8 MWt at 140 bar and 560 °C. Pressure and temperature reducing stations and an air-cooled condenser are installed instead of a steam turbine.

The focal points of the research include:

- Investigation of material wear due to corrosion and thermal cycles, and the evaluation of the thermal stability and the ageing of salt mixtures
- Demonstration of the reliability of critical components, such as flexible pipe connections in the solar field and the once-through steam generator
- Validation of the procedures to prevent salt freezing, to fill and drain the system, for blackout scenarios, and to remove blockages caused by freezing.

The technology demonstration characteristics of the work at Évora, together with the results of the salt component tests conducted by Solar Dynamics at Solar TAC ², indicate that the use of nitrate salt as the working fluid in a commercial trough project is not imminent.

2.3 Temperature Control in the Collector Field During Non-Operating Periods

2.3.1 Introduction

Prior to filling with salt, the field piping, the receiver tubes, and the flex hose assemblies are preheated to a nominal temperature of 300 °C. This is to prevent thermally shocking the receiver tubes, and to prevent local freezing in the smaller lines. The field piping is preheated using the electric heat trace system (Section 2.3.2). The receiver tubes and the flex hose assemblies are preheated using a number of portable impedance heating systems (Section 2.3.3).

Once the system is filled, the primary means of temperature control is continuous circulation of cold salt during those periods in which the collector field is not in normal operation (Section 2.3.4).

Loss of Salt Circulation to the Solar Collector Assemblies

If salt circulation stops, then temperature control of the field piping is performed with the electric heat trace system. However, there is no permanent equipment to maintain control of the temperature of the receiver tubes and the flex hose assemblies. Since it is not feasible to rapidly drain the equipment in the collector field (i.e., within hours), a loss of circulation necessarily results in salt freezing in the receiver

² Shining, R. (Solar Dynamics, Broomfield, Colorado), SMART (Simplified Melting And Rotation-joint Technology) Final Report, DOE Contract DE-EE0008140, August 2022

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tubes and in the flex hose assemblies. Conceptual methods to recover from freezing in the receiver tubes and in the flex hose assemblies is described in Section 2.3.6.

Loss of Site Electric Power to the Field Piping

At some point in the life of the project, electric power could be lost at the site. This could occur due to the simultaneous loss of imported electric power from the grid, the loss of emergency power from the standby Diesel generator, or the loss of the transformer supplying electric power to the emergency bus. Should this happen for a period of a few days, then salt will begin to freeze in the field piping. The process for recovering from freezing in the field piping is discussed in Section 2.3.8.

2.3.2 Heat Trace System

All of the fixed salt piping in the collector field will use electric heat tracing, for the following purposes:

- Preheating the equipment prior to filling with salt
- Temperature maintenance during periods when the salt is not circulating
- Freeze recovery.

The design of the heat trace system is identical to the systems described in Section 9.8 and Section 10.6 of Volume 1 - Parabolic Trough Specifications, and in Section 6.8 of Volume 3 - Central Receiver Specifications.

2.3.3 Impedance Heating System

Impedance heating system operates on “Joule heating,” allowing electricity to flow through the pipe network, generating heat from the energy dissipated in the resistance of each component. The impedance heating system is intended for three primary functions:

1. Preheat a collector loop before initial filling with salt (in addition to subsequent filling operations following any maintenance tasks that require draining)
2. Freeze protection for a collector loop which has experienced a loss of flow
3. Freeze recovery of a collector loop which has experienced a loss of flow for an extended period.

The design of the impedance heating system is discussed in Section 10.9.2 of Volume 1, Specifications for Parabolic Trough Projects.

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2.3.4 Continuous Salt Circulation

As with the Therminol in commercial trough projects, once the field piping and receiver tubes are filled, the system remains filled throughout the life of the project. This is the case because there is no on-site storage equipment with sufficient volume to hold the entire inventory of the working fluid.

It can be noted that the salt tanks in the storage system are in one of two conditions: one is empty, and the other is full; or both are partially filled. In principle, there is a large permanent volume to accept salt drained from the collector system. However, if collector system is being drained, then the temperature of the salt from the collector system is likely to be 'low'. Introducing low temperature salt into the hot tank, or transferring salt from the cold tank to the hot tank to make space available in the cold tank, has the potential to thermally shock, and permanently damage, the hot tank. As such, the volume available in the storage tanks is not available to drain the collector field.

The basis operating mode for the collector field is continuous circulation, for the following reasons:

- Since there are no provisions for draining the entire system, the principal means of freeze protection in the field piping and in the receiver tubes is continuous circulation during non-sunny periods ³. Salt is circulated from the cold tank, through the collector field, and returned to the cold tank. Representative heat losses are on the order of 11 W/m² of collector aperture area.
- The inventory in the receiver tubes cools at a faster rate than the inventory in the field piping. Should the circulation stop for some period of time (say, hours), the temperature of the inventory in the field piping will be higher than the temperature of the inventory in the receiver tubes. When circulation resumes, there is the potential for thermally shocking the receiver tubes by introducing relatively warm salt from the field piping into the receiver tubes.

As such, the cold salt pumps must operate, essentially without interruption, for some 300,000 hours during the life of the project.

Thermal energy to compensate for the heat losses from the field piping and the receiver tubes is provided by the cold salt tank. If the temperature of the cold tank falls below a pre-determined value, then supplemental energy is supplied to the tank from external electric salt heaters. Electric power for the heaters is supplied from the grid, or should connection to the grid be lost, from standby Diesel generators.

³ Salt can be drained from individual loops by 1) closing isolation valves at the ends of the loop, 2) connecting a vacuum tank, located on the bed of a truck, to one end of the loop, 3) opening a vent valve at the other end of the loop, and 4) pushing salt from the loop into the vacuum tank by means of air flowing into the vent valve. Salt can be returned to individual loops by 1) closing the vent valve, 2) connecting the loop to the vacuum tank, 3) drawing a vacuum in the loop, 4) opening a vent valve on the vacuum tank to push salt into the loop, and 5) opening the isolation valves at the ends of the loop to establish salt circulation.

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2.3.5 Loss of Circulation Leading to Freezing in Receiver Tubes and Flex Hose Assemblies

In commercial projects, the reliability of the cold salt pumps, the electric salt heaters, and the standby Diesel generators is generally excellent, and the probability that continuous salt circulation can be maintained is regarded as high. Nonetheless, there are a number of conditions that could result in a loss of circulation, as follows:

- The switchgear supplying electric power to the cold salt pumps fails
- A crack develops, almost anywhere, in the headers of the field piping
- A defect in the insulation on the flex hose assemblies or the loop crossover piping leads to a frozen plug
- Salt isolation valves must be moved periodically to prevent sticking associated with corrosion. The isolation valve could move from open to closed, but then fails to re-open
- Operation of the salt pumps may require permissiveness from flow meters, pressure transmitters, or isolation valve positioners. If the signals from the flow meters, the pressure transmitters, or the valve positioners are lost, then the pump will trip. If it appears that repairs to the flow meters, pressure transmitters, or valve positioners will require a period that could challenge the freezing time of a receiver tube in the worst-case condition (broken glass envelope), then the DCS logic could be changed to bypass the permissive. At this time, the pumps could be restarted. However, having a qualified DCS engineer at the site to revise the logic is not a certainty, particularly at night, on the weekends, and during holiday periods
- The pumps develop a mechanical problem, such as a broken coupling, or there is a problem with the variable frequency drive for the motors
- An operator, in an attempt to reduce the auxiliary power demand of the salt circulation pumps, operates the pumps at too low of a flow (leading to overheating), too low of a speed (leading to vibration), or beyond the run-out point (leading to vibration). Overheating and vibration will damage a pump, which then requires the pump to be removed for repair.

It is difficult to quantify the above effects, but the probability that one of the above, or something similar to the above, will occur during the life of the plant is a value approaching 100 percent. Depending on the source of the problem, and depending on the number and the location of the isolation valves in the field piping, something in the range of $1/16$ to $16/16$ of the salt in the receiver tubes and flex hose assemblies could freeze.

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2.3.6 Freeze Recovery in the Receiver Tubes and Flex Hose Assemblies

Impedance Heating of Receiver Tubes

Previous studies looking at impedance heating of molten salt parabolic trough plants considered power densities up to 250 W/m for freeze recovery of the solar field ⁴. A recent set of computational fluid dynamics calculations of the melting process show a total melt time of 12.0 hours for heat collection elements in which the vacuum space is intact ⁵. (Melt times for receiver tubes with a lost vacuum or a broken glass envelope are the subject of separate studies.) The difference in heat losses between the bellows and receiver tube results in an acceptable maximum difference in temperature of 12 °C along the length of the receiver. Finite element calculations from the same study show a heat input of 250 W/m produces stresses of 40 MPa in the receiver tube and 9.5 MPa in the glass envelope, which are well below the yield strength of the absorber tube (205 MPa) and the glass envelope (27 MPa), respectively. The calculated stress values noted above assume the salt is free to expand during melting, as noted in previous freeze-thaw tests ⁶.

Another recent study considers a reduced order modeling approach to model and optimize a freeze recovery system for a commercial molten salt parabolic trough plant. In this case, the impedance heating system is limited by the cost of electrical utilities required to switch the plant from a 240 MW source to a 150 MW sink ⁷. Depending on the size of the project, the ambient conditions, and the capacities of the electric heating systems, the cost constraints resulted in estimated melt times for the entire solar field in the range of 100 to 300 days.

In order to avoid challenges with electrical infrastructure and reduce the capital cost of impedance heating systems for the plant, a mobile heating system has been proposed and evaluated ⁸. The mobile concept can be completely housed in trailers that can be moved by vehicle, each trailer would have a diesel generator and several DC power supplies depending on the plant geometry. It is estimated such a mobile system could significantly reduce the capital cost compared to fixed impedance heating circuits permanently installed at every collector.

⁴ D. Kearney, B. Kelly, and U. Herrmann, "Engineering aspects of a molten salt heat transfer fluid in a trough solar field," Energy, Vol. 29, No. 5-6, pp. 861-870, 2004

⁵ Shining, R., Imponenti, L., Kattke, K., and Marcotte, P. SMART: Simplified Melting And Rotation-joint Technology. United States: N. p., 2022. Web. doi:10.2172/1882508.

⁶ G. Kolb, C. Ho, B. Iverson, T. Moss, and N. Siegel, "Freeze-thaw tests on trough receivers employing a molten salt working fluid", Proceedings of ASME Energy Sustainability, 2010

⁷ C. Prieto, A. Rodríguez-Sánchez, F. J. Ruiz-Cabañas, and L. F. Cabeza, "Feasibility study of freeze recovery options in parabolic trough collector plants working with molten salt as heat transfer fluid," Energies, vol. 12, no. 12, 2019

⁸ Shining, Ryan, Imponenti, Luca, Kattke, Kyle, and Marcotte, Patrick. SMART: Simplified Melting And Rotation-joint Technology. United States: N. p., 2022. Web. doi:10.2172/1882508.

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During freeze recovery, salt in the headers and the flex hose assemblies should be melted prior to solar heating of the receiver tubes to allow the salt to expand in the axial direction. In a commercial system, there are many factors that cause salt to melt unevenly in a receiver tube (damaged glass envelopes, lost vacuum, broken mirrors, etc.), so it is possible that sections of the solar field could experience constrained salt expansion during freeze recovery. Modeling this phenomenon is very difficult. In addition, the initial conditions after freezing, in terms of void space in the receiver tubes, are not necessarily known. For these reasons, any freeze recovery system will have to be demonstrated at full scale (i.e., a complete loop) before commercial application. To date, demonstrations of freeze protection using impedance heating have occurred with nitrate salts in 2-collector and loops operating with design temperatures of 500 °C⁹, and freeze recovery has only been demonstrated on a smaller string of HCEs³.

Solar Heating of Receiver Tubes

The solar heating method, or controllable solar flux heating, uses solar energy for freeze protection and recovery. Recent studies suggest it is possible to thaw frozen salt in the receiver tubes if the heat input is attenuated to reasonable levels [Imponenti et al, 2020]. One method to accomplish this is via a solar ‘pass-through’, i.e., cyclically moving the mirrors back and forth through the focal point of the collector. The heat input to the receiver tubes is adjusted by pausing the mirrors in the off-focus position before completing the next pass-through cycle. This method is attractive as it requires no additional hardware; however, additional testing is required before the technology can be considered commercially viable.

The main problems regarding solar heating involve local non-uniform heat inputs and losses from the following effects:

- Receiver tubes with lost vacuum or broken glass envelopes
- Non-illuminated portions of the receiver tubes at the end of a collector at low sun incidence angles
- Broken or missing mirrors.

Additional aspects include the time and the weather conditions required for a full freeze recovery, which may restrict the effectiveness to the summer months at many plant locations.

While modeling studies have shown the stress levels in the receiver tubes to be well below allowable values, no analyses have been performed on the collector drives to determine if the continuous back and forth motion effects the lifetime of this component.

⁹ Gaggioli, W., Fabrizi, F., Rinaldi, L., D. Ascenzi, “Experimental tests about the cooling/freezing of the molten salts in the receiver tubes of a solar power plant with parabolic trough”, AIP Conference Proceedings 1850, 2017

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Since solar heating is only applicable for the illuminated portions of the receiver tubes, an impedance heating system will still be required for both the receiver tubes and the flex hose assemblies. Solar heating may be used to supplement the impedance heating systems, allowing capital and operational cost savings.

2.3.7 Loss of Site Electric Power Leading to Salt Freezing in the Field Piping

As noted above, salt freezing in the field piping requires the simultaneous loss of imported electric power from the grid and emergency power from the standby Diesel generator. The probability that both sources of electric power would be simultaneously lost is low. However, the project lenders will assign a non-zero probability to the condition, and the project will need to specify, and evaluate, a method for recovery.

2.3.8 Freeze Recovery in the Field Piping

Technical Considerations

In commercial solar projects, the electric heat tracing performs the following functions:

- Preheat an empty pipe from an initial temperature of about 25 °C to the required salt fill temperature of 275 °C. The preheat period, which determines the heat input to the pipe, ranges from 8 to perhaps 16 hours
- Once the pipe is filled, but there is no flow in the pipe, maintain the temperature of the line at a value of at least 275 °C to prevent freezing.

As one might imagine, the thermal input to the pipe to meet the preheat requirement can be fairly modest, as the thermal mass of the system is limited to the thermal mass of the metal, plus a nominal one-half of the thermal mass of the insulation. However, there will be instances in which the heat trace system will be required to thaw the salt in the field piping. In this case, the thermal mass of the system increases substantially; i.e., by the sum of the sensible heat required to increase the temperature of the salt from ambient to the melting point, plus the latent heat required to convert the solid salt to a liquid.

The time required to thaw a frozen line is a function of the diameter of the line. As the diameter of the line increases, the surface to volume ratio decreases, which generally increases the thawing period. In a commercial project, the largest line size is about DN 600 (24 in.). If the heat trace is designed to preheat an empty line in a period of 8 hours, then the required duty is a nominal 700 W/m. If the line, after filling, becomes frozen, then the time required to thaw the line is approximately 10 days.

If the heat trace system is operated with a duty cycle of 100 percent over a period of 10 days, then the pipe will reach a steady state temperature; in this case, a value of just over 600 °C. For a hot salt header,

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fabricated from stainless steel, this metal temperature is acceptable. However, for a cold salt header fabricated from carbon steel, the maximum metal temperature should be limited to 400 °C to prevent high metal corrosion rates. Doing so limits the allowable thermal input from the heat trace system to a nominal 400 W/m, which, in turn, increases the thawing period to about 20 days.

The 10 and 20 days thawing periods are based on 7 in. of calcium silicate insulation in an as-new condition. However, the quality of the insulation is expected to degrade over time due to 1) gaps developing in the insulation from daily thermal expansion and contraction cycles, 2) moisture intrusion, and 3) insulation compaction due to foot traffic. If the thermal conductivity of the insulation increases by 20 percent, then the thawing periods increase to about 12 days for the stainless steel lines and 24 days for the carbon steel lines.

A worst case condition be associated with local damage to the insulation which increased the thermal conductivity by a value of 50 percent. The damage could be due to personnel walking on the salt piping, incomplete repairs to the insulation, or salt leakage from a failed valve packing saturating the insulation. Should these conditions occur, the time to thaw a carbon steel header is on the order of 1 month.

As discussed in Section 6.4 (Risk Analysis) of Volume 1, Specification for Parabolic Trough Projects, the time to recover from a frozen collector field is in the range of 35 to 60 days. In practice, the time to recover may be longer, depending on the condition of the field piping insulation, the availability of the mobile impedance heating systems, and whether electric power to the field piping is interrupted during the freeze recovery process.

Financial Considerations

It is not unreasonable to expect that the first commercial projects will freeze salt in the collector field more often than expected, and the recovery times will be longer than expected. A potential, but not implausible, scenario might be one in which salt freezes in the field once in each of the first 5 years of commercial operation, and the recovery time in each case is 75 days. This would result in an availability in the first 5 years of commercial service of 70 to 75 percent, rather than the expected 90 percent. A worst case scenario might be on in which salt freezes twice in the first year of commercial service, and the recovery time in each case is 90 days. This would result in an availability of 50 percent in the first year of service.

Plant availabilities in this range, particularly early in the life of the project, will have a noticeable, and detrimental, effect on the return on investment to the equity providers. In principle, the equity providers can compensate for the availability risks by increasing the return on investment. However, this will increase the levelized cost of energy. At some point, the risk premium for the use of an inorganic heat transfer fluid will offset the benefits of using an inorganic fluid; i.e., a reduction in the unit cost of the storage system, and an increase in the gross Rankine cycle efficiency. The economics of a plant using an inorganic fluid may essentially be a on par with the economics of a plant using Therminol as the

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working fluid. In this case, the investment community would, in their own interests, continue with the current state of the art in trough technology, and the commercial potential of the advanced working fluid may not be realized.

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3. Oil-to-Salt Heat Exchangers in Parabolic Trough Projects

3.1 Configuration

Shell and Tube

The industry has generally adopted shell-and-tube heat exchangers for oil-to-salt heat exchangers. The designs include 1) fixed tubesheet with U-tube bundles, 2) floating tubesheet with straight-tube bundles, and 3) header/coil arrangements with circular tubesheets.

The combination of the heat transfer properties of salt and the optimum approach temperatures require long heating lengths, which, in turn, often results in 3 to 6 heat exchangers in series. For the heat exchangers at the low temperature end of the series, the shell and tube materials can be carbon steel. For the heat exchangers at the high temperature end of the series, the tubes are stainless steel to provide the required corrosion resistance to the chlorides in the salt.

Flat Plate

Rather than shell and tube designs, two commercial projects selected a flat plate design. Stainless steel plates are stamped to provide the desired flow pattern. A group of plates are stacked, and then seal welded at the edges. Distribution boxes at each end of the exchanger align the salt flow and the oil flow in alternating, counterflow layers. An external steel box frame provides the required resistance to bowing of the plates generated by the fluid pressures.

The design offers a favorable combination a large heat transfer area in a small footprint, and low pressure drops on both the oil- and salt-sides. The principal liability is a significant difference in the low thermal inertia of the plates, and the relatively high thermal inertias of the perimeter plate welds and the external frame. To limit the stresses in the transitions between the thin metal sections and the thick metal sections, the allowable rate of temperature change is typically restricted to values (~ 1.5 to $2\text{ }^{\circ}\text{C/min}$) which are lower than values for shell and tube designs ($\sim 10\text{ }^{\circ}\text{C/min}$). Unfortunately, a rate of $2\text{ }^{\circ}\text{C/min}$ is impractically restrictive for equipment that receives heat directly from the solar field. In commercial service, the heat exchangers developed cracks after only a few years in operation. The cracks allowed Therminol, at a pressure of 10 bar, to leak into the salt side, at a pressure of perhaps 2 bar. A pressure of 2 bar is less than the vapor pressure of Therminol at the design temperature. As a result, the Therminol flashes on the salt-side of the heat exchangers, carrying Therminol vapor into the ullage space of the storage tanks. Some of the Therminol eventually makes its way to the atmosphere, which represents both an emissions problem and a financial penalty to replace the lost Therminol.

On a point related to the leaks, it is difficult to access the portions of the heat exchangers where the cracks develop. The experience with leak repair has been decidedly mixed, and in some cases, the only method to repair the leak is to replace the heat exchanger.

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3.2 *Equipment Location*

The heat exchangers are large and heavy pieces of equipment. The lowest cost approach to supporting the equipment is to place the heat exchangers at grade. However, commercial projects have located the equipment in one of the following two locations:

- On an elevated platform between the storage tanks, with a height slightly above the mid-point of the wall height of the tank
- On an elevated platform between the tanks, at a height above the top of the wall of the tanks.

These approaches are not inexpensive, as 1 to 1.5 kg of structural steel are required to support each kg of equipment. In addition, the wind loads on an elevated structure translate to higher foundation costs. Nonetheless, the benefits associated with elevating the equipment are judged to outweigh the costs, for the following reasons:

- With the heat exchangers located at grade, a drain tank, located below grade, is required. The drain tank requires a pit, electric heat tracing, thermal insulation, a range of instruments, and two drain pumps. This approach will incur a cost (but it will be less than the cost of an elevated structure)
- The drain pumps in the drain tank will have a certain capacity, which in turn, defines the time required to drain the heat exchangers. If the heat exchangers must be drained quickly in an emergency, then the drain pumps may not have the desired capacity unless large, and expensive, pumps are initially selected
- If the height of the elevated platform is at least the height of the mid-point of the tank walls, then the heat exchangers can be drained by opening a valve to the tank with the lowest inventory level. This passive drain approach is more reliable, and has a higher availability, than the active drain method associated with a drain tank and drain pumps. It can be noted that the mass of salt in the heat exchangers and in the associated piping is 'small' relative to the mass of salt in a storage tank, even with the tank at the minimum level. Assuming that adequate mixing occurs between the inventory in the tank and the salt entering the tank, then the thermal transient experienced by the tank during a drain sequence will be well within the limits set by the tank supplier.
- Most importantly, the heat exchangers, at some point in the life of the project, will need to be drained for inspection and maintenance. If the heat exchangers are located above the storage tanks, then there is no method by which salt can enter the heat exchangers after the equipment has been drained, and expose the maintenance personnel to potential harm. Similarly, if the heat

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exchangers are located at the mid-point of the tank wall height, and if the inventory levels in both tanks are set the same, then there is no method for salt to enter the heat exchangers. This is a subtle point regarding safety. There have been instances in which pipes were allowed to cool to ambient temperature, forming frozen plugs in the lines. The project staff knew, with certainty, that there was no danger to the personnel working downstream of the frozen equipment. In separate cases, the frozen plugs thawed, resulting in severe injury in one case and death in a second. Locating the equipment on an elevated platform is, in effect, a form of insurance. And like all insurance, it is a complete waste of money, until it is needed.

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4. Salt Tanks in Parabolic Trough Projects

4.1 Material Selection

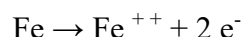
A range of corrosion studies have shown acceptable corrosion characteristics of carbon steel, at temperatures up to 400 °C, with chloride contents up to 0.6 percent ^{10,11}. As a result, all commercial parabolic trough projects fabricate the cold tank and the hot tank from carbon steel.

4.2 Corrosion Characteristics

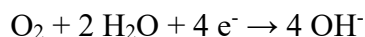
At least one commercial project has experienced problems with significant corrosion in the salt tanks. The repairs involved suspending the tank on external supports, and replacing a portion of the bottom course in the tank wall.

An examination of the sections removed showed evidence of stress corrosion cracking. This form of corrosion is different than the uniform surface corrosion processes observed in the Abengoa and the Sandia studies.

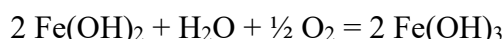
Stress corrosion cracking is a combined mechanical and electro-chemical corrosion process which results in cracking of certain materials ¹². The mechanism is not simply an overlapping of corrosion and mechanical stresses, but an auto-catalytic, self-accelerating process leading to high metal dissolution rates. The process includes an oxidation reaction:



The oxidation reaction is accompanied by a reduction reaction, typically involving the oxygen and water:



Further reactions can then occur, including:



¹⁰ Sanchez, L., and Madina, V. (Abengoa Solar NT, Sevilla, Spain), 'Estudio de corrosion del acero A516Gr70 en tanques piloto. Tiemp 1', Documento 12-0106-IN-641-000, July 2009

¹¹ Bradshaw, R. W., and Clift, W. M. (Sandia National Laboratories, Livermore, California), 'Effect of Chloride Content of Molten Nitrate Salt on Corrosion of A516 Carbon Steel', Sandia Report SAND2010-7594, November 2010

¹² Corrosion in Carbon Steels – IspatGuru

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The hydrated oxides can lose water during dry periods, and revert to anhydrous oxides, including ferrous oxide (FeO) and ferric oxide (Fe₂O₃, hematite).

Initially, a small pit is formed at the surface, and develops into a crack due to applied or residual stresses in the material. The crack formation opens up a new active (non-passive) metal surface, which corrodes very easily. This leads to further crack propagation and exposure of new highly active metal surfaces in the crack. Metal dissolution in the crack advances rapidly until mechanical failure occurs. Brittle failure can occur in normally ductile metals at stress levels which are less than the yield strength. Internal stresses in a material can be sufficient to initiate an attack of stress corrosion cracking.

Stress corrosion cracking of carbon steel is rare compared with stress corrosion cracking of stainless steel. One potential source of cracking in carbon steel is the exposure of the metal to liquid water in the presence of nitrates. One hypothetical failure mechanism is as follows:

- The outside of the tank is exposed to nitrate salt. The source of the salt could be 1) vapor condensation from ullage gases passing through a vacuum / pressure relief valve, or 2) salt leakage from pumps, valves, instruments, or piping above the tanks
- Water comes into contact with the outside of the tank. The source of the water could be 1) site flooding during a particularly heavy rain, 2) rainwater entering gaps in the tank insulation, or 3) maintenance personnel using wash water to remove the salt
- As noted above, process conditions (i.e., hydrostatic loads) can produce stresses, which are 1) within the ASME allowable values, but 2) high enough to initiate cracking. Additional stresses can be generated should water come into contact with the tank. Heat transfer will produce local temperature gradients, which, in turn, can generate local thermal stresses. Temperature gradients on the order of 45 °C can produce stresses equal to the ASME allowable value, and a gradient of approximately 115 °C can cause the material to yield.

Should the above combination of conditions occur, then a failure due to stress corrosion cracking may be the result.

To limit the potential for stress corrosion cracking, the following approaches can be adopted:

- If frozen salt is to be removed from the metal, methods should be limited to mechanical abrasion or media blasting
- To prevent runoff, following heavy rains, from coming into contact with the bottom of the tank, a comprehensive drainage system can be provided. Alternately, the tanks can be elevated above grade; however, this approach can be more expensive than a drainage system.

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- Ensure that the metal jacket covering the insulation is always free of gaps and defects
- To the extent possible, locate equipment, such as pumps, valves, instruments, and fittings, away from the top of the tank
- Substitute a ferritic material, such as 2¼ Cr - 1 Mo, for the carbon steel. The chromium offers a more robust surface passivation layer than can be provided by carbon steel, and the absence of nickel in the alloy limits the potential for stress corrosion cracking. It can be noted that 1) the reliability of salt tanks fabricated from carbon steel has been excellent, and 2) the occurrences of stress corrosion cracking in carbon steel are rare, and the sources of the corrosion are both understood and preventable. Selecting a ferritic material would only apply to rare cases where carbon steel would be routinely, and unavoidably, exposed to combinations of salt and liquid water.

4.3 *Survival Hypothesis for Carbon Steel Tanks*

The cold tanks in central receiver projects, and the hot and the cold tanks in parabolic trough projects, are fabricated from carbon steel. Known failures in carbon steel tanks include weld defects (1 tank) and stress corrosion cracking from water/salt mixtures on the tank exterior (several tanks). However, in contrast to stainless steel tanks, there have been no known failures due to leaks in the floor.

The similarities between carbon steel tanks, and the stainless steel hot salt tanks in central receiver projects, are numerous. The common features include the following:

- The system geometries are essentially identical: flat bottom tank with a self-supporting domed roof; insulated foundation with cooling air passages; and inlet salt flow distribution by means of a distribution ring header
- The floors consist of a staggered array of thin (7 to 8 mm) rectangular plates. The plates are connected to one another by butt welds. The floor deforms during the welding process, producing an array of shallow domes
- Prior to filling with salt, the tanks are preheated using combustion gases. Less-than-perfect control over the flow distribution of the combustion gases can produce radial temperature gradients in the floor, which are of sufficient magnitude to potentially buckle the floor
- When the tank is filled, hydrostatic forces push the floor flat against the foundation
- Residual stresses due to welding and floor deformations tend to persist; i.e., stress relaxation is moderate due to the low operating temperatures

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- Radial temperature gradients are an inherent characteristic of the tank and the foundation design, and the gradients have the potential to damage the floor
- Daily reversals in floor friction forces, due to daily thermal cycles, are also an inherent characteristic, and have the potential to damage the floor.

Nonetheless, there are important differences with the hot salt tanks in projects operating at design temperatures above 500 °C, as follows:

- The coefficient of thermal expansion for carbon steel (11 to $12 \cdot 10^{-6} / ^\circ\text{C}$) is approximately 70 percent of the coefficient of expansion for stainless steel (16 to $17 \cdot 10^{-6} / ^\circ\text{C}$). For a given temperature differential, the thermal stresses due to self-restraint in a carbon steel tank are about 30 percent lower than the thermal stress due to self-restraint in a stainless steel tank. This characteristic is important in reducing the effects of radial temperature gradients in the floor
- The thermal conductivity of carbon steel (40 to $45 \text{ W/m}\cdot^\circ\text{C}$) is at least a factor of 2 higher than the thermal conductivity of stainless steel (15 to $20 \text{ W/m}\cdot^\circ\text{C}$). The higher conductivity has a second-order, but useful, effect in terms of reducing thermal stresses during transient conditions
- For carbon steel, at 370 °C, the stress amplitude associated with a fatigue life of 40,000 cycles is 209 MPa. For stainless steel, at 540 °C, the stress amplitude associated with a fatigue life of 40,000 cycles, is 159 MPa. For a given stress amplitude, carbon steel accumulates fatigue damage at a lower rate than stainless steel
- In a central receiver project, during transient conditions, the salt flow enters the cold tank at a temperature that is higher than the bulk inventory. Due to density differences, the incoming flow rises, rather than sinks. This reduces the potential for local temperature gradients in the floor, and the associated thermal stresses
- The temperature gradients in the tank are a function of the inventory temperature, the thermal resistance of the insulation, and the degree to which the properties of the insulation have degraded over time. In a parabolic trough project, the inventory temperature in the hot tank is approximately 180 °C lower than the inventory temperature in the hot tank of a central receiver project. The lower temperatures reduce the potential to produce large temperature gradients, and thereby large thermal stresses, in the tank, particularly in cases in which the insulation has degraded or been subject to mechanical damage.
- The response time of the collector field in a trough project is longer than the response time of the heliostat field / receiver in a central receiver project. As such, the rates of temperature change in the hot tank of a trough project are relatively slow. This reduces the potential for non-isotropic

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temperatures in the bulk inventory, and reduces the magnitude of intra-tank thermal stresses due to self-restraint

- Both the cold tank and the hot tank in parabolic trough plants operate at temperatures below which creep damage can start to develop. The opposite applies to the hot tanks in central receiver projects
- The niobium in Type 347H generates niobium carbonitrides. The non-uniform precipitation of the carbonitrides, as discussed in Section 5.16.2, leads to non-uniform strength distributions, stain localization, internal cracks, internal voids, and eventually external cracks. The precipitation characteristics are the most pronounced in areas of high strains; i.e., the tops of buckles. The floors in carbon steel tanks likely experience similar deformations and areas of high strains. However, unlike Type 347H stainless steel, the strength and the ductility of carbon steel do not decay over time.

In general, if a carbon steel tank can survive the first 3 years of service, then it is likely to survive the full 30 years.

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5. Hot Salt Tanks in Central Receiver Projects

5.1 *Representative Commercial Design*

5.1.1 Tank

A representative storage quantity for a commercial project is 30,000 to 40,000 metric tons of salt. Since the salt has an extremely low vapor pressure (< 1 kPa at 600°C), the lowest cost storage approach is a vertical tank with a flat bottom and a self-supporting roof. Due to the chemical stability of the salt, the tank can be vented directly to the atmosphere.

Design standards for this type of tank, operating at temperatures up to 600°C , have yet to be developed. In the interim, the closest design standard is American Petroleum Institute Standard 650, Welded Tanks for Oil Storage. However, the Standard only considers operating temperatures up to 260°C , and low cycle fatigue conditions are not part of the Standard. As a result, a hybrid design approach has typically been adopted, as follows:

- For design temperatures above 260°C , allowable material stresses are derived from Section II of the ASME Code
- For the evaluation of low cycle fatigue, the methods described in Section VIII, Division 2, of the Code are often adopted.

It can be noted that, in some project jurisdictions, an evaluation of the fatigue life using Section VIII Division 2 is not required. However, this can present a substantial risk to the project in terms of the reliability of the storage system. A discussion of the recommended design codes is presented in Section 5.14.1.

For design temperatures above 538°C , the H grades of stainless steel, with a minimum carbon content of 0.04 percent, are required by the ASME Code. Candidate materials include Type 304H, Type 316H, and Type 347H stainless steel. At the 10 MWe Solar Two demonstration project, the hot tank was fabricated from Type 304H¹³. However, to avoid potential problems with intergranular stress corrosion cracking, the solar industry has adopted Type 347H, a stabilized stainless steel, in the current generation of projects.

5.1.2 Foundation

The foundation consists of the following elements, moving from the top down:

¹³ Pacheco, James (Sandia National Laboratories, Albuquerque, New Mexico), 'Final Test and Evaluation Results from the Solar Two Project', Sandia report SAND2002-0120, January 2002

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- A layer of solid lubricant; candidate materials include sand, perlite, and calcium silicate
- The primary tank support / insulating layer under the floor. Candidate materials include expanded clay, expanded glass, and refractory bricks
- A concrete base mat below the insulating layer
- A concrete ring wall to retain the insulating layer underneath the tank.

The foundation base mat has a series of cooling air pipes embedded in the concrete. Air flow in the cooling passages is established by fans or by natural convection. The air flow rate is set to maintain the soil temperature, directly beneath the foundation base mat, at values no higher than 75 °C. The goal is to limit the desiccation of the soil, and the oxidation of organic materials, to values that prevent unpredictable settlements of the tank.

5.2 *Failure Patterns*

The number of commercial central receiver projects in service, excluding those in China, is six: Gemasolar, Cerro Dominador, Crescent Dunes, Noor III, Redstone, and Noor Energy 1. The first four plants have been in service for some number of years, and the last two have recently entered service.

Among the first four, the hot salt tank at Gemasolar has failed, and a replacement tank has been built alongside the first tank. Also, the hot salt tank at Crescent Dunes has developed leaks in the floor on three occasions, and problems have developed with two other projects. For the plants which have been in service for more than 1 year, the failure rate, for all intents and purposes, is essentially 100 percent. Given the limited number of commercial projects the failure rate of hot salt tanks is not commercially sustainable.

With the limited number of cases, it is difficult to define a generic or a typical failure mechanism. Nonetheless, all of the tanks share a common set of characteristics which, when combined, are potential sources of future problems. The common design and operating characteristics include the following:

- The floor is fabricated from an array of rectangular plates. Welding the plates to one another generates residual welding stresses which are equal to, or higher, than the yield stress (see Section 5.5). Weld shrinkage establishes tensile stresses at the top of the plates, which are balanced by compressive stresses at the bottom of the plates. The stresses cause the plates to bow upward at the center, and the plates take on the shape of a shallow dome. The height of the dome is in the range of 50 to 150 mm. It can be noted that the floor is a large, thin sheet. The sheet, having been deformed by the welding process, has a low structural stability in terms of its ability to tolerate compressive stresses

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- The tank is filled with water for the hydraulic tests. When the hydrostatic pressure reaches a value of about 7 kPa (approximately 1 m of water), the plates are pushed flat against the foundation (see Section 5.6). Changing the shape of the plates from domed to flat generates compressive stresses at the edge of the plates that are at, or above, the yield stress. When the water is removed at the end of the hydraulic tests, the plates revert from flat back toward the original domed shapes. However, the residual stresses and the shape of the floor after the hydraulic tests will be different than the residual stresses and the shape of the floor before the hydraulic tests
- Radial temperature gradients in the floor are an inherent characteristic of a flat bottom tank on an insulated foundation (See Section 5.7). Specifically, the conduction heat transfer into the foundation near the center of the tank is 'low' due to the long conduction heat path from the center of the foundation to the ambient. In contrast, the conduction heat transfer into the foundation, and to environment, near the perimeter of the tank is 'higher' due to the relatively short conduction heat paths to the ambient. A radial temperature gradient establishes a combination of radial and tangential stresses in the floor. At some value for the gradient (yet to be defined, but perhaps 30 to 40 °C), the combination of radial and tangential stresses can reach the yield value near the center, and at the perimeter, of the tank
- The tank is preheated using combustion gases from a fired air heater (see Section 5.8). The gas flow patterns inside the tank are determined by the arrangement of the inlet duct and the outlet chimney. However, high temperature air is a relatively poor mechanism for convection heat transfer, and the uniformity of the gas flow pattern inside the tank is determined by the sophistication of the duct and chimney arrangement. Further, the thermal inertia of the floor is lower than the thermal inertia of the wall, and the thermal inertia of the floor is several orders of magnitude lower than the thermal inertia of the foundation. This combination of features has the potential to establish radial temperature gradients in the floor which are high enough to establish compressive stresses at, and above, the yield point. Should this happen, the floor, due to its low structural stability in compression, has the potential to develop further deformations. Preliminary finite element calculations show that the floor can develop a set of orthogonal buckles, which span the diameter of the tank, during the preheating process
- Three items can be noted regarding a finite elements analysis of the floor after the preheating process:
 - Static analyses will not reproduce the buckles seen in the floor; a quasi-static or a dynamic analysis is needed
 - The shape and the dimensions of the buckles are highly influenced by the shape and the dimensions of the initial deformations produced during welding and following the hydraulic tests

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- The deformations produced in the floor during welding, and the associated residual stresses, vary across the floor. The characteristics follow a statistical pattern, but they also contain random elements. Since the shape and the dimensions of the buckles are a function of the welding deformations and the residual stresses, the characteristics of the buckles (length, height, width, residual stress, and residual strain) also contain random elements. As such, developing a generic, or a typical, analysis predicting the creep-fatigue properties of the buckled regions is difficult. Nonetheless, it is possible to state that the creep-fatigue life of the buckled regions will be considerably less than the life of the project.
- When the tank is filled with salt, hydrostatic loads are sufficient to push the plates down flat against the foundation (see Section 5.9) for a second time. This removes the domed deformations that exist after the hydraulic tests (but does not remove any buckles which may have formed during the hydrostatic tests and the preheating process). When the plates transition from a domed shape to a flat shape, the edges of the plates are again placed into compression. The radial stiffness of the wall is several orders of magnitude greater than the radial stiffness of the floor, and some fraction of the compressive loads on the plates can be translated into additional deformations at the buckles, if present
- The rectangular plates used to fabricate the floor do not all have the same dimensions, the plates are not perfectly square, and the plates have some level of curvature. To accommodate the variations in the horizontal and the vertical gaps between plates, weld backing strips are typically used. The backing plates are intentionally welded to the bottom of the floor, and become a permanent feature of the tank. When the tank is heated from ambient to the design temperature for the first time, the backing strips push the sand in the top layer of the foundation toward the edge of the tank. This leads to 1) local portions of the floor which are unsupported, which generates bending stresses in the floor, and 2) local variations in the friction forces generated between the floor and the foundation
- After the tank enters commercial service, the temperature of the inventory increases during the day, and decreases during the evening and during shutdown periods. The thermal expansion and contraction of the tank generates friction forces on the floor, and these forces oscillate between compression and tension (see Section 5.10). Under certain conditions, the tank can be close to full and the temperature of the inventory can increase. The calculated compression stresses near the center of the tank can exceed the yield value. This, in turn, can produce additional plastic deformations, particularly if the displacements occur near buckles in the floor. The difficulty of analyzing this effect is compounded by the non-uniform distribution of friction resulting from the use of weld backing strips
- The flow entering the tank is mixed with the bulk inventory by means of a distribution header located about 1 m above the floor (see Sections 5.11 and 5.12). However, the distance between

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the center of the tank and the distribution header, and the distance between the distribution header and the wall of the tank, is on the order of 8 to 12 m. As such, non-isotropic temperature distributions in the inventory, on the time scale of minutes, can develop during transient conditions. Further, daily transient conditions (i.e., receiver startup) consist of relatively cold salt entering the hot tank. The difference in temperature between the salt entering the tank and the bulk inventory is ideally small ($< 10\text{ }^{\circ}\text{C}$). However, depending on decisions made by the operator, the difference can be as large as $100\text{ }^{\circ}\text{C}$. Under these conditions, cold salt will descend toward the floor due to density differences, which, in turn, can lead to local depressions in the temperature of the floor. Local variations in temperature on the order of $40\text{ }^{\circ}\text{C}$ can generate thermal stresses at, or above, the yield point. The associated strains can add to, or subtract from, the strains already present in the floor

- The commercial hot salt tanks built to date use Type 347H stainless steel. This material is a stabilized stainless steel, which provides additional resistance to intergranular stress corrosion cracking compared to the H grades of Type 304 and Type 316. The stabilizing element added to the alloy is niobium. However, thermodynamic considerations lead to the precipitation of niobium carbonitrides (see Section 5.16.2). Within the grain boundaries, a fine-grained precipitate of niobium carbonitride forms; at the grain boundaries, a coarse-grained precipitate is formed. On a time scale of years, a precipitate-free zone can develop between the grain boundaries. The strength of the coarse-grained precipitate is lower than the strength of the fine-grained precipitate, and the strength in the precipitate-free zone is lower still. When the material is subjected to tensile stresses, the material develops strains, but the strains are concentrated in the area with the lowest strength (precipitate-free zones). This leads to internal cracking, micro-void formation, or a combination of the two. The defects eventually progress to the exterior. The process is known as stress relaxation cracking, and it is accelerated by temperature and strain
- Any section of the floor which has buckled is a region of high strain, with the highest strains occurring at the top of the buckle. The strains at these locations are also influenced by friction forces between the floor and the foundation. The friction forces, which are highest near the center and at the perimeter of the tank, reverse direction daily due to the cyclic thermal expansion and contraction of the tank
- Stress relaxation cracking will proceed most quickly in the regions of the highest strains; i.e., the tops of the buckles. Further, these sections of the tank are 1) subjected to cyclic loading, and 2) the fatigue life of material subjected to previous plastic strains is generally quite low. As such, the area of the tank most prone to failure, through a combination of stress relaxation cracking and low cycle fatigue, is the top of the buckles near the center of the tank. It can be noted that the cracks seen to date in the floors of commercial tanks have occurred at this location.

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5.3 *Consequences of a Leak in the Floor*

Salt has a low surface tension, and as such, it is an effective wetting agent. Expanded clay is an effective thermal insulation due to its large number of extremely small internal cavities. These material characteristics lead to a situation in which salt, when in contact with clay, fills the internal cavities with salt. The saturation process requires a period of days to weeks; however, once the clay is saturated, there is no practical mechanism for removing the salt from the clay.

5.3.1 Heat Losses

The thermal conductivity of salt ($0.55 \text{ W/m-}^{\circ}\text{C}$) is about 2.5 times the thermal conductivity of expanded clay ($0.22 \text{ W/m-}^{\circ}\text{C}$). As such, exposing the clay to salt increases the thermal conductivity of the clay, with the increase a function of the degree of saturation. However, there are little, and often conflicting, data on the increase. A representative value for the increase is on the order of 1.5.

If exposure of the expanded clay to salt increases in the thermal conductivity of the foundation insulation by a factor of 1.5, then the heat transfer through the foundation will increase by a similar factor. As a consequence, the temperature of the soil can rise above the design value of 75°C . To limit the possibility of unpredictable tank settlements, the capacity of the foundation cooling fans will need to be increased. If higher fan capacities are insufficient to maintain soil temperatures below 75°C , air chillers can be added at the inlet to the cooling fans.

5.3.2 Floor Stresses

As discussed in Section 5.7, even moderate radial temperature gradients in the floor can produce stresses at, or above, the allowable values in Section II of the ASME Code.

Should the tank develop a leak, the heat flux into the foundation will increase. An increase in the heat flux will change the radial temperature gradient in the floor. This, in turn, has the potential to either decrease or increase the radial and tangential stresses in the floor. Further, the distribution of the salt in the foundation may be random and change over time. As a result, the heat flux into the foundation, the temperature distribution in the floor, and the stress distribution in the floor could also be random and change over time. As such, the potential exists for new, and unexpected, radial and circumferential temperature distributions to develop in the floor.

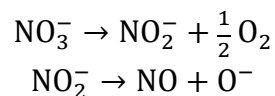
In principle, an estimate could be made of the changes in the stress distribution in the floor due to a leak if the radial and the circumferential temperature distributions in the floor were known. However, there are typically a limited number of thermocouples installed on the floor; perhaps 8 to 12. This number of temperature readings may not be enough to accurately predict a stress distribution in the floor. If so, it may not be possible to estimate the potential for plastic collapse, or the remaining creep and fatigue life,

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of the floor. The Owner may decide that the risks associated with this situation are not acceptable. Unfortunately, the only solution is to remove the foundation material exposed to salt, and replace with foundation with new material. However, due to the size of the tank, replacing the foundation requires that the floor be removed, the tank elevated by means of a jacking system, the foundation excavated, and the floor replaced with a new floor.

5.3.3 NO_x Production

A more significant problem is the chemical reaction of expanded clay with salt. Nitrate salt is an oxidizing agent. The expanded clay, during its production, is fired at temperatures high enough to form an oxide. However, one oxide species can react with an oxidizing agent to form another oxide species. At the temperatures seen in the foundation of a hot tank, clay can undergo the second oxidation step. The oxidation is accompanied by a series of reduction reaction in the salt, as follows:



The NO leaves the clay in the form of a gas, and, upon exposure to the atmosphere, can react with oxygen to form NO₂.

The oxidation / reduction reactions are not particularly fast. However, due to the large quantities of clay in a tank foundation, and due to the large surface-to-volume ratios inside the clay structure, the NO_x emitted from a foundation can be on the order of kg/day. Emission rates this high can present a health hazard to maintenance personnel, and may cause the project to exceed allowable emissions. Further, the NO_x production should, in principle, continue until either all of the clay has been oxidized to its new oxidation state, or all of the salt has been reduced to NO_x. Given the dimensions of the foundation in a commercial plant, this process could involve decades.

5.3.4 Leak Detection System

Identification of Leak Location

At the Solar Two demonstration project, the perimeter of the foundation used hard refractory bricks to support the concentrated load of the roof and the wall. However, the strength of the selected refractory would be reduced if the bricks were exposed to nitrate salt. To help ensure the integrity of the foundation following a leak, a continuous metal sheet was installed between the solid sand lubricant and the refractory. A leak would percolate horizontally through the sand layer, and then fall onto the soil outside of the foundation.

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As might be imagined, a leak would percolate through the sand in essentially all directions, and the point on the soil where the leak accumulated may not be a good indication of the radial location of the leak. To help identify the source of the leak, an array of collection troughs or channels could be located at the top of the foundation.

One potential arrangement is an array of orthogonal channels. The channels would slope down from the center of the foundation to the perimeter; i.e., similar to that of the floor. A leak would flow primarily into 2 channels, and cross-correlating the positions on the perimeter where the leak appeared would provide an indication of the leak in the floor.

A second potential arrangement is an array of radial and tangential channels. A leak entering a radial channel would be directed to the perimeter of the foundation. However, some portion of the leak could enter a tangential channel, and then flow to one or more of the associated radial channels. This collection arrangement would provide information on the circumferential location of the leak, but only limited information on the radial location.

It should be noted that the effectiveness of the leak detection system requires that the foundation directly beneath the floor be as impervious to salt as possible. Specifically, if the material beneath the floor is pervious (i.e., sand), then the leak would flow horizontally as well vertically, and the leak would not be confined to only one or two channels. This, in turn, implies the following:

- The material at the top of the foundation is either a metal slide plate or a hard refractory
- If the material is a refractory, then the selected material must be chemically and structurally stable if exposed to salt. In addition, the coefficient of friction between the floor and the refractory will be different, and perhaps higher, than the coefficient of friction between steel and sand
- If the material is a metal plate, then the following applies:
 - The selected plate material must be resistant to galling with the stainless steel floor in the hot tank
 - A common approach to welding the floor is to tack weld a backing strip along the bottom edge of a plate, locate the adjacent plate, and perform a root pass in the space defined by the edges of the adjacent plates and the backing plate. The backing plate will be lower than the bottom of the floor; however, space for the backing plate is provided by movement of the sand beneath the floor. In contrast, with a metal plate beneath the floor, backing strips can no longer be used. This may require modifications to the floor welding procedure to ensure that the root pass only fuses with the floor plates and not with the plate beneath the tank
 - The floor, after welding, will have depressed sections at the weld seams and elevated sections near the centers of the plates. This will produce non-uniform contact between

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- the floor and the slide plate, which will produce 1) regions of locally high friction forces, and 2) friction forces which vary radially and tangentially in a non-uniform manner. To what extent this influences the stresses in the floor, and the consequent effect on the creep lifetime, will require FEA analyses
- After the tank enters commercial service, the residual welding stresses in the floor will start to relax. This will change the dimensions of the depressed and the elevated sections, which will change the distribution of the friction forces between the floor and the sliding plate.

Isolation of Foundation Insulation

Even if the collection channels can provide only a qualitative location of a leak, the collection equipment provides an important secondary function. Specifically, some fraction, and perhaps a majority, of the leak will be captured by the collection system. This will reduce the net flow rate of salt into the foundation insulation, which will 1) provide the operators with additional time to drain the tank, and 2) reduce the increase in the heat losses through the foundation due to adding salt (with a relatively high conductivity) to the foundation insulation (with a relatively low conductivity).

5.4 Foundation Perimeter Stiffness

The vertical loads on the foundation are a function of the radial location in the foundation. The highest loads occur at the perimeter of the tank, where the weight of the wall and the weight of the roof must be supported.

At the Solar Two demonstration project, the nominal diameters of the storage tanks were 12 m. The perimeters of the tank foundations were constructed by stacking, and mortaring, a group of refractory bricks. The bricks formed a ring wall, which carried the vertical loads of the wall and the roof down into the concrete mat foundation below. During the 3-year life of the project, no differential or unexpected settlements in either the hot tank or the cold tank were observed. A postmortem examination of the inside of both tanks showed no unexpected deformations of the floors or the walls ¹⁴.

Not too surprisingly, the construction of a ring wall using refractory bricks is a labor intensive exercise. Further, refractory bricks are expensive relative to other potential foundation materials. Commercial projects, both parabolic trough and central receiver, after the Solar Two project pursued less expensive approaches to the foundation design. A common approach used expanded clay, such as Utelite, as the primary foundation material under the entire tank. The expanded clay both supported the tank and provided the primary thermal insulation under the floor.

¹⁴ Pacheco, James (Sandia National Laboratories, Albuquerque, New Mexico), 'Final Test and Evaluation Results from the Solar Two Project', Sandia report SAND2002-0120, January 2002

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Allowable stress values for the expanded clay showed that the material could safely support the vertical loads at the tank perimeter due to the weight of the wall and the roof. However, during tank fabrication at one commercial project, depressions in the foundation were observed at the perimeter of the tanks even before the tanks were filled with salt. At another commercial project, depressions in the perimeter foundation were observed after the tanks were filled with salt, and then exposed to a number of thermal expansions and contractions. The depressions were produced by the movement of clay particles relative to one other, and the inability of the particles to remain in fixed positions over an extended period of time when exposed to cyclic loads. An hypothesis for this effect is based on the following:

- Finite element calculations of the foundation show that compressive loads at the perimeter of the tank can safely be accommodated by the foundation insulation. However, at a distance slightly beyond the edge of the floor, the compression loads drop to zero
- The foundation, by definition, does not move. As such, compressive forces in the foundation at the edge of the floor must be balanced by tensile forces in the foundation just outside the edge of the floor. The tensile forces are established by friction between adjacent particles in the expanded clay. However, the tank fills and drains daily, which results in a pattern of cyclic compressive loads on the foundations. This, in turn, results in cyclic tensile loads on the expanded clay. However, the clay particles are not bonded to one another, and cyclic loadings can result in particles moving relative to one another. The path of least resistance is a movement of particles out, up, and away from the zone immediately below the edge of the tank.
- The movement of the clay particles leaves the perimeter of the floor unsupported. The floor is not designed to withstand the bending loads generated by the weight of the wall and the roof, and the floor plastically deforms down in response.

The plastic deformations at the perimeter of the floor produce two undesirable effects:

- The high strains associated with plastic deformations significantly reduce the low cycle fatigue life of the floor at this location
- As the tank expands and contracts, the curved perimeter of the tank tends to push the expanded clay, rather than slide over the expanded clay. This increases the depth of the depression, which 1) increases the effective coefficient of friction between the floor and the foundation, and 2) promotes further plastic deformation of the annular floor plate.

In general, the use of expanded clay as the foundation material near the perimeter of the tank has been shown to not be an acceptable approach. Some form of a rigid material, such as refractory bricks or refractory castings, at the perimeter is required.

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For existing tanks with clay foundations, inserting metal plates, between the bottom of the tank and the top of the clay, at the tank perimeter has been adopted as a corrective measure. However, the long term effectiveness of this approach is something of an unknown.

5.5 Floor Welding

In commercial salt tanks, the floor is typically fabricated from large rectangular plates. The plates are arranged on the foundation, and then butt welded along each face.

The plates, as supplied by the mill, are not all exactly the same dimensions and are not all exactly the same shape. This results in a variation in the gaps between plates. To assist in the welding process, weld backing strips are attached to the bottom of alternating plates. The backing strips also help to control the gas chemistry in the weld region.

The floor plates have a thickness in the range of 6 to 11 mm, depending on the tank vendor. The number of welding passes ranges from 1 to 3.

5.5.1 Residual Stresses

For a floor plate weld requiring 2 passes, the residual welding stresses are a function of the weld length, as illustrated by the red line in Figure 5-1 ¹⁵. Residual stresses in the range of 300 to 700 MPa are well above the yield stress, which leads to plastic strains in the vicinity of the weld ¹⁶.

¹⁵ Osorio, J. (National Renewable Energy Laboratory, Golden, Colorado), Failure Analysis for Molten Salt Thermal Energy Tanks for In-Service CSP Plants, DOE Award Number DE-EE00038475

¹⁶ Huang, Hui, (Oak Ridge National Laboratory, Oak Ridge, Tennessee), 'Finite Element Analysis and In-Situ Measurement of Out-of-Plane Distortion in Thin Plate TIG Welding', UT-Battelle, LLC, under DOE Contract DE-AC05-00OR22725, January 2019

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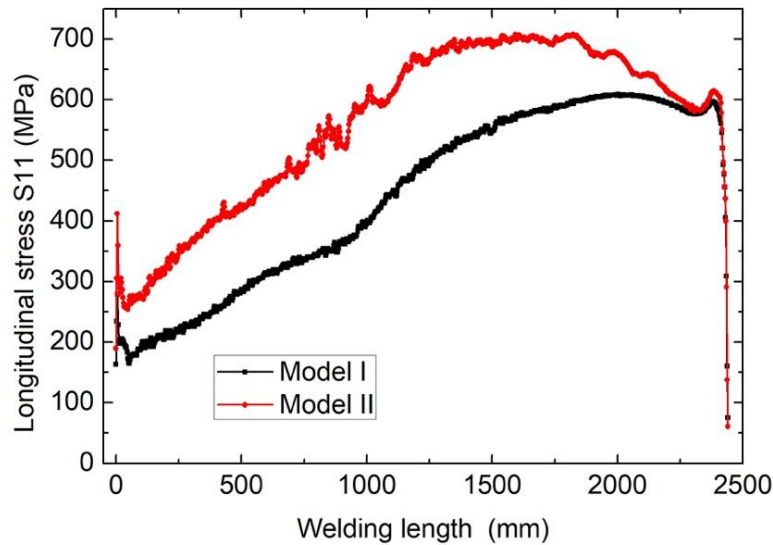


Figure 5-1 Longitudinal Residual Welding Stress

As the weld cools, the metal shrinks, which establishes tensile stress at the top of the weld. This is matched by compressive stresses at the bottom of the weld.

5.5.2 Plate Deformations

The deformation associated with the residual stresses for a subset of the floor welds is illustrated in Figure 5-2. The view shows the edge of two adjacent plates, with the weld located at the mid-point of the curve; i.e., at the lowest point. The overall displacement between the ends of the two plates and the low point at the weld zone is 86 mm.

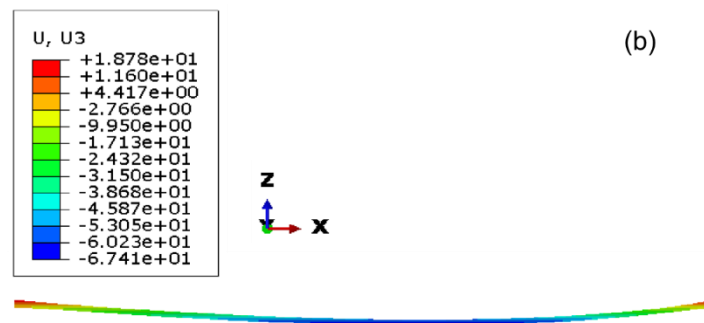


Figure 5-2 Plate Deformations due to Residual Welding Stresses

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When all of the welding for the floor plates is completed, each plate takes on the shape of the top of a shallow sphere with a large radius. The low points in the floor are along the weld seams, and the high points in the floor are the centers of the plates.

It can be noted that the floor, after welding, is no longer flat; i.e., the floor has, in effect, buckled at each of the plate seams. This has important implications for the stability of the floor once the tank has been commissioned and is placed into commercial service.

5.5.3 Stress Relaxation

Stresses in the weld region, either due to welding or due to plate deformations, have values at or above the yield stress. This would normally result in creep lifetimes on the order of 2 to 3 years. However, the stresses do not persist due to stress relief. The effect is illustrated in Table 5-1 for 300-series stainless steels ¹⁷.

Table 5-1 Stress Relief as a Function of Temperature

Temperature, °C	Stress relief, percent
200 to 400	40 (peak)
550 to 650	35
850 to 900	55
950 to 1,050	95

Several months after the start of commercial service, the residual stresses have relaxed to a steady state value of about 125 MPa, as described in Section 5.15.8. This stress is permanent, and other stresses due to friction, radial temperature gradients, and transient thermal stresses can add to, or subtract, from this value.

It can be noted that stainless steels typically have sufficient uniaxial ductility to accommodate the plastic strains associated with the level of stress relaxation above. However, if the material is constrained, then strains can develop in multiple axes of the heat affected zones. Stainless steels with a high carbon content have a multiaxial failure ductility as low as 1 percent. The creep strain that occurs during the relaxation of the weld residual stresses could be sufficiently large to exceed the multiaxial failure ductility, and cracks could develop in the weld zone.

To a first order, the potential to damage the tank is perhaps the largest at the start of commercial service. As the weld and the plate stresses relax over the first 100 hours or so of operation at the design

¹⁷ Peckner, D. and Bernstein, I., Handbook of Stainless Steel, Chapter 4, Table 25, ISBN 0-07049147-X, 1977

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temperature, the residual stresses approach a lower, steady state value. To this end, it may be beneficial to heat treat the tank, at a nominal temperature of at least 550 °C for 100 to 200 hours, to remove some fraction of the residual stresses.

On a related point, the salt inventory at the Solar Two project was melted and then placed in the hot tank. In the hot tank, the inventory was heated to 540 °C over a period of 10 days, and then maintained at that temperature for 20 days. The purpose of the heating phase was to promote the thermal decomposition of the magnesium nitrate in the salt to magnesium oxide and various nitrogen oxides; i.e.,



The decomposition process, which was a one-time event, released the NO_x gas in the hot tank. This avoided potential problems with releasing the gas in the receiver, which may have had a detrimental effect on the heat transfer rates. Interestingly, the 20-day decomposition period may have had an inadvertent, but beneficial, effect of relaxing a portion of the residual welding stresses prior to the start of the test and evaluation program. In new commercial projects, it may be worthwhile to give some consideration to a similar process in which the hot tank is heated to a temperature close to the design temperature, and then held at that temperature for a period which is long enough to relax some portion of the stresses associated with fabrication, preheating, and the initial filling.

5.5.4 Post Weld Heat Treatment

One method for reducing residual welding stresses is a post weld heat treatment. A representative process is described in Section 8.3.2. The heat treatment process requires accurate control over heating rates, hold times, and cooling rates to achieve the desired combination of metal chemistry and stress reduction.

There are specialty contractors who provide heat treating services for large field equipment, such as the wall of a salt tank. However, the floor of the tank is more problematic. Specifically, 1) the floor can only be heated from one side, and 2) the floor is in contact with the foundation. The floor is relatively thin, and achieving the required heating rates and hold times should be feasible. However, with a hold time on the order of 1 hour, the temperature of the foundation directly beneath the weld will increase to a value approaching the treatment temperature. During the cooling period, the foundation will return some of this heat back to the weld, and it may not be possible to achieve the required cooling rate.

Further, should the tank develop a leak, the foundation will absorb some of the salt. Repairs could be made to the floor by draining the tank, removing the damaged sections, and welding in new sections. If the original floor required a post weld heat treatment, then the replacement sections should also receive a post weld heat treatment. However, the heat treatment temperatures are high enough to thermally decompose the salt absorbed by the foundation. The decomposition process releases NO_x, which can be a hazard to plant personnel, and the decomposition also generates a range of oxide species. The oxides

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will produce very high corrosion rates, both during the heat treatment period and after the tank returns to commercial service.

In summary, it may not be feasible to conduct a post weld heat treatment on the tank floor, either during construction or following a repair. As such, the creep and fatigue life assessment of the floor will need to be based on the following:

- The metal properties and the metal chemistry at the boundaries of the plates in the as-welded condition
- A limited relaxation of residual stresses due to metal temperatures no higher than the design tank temperature.

5.6 Hydraulic Tests

A common commercial plate dimension is 2.44 m by 9.75 m (8 ft. by 32 ft.). A representative tank diameter in a commercial project is 42 m. As such, a nominal 13 rows of plates are required to span the diameter of the tank.

As discussed in Section 5.5, the plates in the floor are forced into the shape of the top of a sphere with a large radius. The deformations noted in Figure 5-2 are consistent with evidence from tanks in commercial projects which show deformation at the center of the plate on the order of 50 to 100 mm.

5.6.1 Plate Stresses

When the tank is filled with water for the hydraulic tests, hydrostatic loads on the plates push the plates down toward a flat shape. The deformation at the center of a flat plate, with a uniform loading and fixed at the edges, is given by the following equation from Roark:

$$y_{max} = \frac{\alpha q b^4}{E t^3}$$

where σ is a coefficient based on the length-to-width ratio of the plate [] (i.e., dimensionless), q is the pressure loading [MPa], b is the width of the plate [m], E is the modulus of elasticity [MPa], and t is the thickness of the plate [m]. The floor plates in commercial projects have a representative length of 10.0 m, a width of 2.50 m, and a thickness of 0.00715 m. The length-to-width ratio is 4, which results in a value for σ of 0.0284.

To reach a displacement of 100 mm for y_{max} , the required static pressure is 6,800 Pa, which corresponds to a water depth of 0.8 m. As such, soon after the tank begins to fill, the floor goes from a domed shape to a flat shape.

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The peak stress in the plate occurs at the center of the long edge, and is given by the following equation from Roark:

$$\sigma_{max} = \frac{\beta q b^2}{t^2}$$

where β is a coefficient based on the length-to-width ratio of the plate [], q is the pressure loading calculated in the equation above [0.0068 MPa], b is the width of the plate [m], and t is the thickness of the plate [m]. For a plate with a length-to-width ratio of 4, β has a value of 0.5, and the peak stress is calculated to be 416 MPa. This value exceeds the yield stress at ambient temperature, which means that yielding occurs in a section of the weld region near the center of the plate, which relieves some portion of the imposed stress.

When the plate changes shape from domed to flat, the stress in the center of the plate can be estimated using the following equation from Roark:

$$\sigma_{max} = -\frac{\beta q b^2}{t^2}$$

where β is a coefficient based on the length-to-width ratio of the plate [], q is the pressure loading calculated in the equation above [0.0068 MPa], b is the width of the plate [m], and t is the thickness of the plate [m]. For a plate with a length-to-width ratio of 4, β has a value of 0.25, and the compressive stress is estimated to be 208 MPa.

5.6.2 Plate Deformations

If the center of the plate is raised by 100 mm, then the width of each plate is reduced by about 4 mm. When the tank is filled, hydrostatic loads push down on the plates, and change the shape of the plates from curved to flat. As such, the width of each plate will increase by 4 mm. Across the diameter of the tank, the total change in the width of the plates is $13 * 4 \text{ mm} = 52 \text{ mm}$.

The radial stiffness of the wall is several orders of magnitude greater than the radial stiffness of the floor, particularly if the plates have already buckled due to welding. As such, the 52 mm increase in plate dimensions is converted into some combination of elastic and inelastic deformations. The deformations will be determined, in part, by the residual stresses produced during welding. Since the residual stress distribution is difficult to predict, and somewhat random, the additional plate deformations will also be difficult to predict and somewhat random.

When the water is removed at the end of the hydraulic tests, the hydrostatic loads are also removed. Residual stresses in the plates will cause the plate to return toward the domed shapes produced during

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the welding process. However, any plastic deformations produced in the plates when the tank was full will result in a different floor shape after the hydraulic tests than before the hydraulic tests.

To support a finite element calculation of the stresses in the floor, measurements can be taken of the floor shape before and after the hydraulic tests.

5.7 Radial Temperature Distributions in the Floor

Commercial salt storage tanks are placed on a foundation, which typically consists of the following elements:

- A concrete base mat, which is the lowest level of the foundation. A series of parallel cooling air pipes is located in the mat, and a flow of air is forced through the pipes by means of a fan. The air flow removes the majority of the heat flux (> 90 percent) passing from the tank into the foundation. The heat removal from the base mat maintains the soil temperature beneath the base mat to a maximum temperature of about 75 °C. This temperature ensures that the desiccation of the soil, and the oxidation of organic materials in the soil, do not reach levels which could cause settlement of the tank.
- A primary insulating layer, beneath the majority of the floor. The insulating material can be an expanded clay, such as Utelite[®], or an expanded glass, such as FoamGlas[®].
- A ring wall, composed of a hard refractory material, at the perimeter of the tank. The refractory provides the foundation stiffness needed to support the concentrated weight of the wall and the roof.
- A solid lubricant, such as sand, on top of the primary insulating layer and the refractory ring wall.

Near the center of the tank, conduction heat transfer from the floor into the foundation is primarily one-dimensional; i.e., straight down. The thermal resistance to conduction heat transfer is largely the thermal resistance of the primary insulating material. The overall thermal resistance in this direction is 'large'. Conversely, near the perimeter of the tank, conduction heat transfer from the floor is a combination of vertical heat transfer through the refractory material and horizontal heat transfer through the soil surrounding the refractory ring wall. The thermal conductivity of many refractories is about double the thermal conductivity of expanded clay, and the thermal conductivity of soil is a factor of 4 to 5 times the thermal conductivity of expanded clay. As such, the heat flux from the floor to the environment is a function of the radial position in the floor, as illustrated in Figure 5-3.

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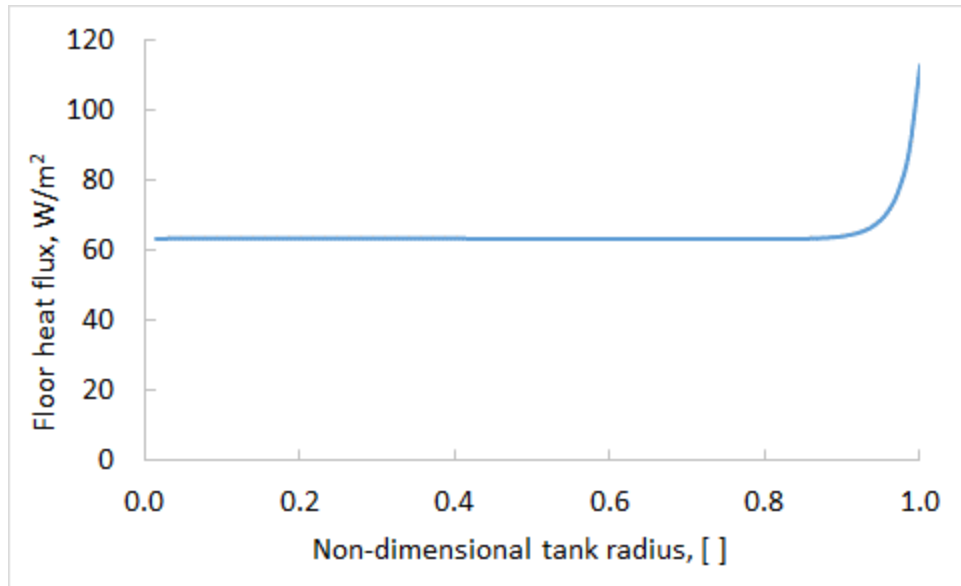


Figure 5-3 Radial Distribution of Heat Flux from the Floor in a Hot Salt Tank

Due to mixing within the tank inventory, the temperature of the inventory is largely isotropic. As such, if the heat flux from the floor to the foundation is a function of the radial position, then the temperature of the floor must also be a function of the radial position.

A two-dimensional, steady-state conduction heat transfer model of the foundation was developed to explore this effect. The model is simplified by simulating the convection heat transfer within the inventory as follows:

- Case 1 - The convection heat transfer is simulated by multiplying the thermal conductivity of the salt by a factor of 100
- Case 2 - The convection heat transfer is simulated by multiplying the thermal conductivity of the salt by a factor of 10
- Case 3 - The convection heat transfer is simulated by multiplying the thermal conductivity of the salt by a factor of 1. This represents a case in which the salt is completely stagnant. Such a condition can occur when 1) the temperature of the inventory is, say, 500 °C, 2) the plant starts an extended shutdown, and 3) the electric salt heaters do not start operation until the temperature of the inventory has decayed to, say, 300 °C.

In each of the three Cases, the depth of the salt inventory is taken to be 3.5 m, and the temperature at the top of the inventory is assumed to be a constant value of 560 °C.

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In Case 1, the radial temperature gradient in the floor (blue line) is illustrated in Figure 5-4. The radial gradient is a modest 4 °C.

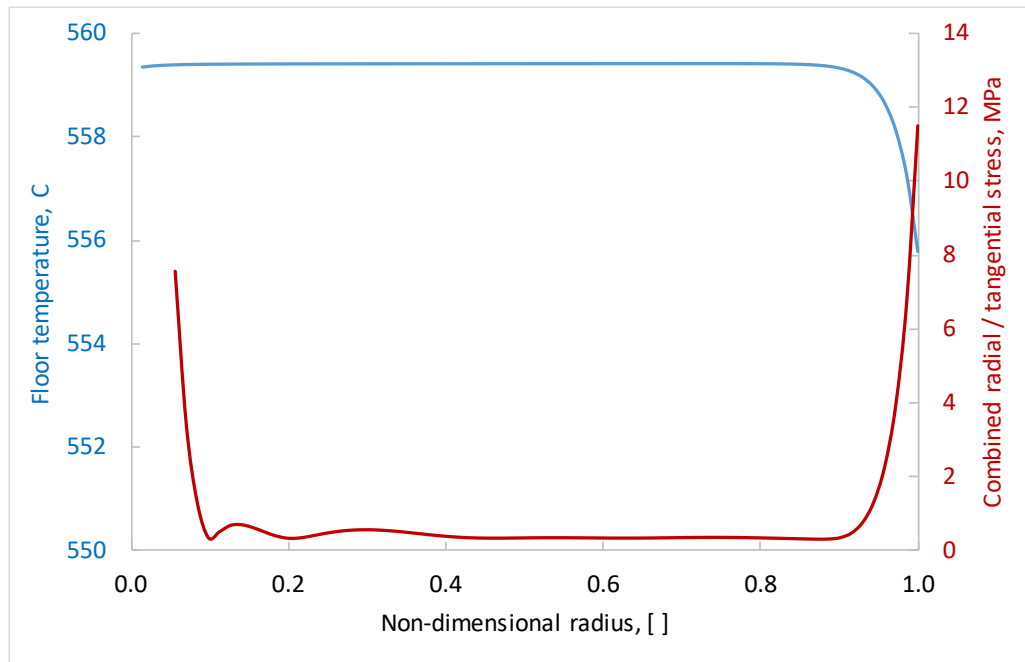


Figure 5-4 Case 1 - Radial Distribution of Floor Temperature and Floor Stress

For an unconstrained circular plate, subject to a radial temperature gradient, the radial stress and the tangential stress are given by the following formula in Roark ¹⁸:

$$\sigma_r = \gamma E \left(\frac{1}{R^2} \int_0^R T r \, dr - \frac{1}{r^2} \int_0^r T r \, dr \right)$$

$$\sigma_t = \gamma E \left(-T + \frac{1}{R^2} \int_0^R T r \, dr - \frac{1}{r^2} \int_0^r T r \, dr \right)$$

where γ is the coefficient of thermal expansion (1/°C), E is the modulus of elasticity (kg/m-sec²), R is the radius of the disc (m), and T is the temperature at any point a distance r from the center minus the temperature of the coldest part of the disc (°C). The radial and the tangential stresses are combined using the following formula:

$$\sigma_{Combined} = \sqrt{\sigma_r^2 - \sigma_r * \sigma_t + \sigma_t^2}$$

¹⁸ Young, W. C., and Budynas, R. G., 'Roark's Formulas for Stress and Strain', Seventh Addition, 2002

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In Case 1, with the convection modeled as 100 times the thermal conductivity of the salt, the peak stress in the floor is 12 MPa and it occurs at the perimeter (the calculated stress at the center is infinite, as the equation involves a division by zero). This value is approximately 11 percent of the allowable stress listed in Section II of the ASME Code for Type 347H stainless steel at 560 °C, and approximately 10 percent of the allowable stress associated with a creep lifetime of 300,000 hours as defined in API 579-1 / ASME FFS-1 (see Section 5.15.3).

Repeating the calculations for Case 2, in which the convection is modeled as 10 times the thermal conductivity of the salt, produces the temperature and the stress curves shown in Figure 5-5.

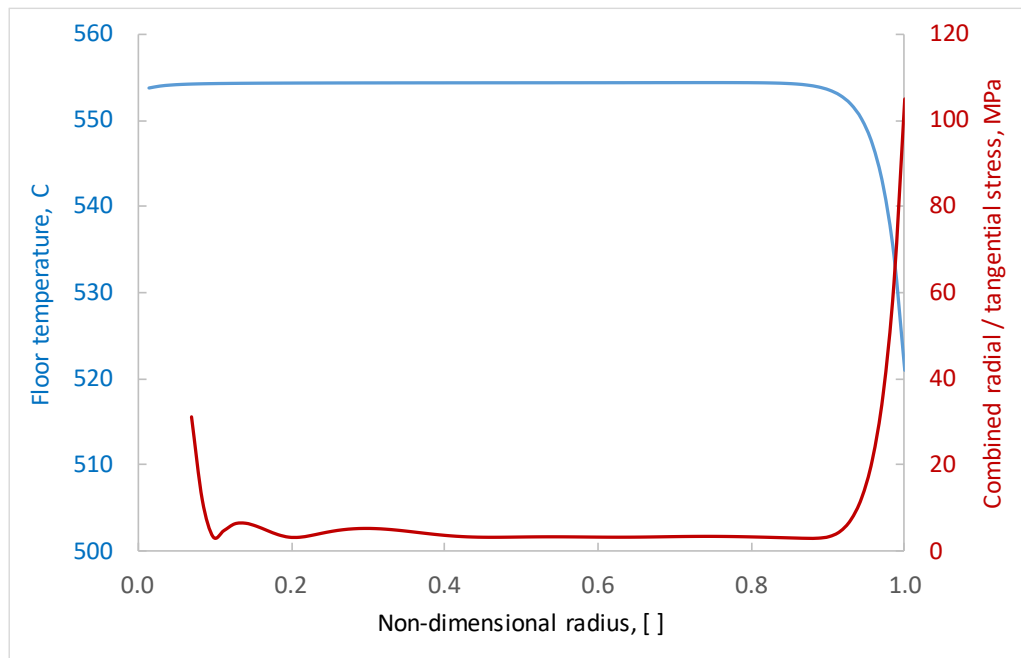


Figure 5-5 Case 2 - Radial Distribution of Floor Temperature and Floor Stress

The radial temperature gradient has increased to 33 °C, and the peak stress has increased to 107 MPa. This value is nominally equal to the ASME allowable stress, and approximately 85 percent of the allowable stress associated with a creep lifetime of 300,000 hours. Informal discussions with other organizations developing finite element models of the floor have reached similar results.

Repeating the calculations for Case 3, in which the inventory is assumed to be stagnant, produces the temperature and the stress curves shown in Figure 5-6. Due to the relatively low thermal conductivity of salt, a radial temperature gradient on the order of 175 °C is established.

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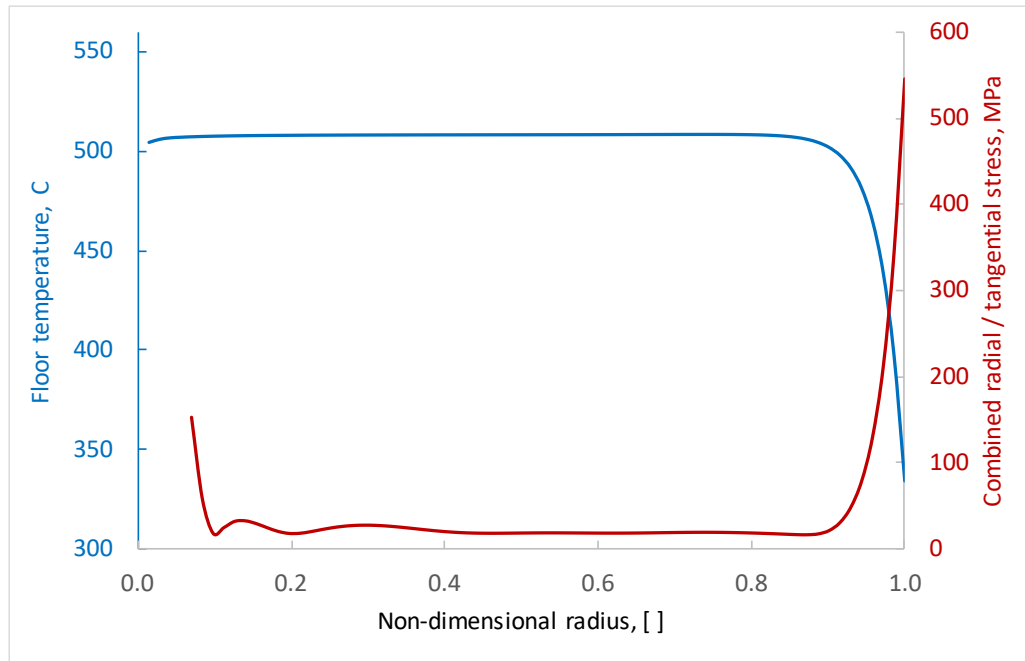


Figure 5-6 Case 3 - Radial Distribution of Floor Temperature and Floor Stress

Clearly, Case 3 represents something of a worst case condition, as 1) the time required to develop the steady-state temperature profile shown in Figure 5-6 could be several weeks, and 2) the operators are likely to recognize the problem, and establish some form of circulation in the tank, before salt temperatures reach values this low. Nonetheless, the calculated temperature profile indicates the potential for two significant problems:

1. The calculated stresses at the perimeter are the same order of magnitude as the tensile strength, and rupture of the floor cannot be discounted.
2. The temperature of the metal at the perimeter of the tank (330 °C) is perhaps 100 °C above the freezing point of the salt. Nonetheless, there is the potential for salt to freeze 1) if there are local defects in the insulation, 2) if the insulation has become wetted by salt due to leakage from the pumps, valves, and instruments above the tank, or 3) if water has infiltrated the insulation due, for example, to heavy rains. If frozen salt develops in the floor near the perimeter of the tank, then the salt will be thawed by recirculating the inventory within the tank. In the regions beneath the frozen salt, the heat flux into the metal will be defined by the thermal conductivity of the solid salt. However, in the regions adjacent to the frozen salt, the heat flux into the metal will be defined by the convection heat transfer coefficients of the circulating salt. The two heat fluxes

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could differ by an order of magnitude, which, in turn, could produce local temperature gradients large enough to damage the tank. Unfortunately, the freeze / thaw progression would be largely random and beyond the control of the project. As such, it may be impossible to assess the potential damage to the tank from a freeze / thaw cycle.

Clearly, the steady-state model only mimics the effect of convection within the salt inventory. Nonetheless, the calculations show that a radial temperature gradient about 35 °C establishes stresses equal to the ASME allowable value, and a gradient of approximately 50 °C produces stresses at the yield value. Further, should the inventory becomes stagnant for even a moderate period of time (i.e., days), then permanent damage to the floor may be inevitable.

Should the floor plastically deform due to a radial temperature gradient, and the gradient is then removed, the stress distribution in the floor will be a combination of the new temperature gradient, plus a hysteresis effect from the plastic deformation. Further, the magnitude of the hysteresis effect will be difficult to both calculate and verify.

5.8 Tank Preheating

Prior to filling the tank with salt, the tank and the top of the foundation are preheated to a nominal temperature of 320 °C to prevent a thermal shock to the tank. The heating medium is hot air from a propane- or a natural gas-fired direct air heater. The hot air is directed through a manway in the roof, flows down a short (2 m) temporary gas duct suspended from the roof, enters the tank with the flow directed nominally toward the center, and then exits through a second manway in the roof.

The preheating period ranges from 7 to 20 days. The allowable rate of temperature change is limited; values in the range of 1.5 to 2 °C/hr are typical. Limits are also placed on the intra-tank temperature differential; however, the allowable differentials are relatively high. In one commercial project, differences in temperature as high as 33 °C between the center of the tank and any wall location are permitted.

The preheating process is one in which damage could occur to the tank, as follows:

- Due to non-uniform heat losses from the tank, radial temperature gradients of 25+ °C can be established in the floor
- The flow of preheating air enters the tank at only one location, and the air flow path is dictated by the location and the shape of the inlet duct. As such, the air flow path is known on a gross basis, but is largely beyond the control of the operators. Further, air at high temperatures has a low density, and is generally a poor mechanism for heat transfer. The combination of inexact control over the air flow and the low heat transfer coefficients can result in unavoidable non-uniform temperature distributions in the tank

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- The allowable measured intra-tank temperature differentials are the same order of magnitude as the radial temperature gradient (35 °C in Case 2 above). A radial gradient of 35 °C, particularly in combination with the friction forces between the floor and the foundation, has the potential to plastically deform the floor
- The thermocouples located in the tank may not be in the exact locations required to measure the maximum intra-tank temperature differentials. As such, the measured maximum differential might be 33 °C, but the actual maximum differential could be in excess of 33 °C. (One method to avoid this limitation would be to monitor the temperature of the floor and the wall using an infrared thermometer. The measuring portions of the instrument will need to tolerate temperatures as high as 400 °C.)

The floor, at the start of the preheating phase, has already deformed due to the welding processes. During preheating, a radial temperature gradient in the floor will develop, with the highest temperature near the center and the lowest temperature at the perimeter. If the gradient exceeds some threshold value, probably on the order of 35 to 50 °C, then compression stresses near the center of the tank can exceed the yield value. The buckling resistance of the floor will be very low due to 1) the preexisting plate deformations, and 2) the absence of stabilizing hydrostatic forces from the salt. As such, the floor could develop permanent buckles (ridges) even prior to the start of commercial service. As with every metal, the stresses associated with plastic deformation will have a significant detrimental effect on the creep life and the low cycle fatigue life.

As noted above in Section 5.2, the characteristics of the buckled regions (length, height, width, residual stress, and residual strain) are functions of the welding deformations and residual stresses. These characteristics will have some random features, and it is difficult to develop a generic analysis which applies to all buckled regions. However, the creep-fatigue life of a buckled region is always expected to be less than the 30-year life of the project.

5.9 Tank Filling

Following the completion of the tank preheating, the shape of the floor will consist of a combination of domed sections (generated at the end of the hydraulic tests) and perhaps some buckled regions. The salt melting process begins, as described in Section 9.4, Nitrate Salt Handling and Melting Specification, in Volume 1 - Specifications for Parabolic Trough Projects. As the tank is filled with salt, hydrostatic loads will push the floor plates down against the foundation for a second time. The required hydrostatic pressure is on the order of 7 kPa, which corresponds to a salt depth of about 0.4 m. Since the minimum salt level in the tank is 1.0 m during commercial operation, the floor remains flat against the foundation for the life of the tank.

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Changing the shape of the plates from domed to flat will generate stresses at both the edges and at the center of the plates. The deflection of the plates will depend on the shape of the floor at the completion of both the hydrostatic tests and the preheating. However, the shape of the floor at the end of preheating may not be known, or it may be difficult to measure. As such, estimating the stresses at the edges of the plates, and at the centers of the plates, after the fill process is likely to involve large uncertainties. Nonetheless, some portions of the plate edge and the plate center stresses will remain as permanent stresses. Under various conditions, the permanent stresses could add to, or subtract from, other stresses generated by friction, radial temperature gradients, and transient temperature conditions.

From a review of the plate deformations, two observations can be made:

- The vertical dimensions of the buckles seen in commercial tanks are larger than the cumulative changes in the plate widths discussed above. However, there are a number of mechanisms, discussed in the Sections below, which can produce ratcheting, the starting point of which are the buckles produced during filling (and likely preheating)
- Buckles, once produced, are permanent. Specifically, there are only 2 mechanisms that could, in principle, remove a buckle:
 - Hydrostatic forces. The maximum hydrostatic load of approximately 250 kPa occurs when the tank is full. However, the stiffness of a triangular, or a semi-circular, buckle is such that a load of 250 kPa produces vertical deflections in the buckle that are much smaller than the height of the buckle
 - Inverted radial temperature gradients. If the floor is placed in tension, the tensile forces will pull on the sides of the buckles, which causes the height of a buckle to decrease. However, for the buckles to permanently deform, the tensile stresses on the buckle must exceed the yield value. This can occur if the temperature of the floor at the perimeter of the tank is 40 °C, or more, above the temperature of the floor at the center of the tank. However, the heat loss pattern in the tank, plus the flow distribution inside the tank, will not allow this condition to occur. Further, to remove the buckle, the height of the buckle would need to be converted into a linear, plastic, compression of the floor. But it is not possible to simultaneously subject local portions of the floor to compression stresses and other portions of the floor (i.e., the buckles) to tensile stresses.

5.10 Friction Between the Floor and the Foundation

5.10.1 Daily Thermal Cycles

Over the course of a typical operating day, the hot tank undergoes the following cycle:

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- In the afternoon, the inventory level and the inventory temperature reach their maximum values
- Early in the evening, the inventory level reaches its minimum value. Heat losses from the tank cause the temperature of the inventory to decay.
- Early in the morning, the receiver is started. Over the next 20 minutes or so, both the flow rate and the outlet temperature from the receiver increase. At a pre-defined downcomer temperature, salt flow is directed to the hot tank. The crossover temperature is on the order of 30 to 60 °C lower than inventory temperature. The goal is to capture as much useful thermal energy from the receiver as possible for electric power production. Note that this phase of the receiver startup results in a decay in the inventory temperature.
- Once the receiver reaches its normal design temperature, the temperature of the hot tank inventory starts to increase.
- The continuous heat losses from the tank, in combination with a maximum salt inlet temperature of 565 °C, mean that 1) the temperature of the inventory ranges daily between a minimum value of about 540 °C and a maximum value of about 560 °C, and 2) the inventory temperature never reaches the design receiver outlet temperature of 565 °C.

As such, the tank is subjected to a daily condition, typically in the afternoon, in which the level and the temperature of the inventory both increase. If the temperature of the inventory increases, then the tank will expand. Friction between the tank and the foundation, in turn, places the floor into compression. As discussed in the sections above, the floor has a low buckling resistance due to the plate deformations associated with welding, and due (potentially) to the plate deformations associated with preheating.

5.10.2 Initial Thermal Expansion

As an example of the thermal expansion, a hot salt tank in a representative commercial project has a nominal diameter, at ambient temperature, of 42 m. As the temperature of the tank is increased from ambient conditions to the normal design temperature of 565 °C, the radius of the tank increases by $\delta = R * \gamma * \Delta T = 0.219$ m, where R is the tank radius [m], γ is the coefficient of thermal expansion [1/°K], and ΔT is the change in metal temperature [°K]. Thus, at the design temperature, the radius of the tank is 21.219 m.

5.10.3 Maximum Friction Deflections

As noted above, the tank experiences daily thermal cycles, and the movement of the tank is constrained by friction forces between the floor and the foundation. The maximum amount which the floor can be radially deflected by static friction (i.e., before the tank starts to slide) is given by the following expression:

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$$\delta = \frac{(1 - \nu) \mu \rho H g R^2}{3 E t_p}$$

where δ is the deflection of the floor, m
 ν is Poisson's ratio, []
 μ is the coefficient of friction, []
 ρ is the density of the salt, kg/m³
 H is the depth of the salt, m
 g is the acceleration due to gravity, m/sec²
 R is the outer radius of the tank, m
 E is the modulus of elasticity for stainless steel, MPa
 t_p is the thickness of the floor, m

With a representative maximum salt height of 11.5 m, and assuming that the coefficient of friction is 0.3, then the maximum deflection is 0.0055 m.

Along this line of thought, if the tank is at a condition where the friction loads are zero, and the temperature of the inventory increases, then the tank will expand by an amount equal to $R * \gamma * \Delta T$. If the expansion amount is set equal to the maximum deflection possible due to static friction, then the associated change in inventory temperature is 15 °C. Stated another way, if the tank is full, and if the temperature of the inventory increases by 15 °C, then the maximum friction force will be applied to the foundation.

Operating experience from commercial plants shows that the hot tank experiences multiple cycles each year in which the tank is full, or nearly full, and the temperature of the inventory increases by at least 15 °C. One such situation in which this can occur is as follows:

- The receiver starts in the morning, but the skies are partially cloudy, and the receiver outlet temperature is set to a value less than 565 °C to prevent temperature overshoots during cloud transients
- The steam generator is not started until noon, which means that the inventory level in the hot tank is rising throughout the morning
- In the afternoon, the steam generator demand is close to the receiver output, and the level in the hot tank remains essentially constant
- The weather clears, and the receiver outlet temperature is increased to the design value of 565 °C.

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As such, the floor is exposed to some number of cycles in each year in which the floor starts in a state with a low stress (i.e., close to empty) and then the stress in the floor reaches the maximum value possible due to friction.

5.10.4 Analytical Solution for Floor Stresses due to Friction

The tank expands and contracts due to daily changes in the temperature of the inventory. The friction force, kg-m/sec² (i.e., N), which can be applied by the tank to the foundation is as follows:

$$\tau_{Friction} = -\mu \rho H g A$$

where μ is the coefficient of friction, []
 ρ is the density of the salt, kg/m³
 H is the depth of the salt, m
 g is the acceleration due to gravity, m/sec²
 A is the floor area, m²

The shear force, kg-m/sec², that the foundation applies to the tank is given by the following:

$$\tau_{Foundation} = \frac{G * u * A}{t_{Foundation}}$$

where G is the shear modulus of the foundation, kg/m-sec²
 u is the deflection of the foundation, m
 A is the floor area, m²
 $t_{Foundation}$ is the thickness of the foundation, m

A differential element of the floor is illustrated in Figure 5-7.

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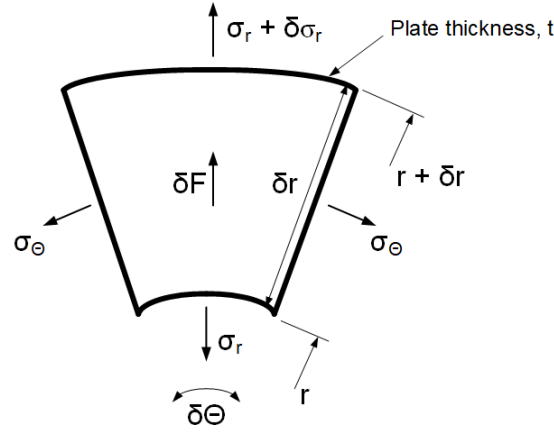


Figure 5-7 Differential Element of the Floor

Setting the force balance to zero yields the following:

$$2 \sigma_{\theta} \sin\left(\frac{\delta\theta}{2}\right) \delta r t - \sigma_r \delta\theta r t + (\sigma_r + \delta\sigma_r) (r + \delta r) \delta\theta t + \delta F = 0$$

Using a small angle approximation $\sin\left(\frac{\delta\theta}{2}\right) \approx \frac{\delta\theta}{2}$, setting $\delta A = \delta r r \delta\theta$, dividing by $\delta A / t$, and setting $\tau = \frac{\delta F}{\delta A}$ yields:

$$\frac{\delta\sigma_r}{\delta r} + \frac{1}{r}(\sigma_r - \sigma_{\theta}) = -\frac{\tau}{t_p}$$

where σ_r is the radial stress, kg/m-sec² (i.e., Pa)

r is the radius, m

σ_{θ} is the tangential stress, kg/m-sec²

τ is the unit shear force between the floor and the foundation, kg/m-sec²

t_p is the thickness of the floor, m

In an axisymmetric plate, the radial strain, $[\]$, is:

$$\varepsilon_r = \frac{\delta u}{\delta r}$$

the tangential strain, $[\]$, is:

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$$\varepsilon_{\theta} = \frac{u}{r}$$

the radial stress, kg/m-sec², is:

$$\sigma_r = \frac{E}{(1-\nu^2)} (\varepsilon_r + \nu \varepsilon_{\theta})$$

where ν is Poisson's Ratio, []

and the tangential stress, kg/m-sec², is:

$$\sigma_{\theta} = \frac{E}{(1-\nu^2)} (\varepsilon_{\theta} + \nu \varepsilon_r)$$

Substituting in the equations above results in a second order differential equation for the deflection:

$$\frac{d^2 u}{dr^2} + \frac{1}{r} \frac{du}{dr} - \frac{1}{r^2} u + \frac{(1-\nu^2)}{E} \frac{\tau}{t_p} = 0$$

The solution involves a number of terms, including the following:

- Concentrated vertical load of the wall and the roof, carried down to the perimeter of the floor
- Coefficients of friction, which can have different values for area under the majority of the floor and the area under the annular plate beneath the wall
- Thermal expansion and contraction of the tank due to daily changes in the temperature of the inventory
- Shear stiffness of the foundation
- Radial location in the floor where the shear force reaches a maximum.

The solution is subject to a boundary condition, in which the radial stress at the center of the floor is equal to the tangential stress at the center of the floor.

The location in the floor where the shear force reaches a maximum is designated r_2 . For radial locations greater than r_2 , the radial stress in the floor is given by:

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$$\sigma_r = \frac{W_{Wall} g C_{F_{Perimeter}}}{2 \pi R t_p} + \frac{(3 + \nu)}{3} \frac{(C_{F_{Floor}} \rho g H)}{t_p} (R - r)$$

where W is the weight of the wall and the roof, kg

$C_{F_{Perimeter}}$ is the coefficient of friction beneath the annular plate, []

$C_{F_{Floor}}$ is the coefficient of friction beneath the majority of the floor, []

R is the tank radius, m

r is the radius at which the stress is calculated, m

and the tangential stress is given by:

$$\sigma_\theta = \frac{W_{Wall} g C_{F_{Perimeter}}}{2 \pi R t_p} + \frac{1}{3} \frac{(C_{F_{Floor}} \rho g H)}{t_p} ((3 + \nu)R - (1 + 3\nu)r)$$

For radial locations less than r_2 , the expressions for the radial and the tangential stresses are as follows:

$$\sigma_r = \frac{W_{Wall} g C_F}{2 \pi R t_p} + \frac{(3 + \nu)}{3} \frac{(C_F \rho g H R)}{t_p} - \left(\frac{5}{24}\right) \frac{(3 + \nu) (C_F \rho g H)^2}{(t_p g_F \alpha \Delta T)} - \left(\frac{1}{8}\right) \frac{(3 + \nu) (g_F \alpha \Delta T) r^2}{t_p}$$

$$\sigma_\theta = \frac{W_{Wall} g C_F}{2 \pi R t_p} + \frac{(3 + \nu)}{3} \frac{(C_F \rho g H R)}{t_p} - \left(\frac{5}{24}\right) \frac{(3 + \nu) (C_F \rho g H)^2}{(t_p g_F \alpha \Delta T)} - \left(\frac{1}{8}\right) \frac{(1 + 3\nu) (g_F \alpha \Delta T) r^2}{t_p}$$

Coefficient of Friction Values

A common solid lubricant used under the floors of commercial tanks is sand. A paper by Uesugi¹⁹ summarizes the results of experiments to measure the coefficient of friction between a steel plate and sand.

The coefficient is a function of the surface roughness of the plate, the size of the sand particles, the vertical loading on the sand, and the number of strain reversals. However, the coefficients generally fall within a fairly narrow range of 0.5 to 0.7.

Representative Results for a Commercial Tank

For the purposes of the discussion, stresses due to friction were calculated using the following parameters:

¹⁹ Uesugi, Morimichi, et. al., (Department of Architectural Engineering, Chiba University, Japan), 'Friction Between Sand and Steel under Repeated Loading', Soils and Foundation Volume 29, No. 3, 127-137, September 1989, Japanese Society of Soils Mechanics and Foundation Engineering

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- Tank radius of 21.0 m at ambient temperature
- Coefficient of friction of 0.6 between the floor and the sand
- Coefficient of friction of 1.0 between the perimeter of the floor and refractory bricks beneath the perimeter of the floor
- Floor thickness of 7.14 mm
- Foundation shear stiffness of 320,000,000 kg/m²-sec².

Case 0

Case 0 corresponds to the following transition: 1) the tank is empty, at a temperature of 20 °C, followed by 2) the tank is preheated to a temperature of 320 °C over the course of several days.

A plot of the radial and the tangential stresses is shown in Figure 5-8. Negative values correspond to compressive stresses; i.e., the expansion of the floor is resisted by the shear stiffness of the foundation.

Since the tank is empty, the vertical loads on the floor are largely determined by the weight of the floor; i.e., close to zero. The radial and the tangential stresses, which are determined by the friction forces at the perimeter of the tank due to the weight of the wall, the roof, and the insulation, have nominally the same values.

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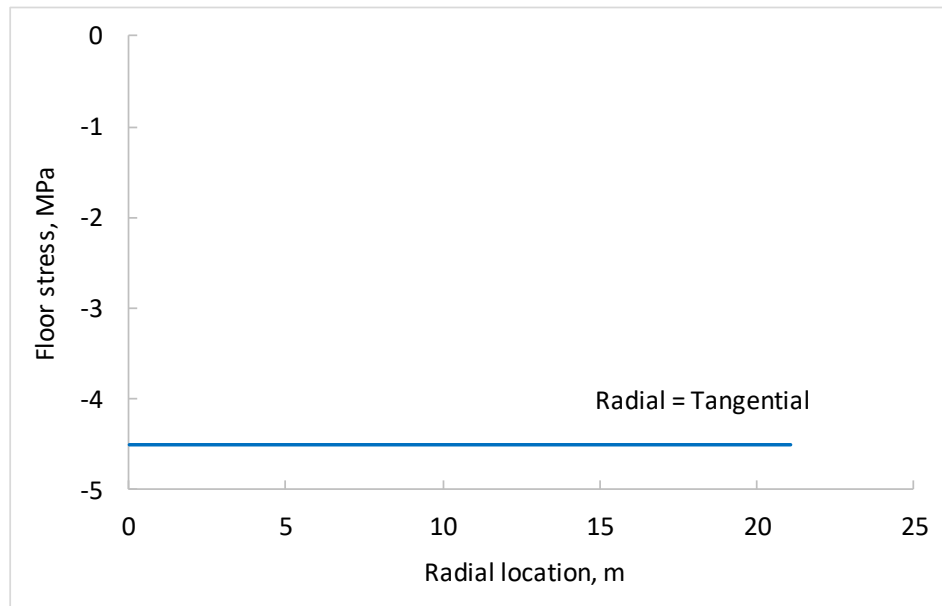


Figure 5-8 Radial and Tangential Floor Stresses for Case 0

Case 1

Case 1 corresponds to a condition in which 1) salt, at a nominal temperature of 320 °C, is loaded into the tank to a depth of 1.5 m after the tank has been preheated, followed by 2) the inventory temperature is raised to a value of 565 °C over the course of several days. The inventory depth remains at 1.5 m during this multi-day process to minimize the friction loads between the floor and the foundation.

With the inventory depth at the minimum, the radial and the tangential stresses in the floor, as a function of the radial location, are shown in Figure 5-9. Negative values for the stresses correspond to compressive loads.

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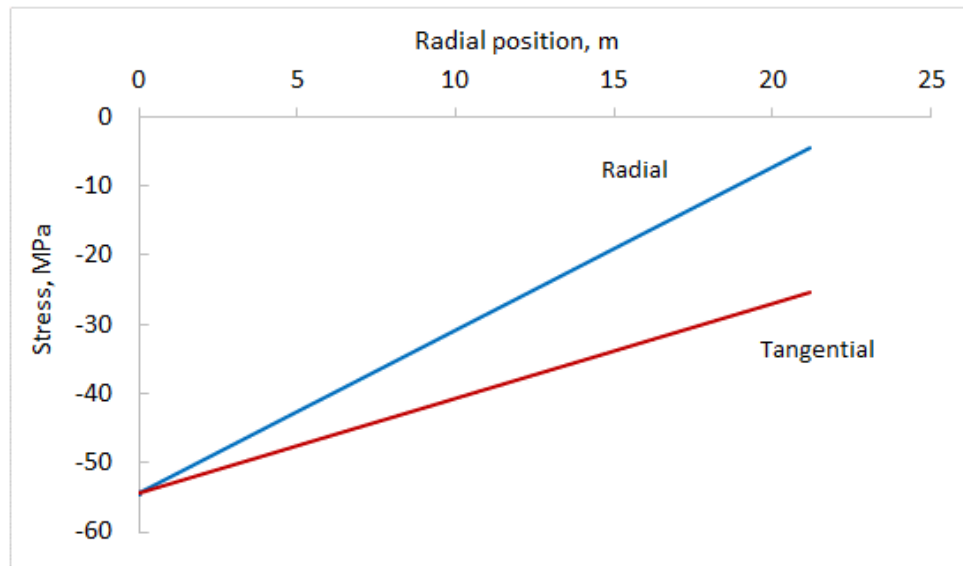


Figure 5-9 Floor Stresses for Case 1: 21 m Tank Radius; 7 mm Floor Thickness

The radial stress at the perimeter of the tank is not equal to zero due to the friction forces generated at the perimeter by the weight of the wall, the roof, and the insulation.

Case 2

In Case 2, the plant is in normal commercial service. At the beginning of the day, the inventory depth is 1.5 m. Over the course of the day, the inventory depth increases to 10.5 m. The inventory temperature remains constant at 565 °C. Since the tank neither expands nor contracts, the stresses in the floor remain constant at the profiles shown in Figure 5-9.

The project decides to postpone the startup of the turbine for a period of 8 hours to take advantage of higher energy sales prices. During this period, the tank cools from an initial temperature of 565.0 °C to a new temperature of 562.9 °C ($\Delta T = -2.1$ °C). The temperature decays causes the tank to contract, and a ΔT of -2.1 °C represents the largest temperature change which can occur before the tank begins to slide on the foundation.

The floor stresses generated by this condition are shown in Figure 5-10. Positive values correspond to tensile loads.

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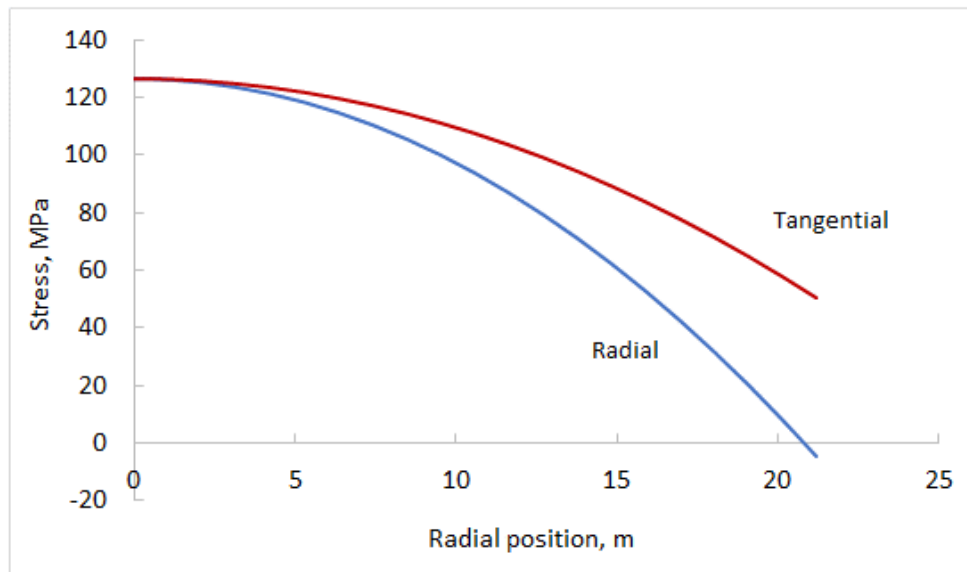


Figure 5-10 Initial Floor Stresses for Case 2: 21 m Tank Radius; 7 mm Floor Thickness

The stresses shown in Figure 5-10 assume that the floor starts in an unstressed condition. However, this is not the case, as the floor starts with the stress distribution shown in Figure 5-9. The stresses shown in Figure 5-9 can be combined with the stresses shown in Figure 5-10 if the change in shear stress with radius is a constant value; i.e., the tank has not yet started to slide. The linear superposition of stresses (i.e., Figure 5-9 + Figure 5-10) is shown in Figure 5-11.

Of note in Figure 5-11 are 1) the stress values, and 2) a situation in which a change in the inventory temperature of small as 2 °C can produce stresses within 80 percent of the allowable stresses listed in Section II of the Code.

Case 3

In Case 3, the tank begins the day with an assumed inventory temperature of 550 °C and an inventory level of 10.5 m. The inventory level in the morning is initially a high value in anticipation of meeting a high demand for electric energy during the day.

In the morning, the steam turbine is placed in service. Over the course of the day, the thermal power supplied by the receiver is nominally similar to the thermal power demand of the steam generator. As such, the inventory level in the hot tank largely remains constant at 10.5 m. However, the inventory temperature can increase above 550 °C if the solar radiation conditions are favorable and the receiver can operate for several hours at the design temperature of 565 °C.

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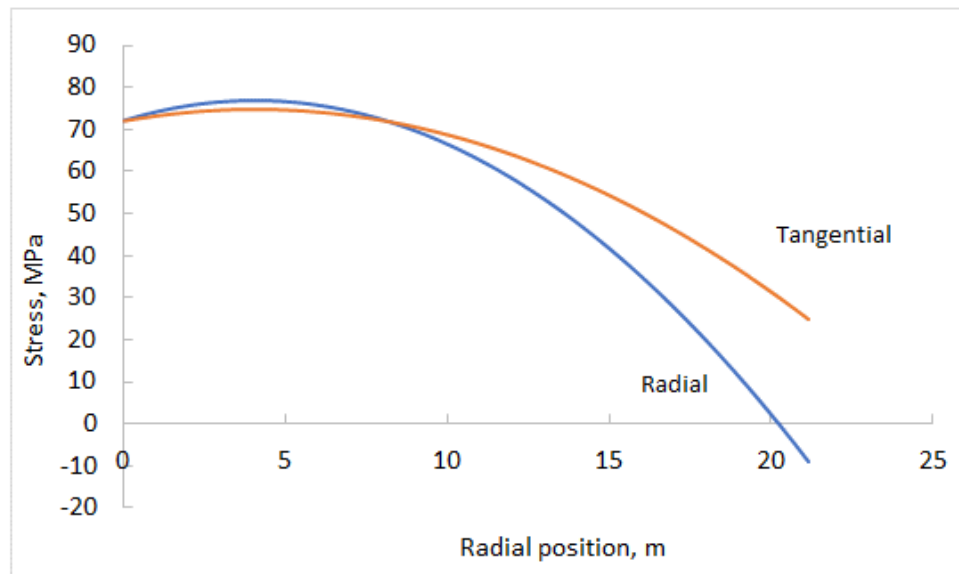


Figure 5-11 Final Floor Stresses for Case 2: 21 m Tank Radius; 7 mm Floor Thickness

Assuming that the inventory temperature reaches the design value of 565 °C, the stresses produced in the floor due to the expansion of the tank are shown in Figure 5-12. The combination of a high inventory level (10.5 m) and an increase in the inventory temperature (15 °C) places compressive loads on the floor near the center of the tank, and these stresses exceed the yield value. Similarly, if the tank has a high inventory level, but cools by 15 °C, then the tank will contract, generating a mirror image (tensile, rather than compressive) of the floor stresses.

Several items can be noted in this discussion:

- The combination of a high inventory level and an increase in the inventory temperature is not a hypothetical condition. It has been observed in commercial projects, when the project is trying to tailor electric energy deliveries to the needs of the local utility
- Similarly, the combination of a high inventory level and a decrease in the inventory temperature can occur if an afternoon with intermittent clouds results in a series of receiver start / stop cycles, each of which adds relatively cold salt to the hot tank

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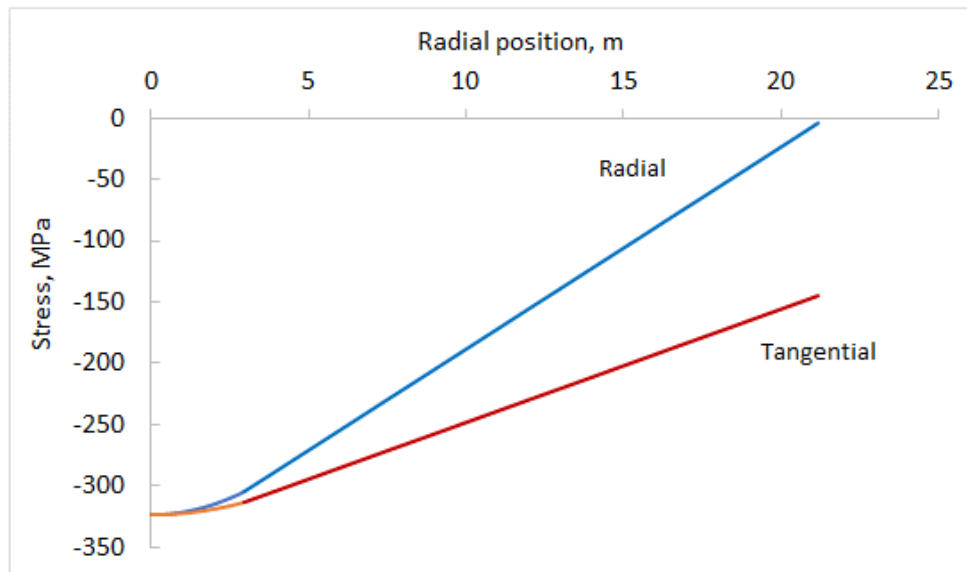


Figure 5-12 Floor Stresses for Case 3: 21 m Tank Diameter; 7 mm Floor Thickness

- The stresses shown in Figure 5-12 assume that the tank starts in condition with zero initial stress. Clearly, this will not be the case, as the stresses at any particular time will be determined by the temperature / level history of the tank up to that point. As might be imagined, 1) the history of the tank will largely be unknown, or 2) even if the history was known, it's unlikely that the project could develop a running calculation of the floor stresses which had a meaningful accuracy. To bound the problem, the stresses shown in Figure 5-9 (Case 1) represent a case in which 1) the inventory is at, or near, the minimum level (1.5 m), and 2) the tank has been sliding over the entire floor area, except for a small area near the center of the tank. This, in turn, should produce, in essence, a baseline stress at an inventory level of 1.5 m. As such, with inventory depths at or near the minimum, the stresses at the center of the tank should oscillate between two baseline values: -55 MPa (tank expanding); and +55 MPa (tank contracting). To a zeroth-order, the 55 MPa baseline stresses at the center of the tank will either add to, or subtract from, the stresses at the center of the tank developed at other combinations of level and temperature; i.e., those shown in Figure 5-12. The exact value for the stress at the center of the tank will require a more sophisticated calculation than the analytical solution noted above. Nonetheless, stresses well above the yield value are expected, particularly if compressive (tensile) stresses developed during the day add to baseline compressive (tensile) stresses carried over from when the tank was last at the minimum level.

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Parametric Tank Designs

The friction stresses in a representative tank (21.0 m radius; 7 mm floor thickness), operating through typical daily and annual thermal cycles, are likely to be well above values which could be considered consistent with a creep lifetime of 30+ years.

To prevent potentially damaging stresses in the floor, the temperature of the inventory must be controlled to values in the range of perhaps ± 2 °C of a reference value. However, the transient nature of the plant operations, plus conditions outside the project's control (i.e., forced outages), are likely to exceed the ± 2 °C operating constraint on a regular basis.

One possible method for reducing the friction stresses in the floor is to make change changes to the tank design. As an example, one 100-percent capacity hot tank could be replaced by two 50-percent capacity tanks, and the thickness of the floor could be doubled from the minimum specified in API 650 (i.e., 2 x (6.35 mm + corrosion allowance)). This would result in a tank radius of 14.9 m and a floor thickness of 14.3 mm.

For this analysis, the balance of the design parameters, including the coefficient of friction between the floor and the foundation (0.6), the coefficient of friction between the annular perimeter plate and the foundation (1.0), and the shear modulus of the foundation (320,000,000 kg/m²-sec²), are assumed to remain unchanged.

Case 1

As in the previous section, Case 1 corresponds to a condition in which 1) the tank is preheated to 300 °C, 2) salt, at a nominal temperature of 320 °C, is loaded into the tank to a depth of 1.5 m, and 3) the inventory temperature is raised to 565 °C over the course of several days.

With the inventory at the minimum depth, the radial and the tangential stresses in the floor, as a function of the radial location, are shown in Figure 5-13. Negative values correspond to compressive loads.

To a first order, reducing the tank radius from 21 m to 15 m, and doubling the floor thickness from 7 mm to 14 mm, reduces the peak stresses at the center of the tank by about 60 percent (55 MPa down to 21 MPa).

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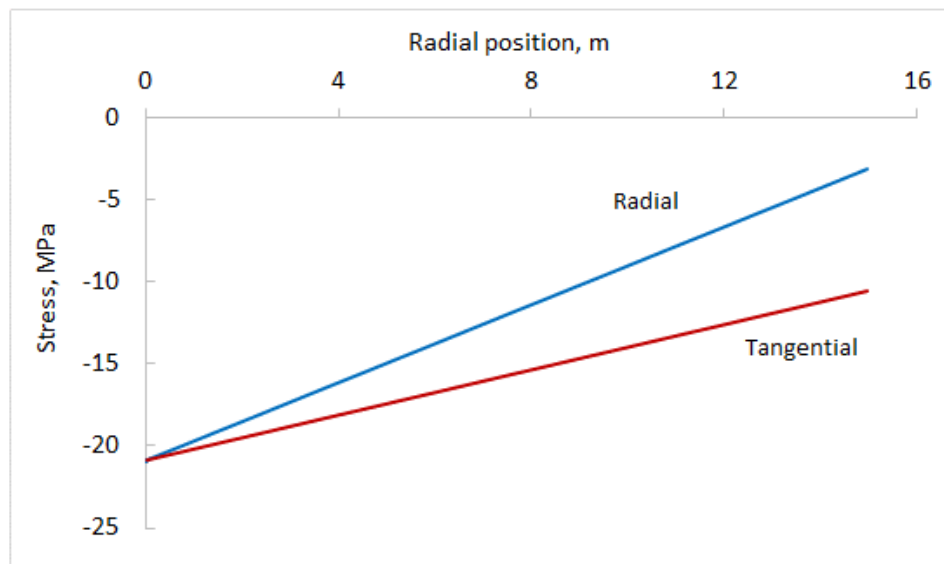


Figure 5-13 Floor Stresses for Case 1: 15 m Tank Radius; 14 mm Floor Thickness

Case 2

Similar to Case 2 above, the tank cools from an initial temperature of 565.0 °C down to a new temperature of 562.0 °C ($\Delta T = -3.0$ °C). The temperature decays causes the tank to contract, and a ΔT of -3.0 °C represents the largest temperature change which can occur before the tank begins to slide on the foundation. The associated stresses are shown in Figure 5-14. Positive values represent tensile stresses.

As above, the compressive stresses shown in Figure 5-13 can be combined with the tensile stresses shown in Figure 5-14 using linear superposition. The results are shown in Figure 5-15. To a first order, reducing the tank diameter by a factor of 2 and doubling the floor thickness reduces the peak stresses at the center of the tank by a nominal 65 percent (78 MPa down to 25 MPa).

Case 3

Similar to Case 3, the tank begins the day with an assumed inventory temperature of 550 °C and an inventory level of 10.5 m. The inventory level remains constant during the day; however, the inventory temperature rises by 15 °C to a value of 565 °C by late in the afternoon. The associated stresses are shown in Figure 5-16.

Reducing the tank diameter by a factor of $\sqrt{2}$ and doubling the floor thickness reduces the peak stresses at the center of the tank by a nominal 65 percent (320 MPa down to 115 MPa). Importantly, the values are now the same order of magnitude as the allowable stresses listed in Section II of the Code.

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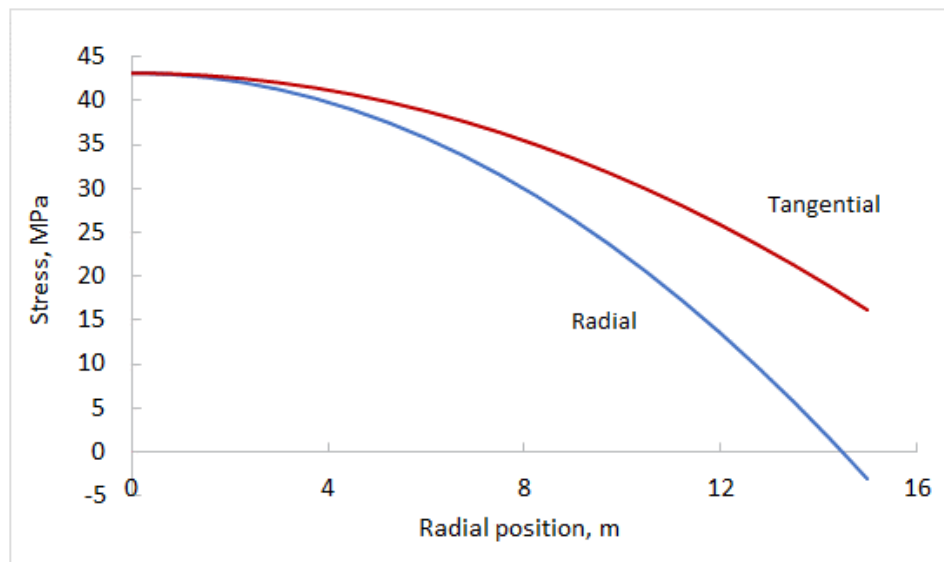


Figure 5-14 Initial Floor Stresses for Case 2: 15 m Tank Radius; 14 mm Floor Thickness

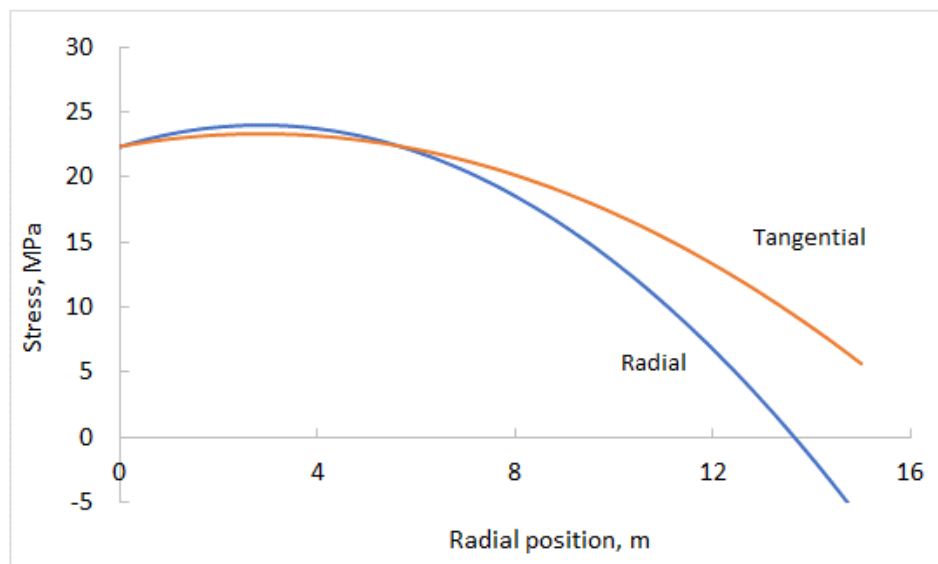


Figure 5-15 Final Floor Stresses for Case 2: 15 m Tank Radius; 14 mm Floor Thickness

As noted above in the discussion for the 21 m tank, an accurate estimate of the floor stresses for the 15 m tank will require 1) a more sophisticated approach than an analytical solution, and 2) analyzing the history of the floor stresses prior to the condition of interest. As such, it's not possible to state that the floor stresses for the 15 m tank will always be less than the allowable stresses listed in Section II of the Code. Nonetheless, it is apparent that there is likely a combination of tank diameters and floor

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thicknesses that result in stresses due to friction, in combination with stresses due to temperature gradients (Section 5.7), that satisfy the allowable limits discussed in Section 5.14.1, Changes in the Tank Design Codes.

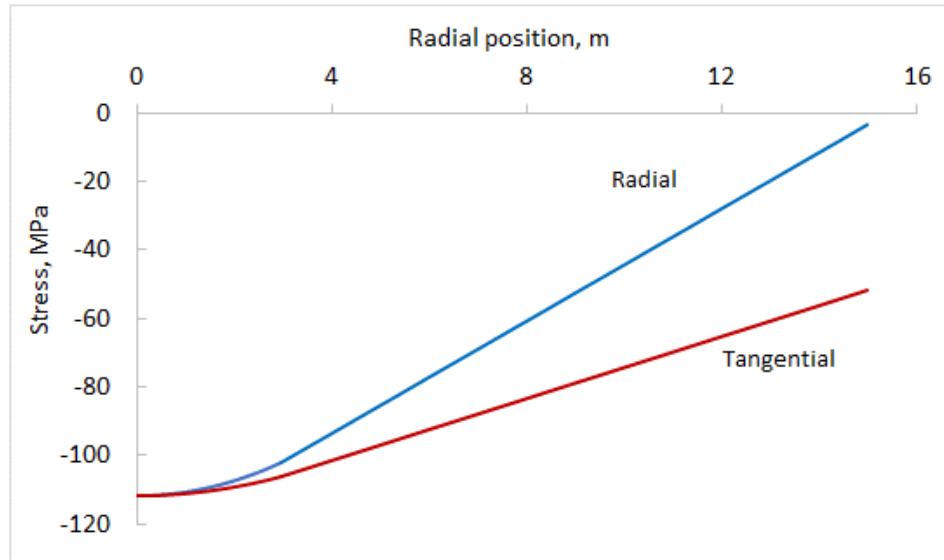


Figure 5-16 Floor Stresses for Case 3: 15 m Tank Radius; 14 mm Floor Thickness

5.11 Salt Inlet Flow Distribution

The salt flow entering the tank is typically distributed by means of a single ring header. The circumference of the header is in the range of 40 to 60 percent of the circumference of the tank. The header has either a series of holes or a group of eductors to distribute the incoming flow with the main inventory.

The distribution arrangement is simple and offers a low cost. However, it can suffer various performance liabilities, as follows:

1. The effectiveness of the mixing is dependent on the velocity of the salt leaving the distribution ring. At low receiver flow rates, the salt velocities in the header will be low, and the extent of the mixing will be limited. Imperfect mixing can either add to, or subtract from, the existing radial temperature gradient in the floor. Any operating condition which increases the radial temperature gradient carries the risk of adding creep or fatigue damage to the floor.
2. During receiver startup, or following a trip of the receiver, both the temperature and the flow rate of the salt can be well below design values. A condition can occur in which relatively cold salt

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leaves the distribution ring, and the salt sinks to the floor due to buoyancy effects. This, in turn, can lead to a non-uniform temperature distribution in the floor immediately below the distribution ring.

5.11.1 Stresses Calculated from Analytical Expressions in Roark

The stresses associated with this flow condition can be estimated using the following expression from Roark:

A thin circular disk at uniform temperature has the temperature changed ΔT throughout a comparatively small central circular portion of radius a . Within the heated part there are radial and tangential compressive stresses $\sigma_r = \sigma_t = \frac{1}{2} \Delta T \gamma E$. At points outside the heated part, a distance r from the center of the disk but still close to the central portion, the stresses are $\sigma_r = \frac{1}{2} \Delta T \gamma E a^2/r^2$ (compression) and $\sigma_t = \frac{1}{2} \Delta T \gamma E a^2/r^2$ (tension); at the edge of the heated portion, there is a maximum shear stress $\sigma_r = \frac{1}{2} \Delta T \gamma E$.

Using representative values for the modulus of elasticity, E , and the coefficient of thermal expansion, γ , for stainless steel, a local temperature depression of 54 °C results in a stress equal to the allowable stress for stainless steel at 560 °C. A local temperature depression of 87 °C produces stresses equal to the yield stress. The local stresses can either add to, or subtract from, the stresses normally present in the floor.

It can be noted that differences between the temperature of the salt in the distribution ring and the temperature of the bulk inventory can differ, during transient conditions, by values up to at least 90 °C.

5.11.2 Stresses Calculated from Finite Element Models

A computational fluid dynamics / finite element model has been developed for a hot salt tank at a commercial project. Using historical operating data from the project for inventory temperatures, flow conditions into the tank (flow rate, temperature), and flow conditions out of the tank (flow rate, temperature), floor temperatures have been calculated for a number of radial and circumferential locations as a function of time.

Using data from one particular day, which included a number of cloud transients, the differences in floor temperatures at two radial locations are shown in Figure 5-17. The two locations are as follows:

- Between Node 2 (17 percent of the tank radius) and Node 3 (33 percent of the tank radius)
- Between Node 5 (67 percent of the tank radius) and Node 6 (85 percent of the tank radius)

The circumferential location is 90° clockwise from North.

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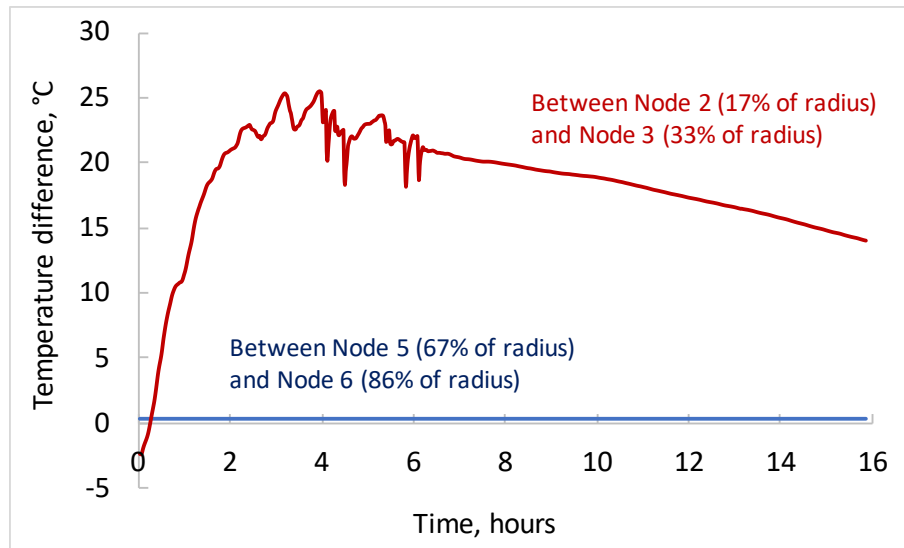


Figure 5-17 Differences in Floor Temperatures as a Function of Time; 90° Circumferential Location

The radial temperature difference is concentrated near, and somewhat inside, the location of the distribution header (between Nodes 2 and 3). Near the perimeter of the tank (between Nodes 5 and 6), the radial gradient has decayed to about 0.5 °C.

A plot of the circumferential temperature differences at Node 6, as a function of time, is shown in Figure 5-18. The temperature differences are calculated between circumferential locations of 45° and 90°.

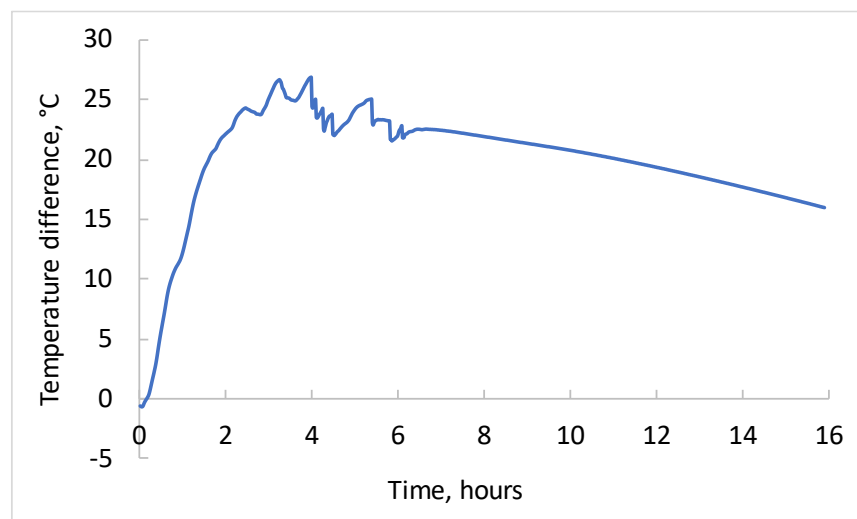


Figure 5-18 Differences in Floor Temperatures as a Function of Time; Node 6 Between 45° and 90°

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With reference to Figure 5-17, the radial temperature difference at Node 6 is less than 1 °C during the 16 hour time span of interest. However, the circumferential temperature difference at Node 6 is greater than 20 °C for most of the duration.

Similar to Figure 5-18, a plot of the circumferential temperature differences at Node 6, for circumferential locations of 90° and 135°, is shown in Figure 5-19.

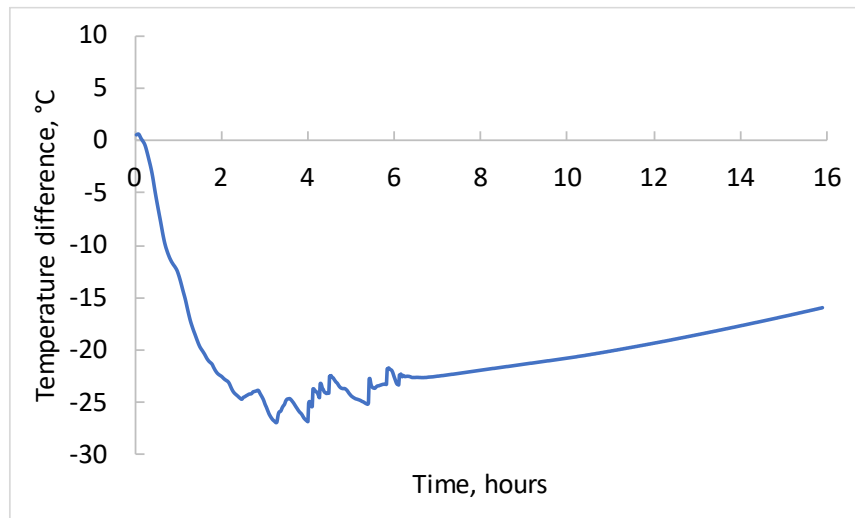


Figure 5-19 Differences in Floor Temperatures as a Function of Time; Node 6 Between 90° and 135°

The temperature differences in Figure 5-18 are essentially the mirror image of the temperature differences in Figure 5-19, which means that temperature differences between the 45° location and the 135° location are only a few °C.

The von Mises stresses in the floor at Node 6, generated by the circumferential temperature differences noted above, are shown in Figure 5-20. Throughout the calculation period, the stresses exceed the yield stress. If the material is routinely operating at or above the yield stress, then the creep life of the floor will be less than the 30-year life of the project.

Several items can be noted from the discussion above:

- The analysis was conducted for only one day from the multiple days of operating data supplied to the study by the commercial project. Only one day of data were analyzed due to the lengthy computational time required (several weeks on a high performance computer)

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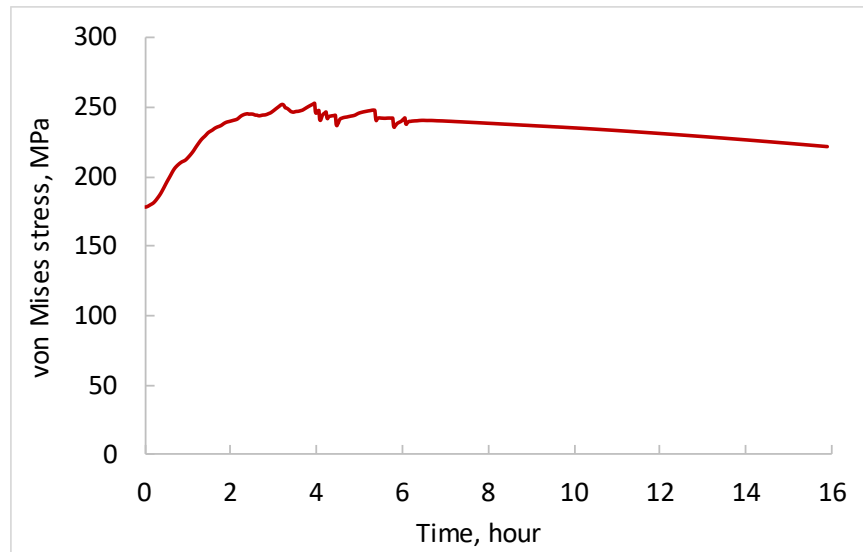


Figure 5-20 Von Mises Stresses at Node 6

- The day analyzed was selected to be representative of an average day with intermittent clouds. This day may, or may not, represent a worst-case condition for floor stresses
- At the start of the calculation, the stresses in the floor are taken to be equal to the stresses on the first day of commercial service. However, the calculation was developed based on plant operating data from (say) Day 905 of commercial service. During this 905 day period, the stresses are expected to evolve due to stress relaxation in the weld regions, plastic deformations associated with creep, plastic deformations generated by high friction forces coincident with changes in the inventory temperature, lateral movement of the tank due to non-isotropic thermal expansion and contraction, and vertical movement of the tank due to foundation settlement. The stress evolution over this 905 day period are unknown, and as such, the initial stress state for the calculation are also unknown. This necessarily introduces some level of uncertainty in the values shown in Figure 5-20.

The evolution of the floor stresses over months or years is difficult to determine, in large part due to the large computational requirements. Nonetheless, it is clear that even modest temperature differentials (25 to 30 °C) in the inventory, either radial or circumferential, can generate stresses that are nominally twice the allowable values discussed in Section 5.15.11.

5.11.3 Optimization Studies for the Inlet Distribution Header

As discussed in the introduction to Section 5.11, inlet distribution systems in commercial projects use either a series of holes in the header or a number of mixing eductors welded to the header. The

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effectiveness of mixing between the incoming flow and the tank inventory will influence the temperature gradients, and the associated stresses, in the floor.

A parametric study of the diameter (fraction of tank diameter) of the distribution header, and the number and the orientation of eductors, was conducted to help define the elements of an optimum design ²⁰. Three sets of parametric studies were conducted.

First Parametric Study

Two load cases were considered, as follows:

- Salt entered the distribution header at a temperature which was 175 °C below the initial reservoir temperature. Flow rates ranged between 10 percent and 100 percent of the design value, and the salt level in the tank was the minimum
- Salt entered the distribution header at a temperature which was 60 °C above the initial reservoir temperature. Flow rates ranged between 10 percent and 100 percent of the design value, and the salt level in the tank was the minimum.

The load cases were developed from operating data provided by a commercial project.

The first parametric study showed that cold salt injection developed the largest temperature gradients in the floor.

Second Parametric Study

A second parametric study was based on the following parameters:

- Distribution header with a diameter equal to 55 percent of the tank diameter
- Option 1: Distribution header with 48 orifices distributed along the top of the pipe
- Option 2: Distribution header with 2 eductors; one pointed inwards, and the other pointed outwards. Both eductors had a 15 degree pitch relative to the horizontal, and both eductors were offset by 10 degrees from the radial orientation
- Cold salt injection, with a flow rate equal to the design value.

²⁰ Vast Energy, 'Reducing Thermal Loads in Tanks through Sparger Design', Presentation at SolarPACES 2024, 10 October 2024

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With Option 1, the peak range in the floor temperature was 34 °C; with Option 2, the peak value was 17 °C. As such, the eductor arrangement in Option 2 is more effective at homogenizing the inventory temperature than the orifice arrangement in Option 1.

Third Parametric Study

A third parametric study was conducted, based on the following parameters:

- Distribution header with a diameter equal to 17 percent of the tank diameter
- Eductors with a pitch of 15 degrees, in combination with radial offsets of 10, 15, and 20 degrees
- Eductors with a pitch of 10 and 20 degrees, in combination with a radial offset of 15 degrees.

The arrangement with an eductor pitch of 15 degrees and a radial offset of 15 degrees showed the smallest range in floor temperature (17 °C). However, all of the combinations of pitch and radial offset showed ranges in the floor temperature that were no higher than 21 °C.

It should be noted that the parametric study examined only one tank diameter and a limited number of eductor arrangements. For a specific project, the optimum design for the distribution header will be a function of the tank dimensions, the allowable pressure drop between the downcomer and the tank inventory, and the selected process requirements (i.e., allowable difference in temperature between the salt in the downcomer and the salt in the tank).

5.12 Operator Actions

During receiver startup, the temperature of the salt leaving the receiver starts at the cold salt temperature (295 °C), and then increases to the normal design temperature (565 °C). Early in the startup process, salt is directed to the cold tank to prevent a noticeable decay in the inventory temperature of the hot tank. When the salt temperature in the downcomer reaches a defined crossover temperature, the flow is directed to the hot tank.

The crossover temperature is a function of the inventory temperature in the hot tank. A typical crossover temperature is 20 to 50 °C below the inventory temperature. Note that this phase of the startup process results in a small decay in the inventory temperature. However, the energy available in the salt at temperatures between the crossover temperature and the design temperature is still useful for electric energy production.

The decay in the inventory temperature is a function of the crossover temperature and the inventory level in the hot tank. The crossover temperature can be programmed in the DCS, and the control system

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will automatically open and close the tank diversion valves as necessary. However, as with most setpoints in the DCS, the operator is free to revise the setpoint. If the crossover setpoint is set to a low value, then additional thermal energy will be stored in the hot tank for electric energy production. During the project warranty and guarantee period, this can be an important consideration for the EPC contractor. However, the lower the setpoint, the greater the risk to the tank in terms of non-uniform temperature distributions, particularly in the floor, and the greater the potential for exceeding the limits from the vendor on the allowable rate of temperature change. Experience from more than one commercial project has shown that protecting the low cycle fatigue life of the tank often takes secondary importance to meeting the contractual requirements on energy production.

5.13 Cumulative Effects

As discussed above, plastic deformations in the floor could result from welding the floor plates, preheating the tank, and filling the tank with salt. The deformations may well take the form of buckles, or ridges, spanning a significant fraction of the diameter of the tank.

Once in commercial service, the tank will undergo a series of changes in level and changes in temperature. Some of these changes will place the floor into compression. If some of these loads cause the floor to yield, then the displacements are likely to appear as increases in the height of the ridges. The deformations can be compounded by local transient thermal stresses in the floor associated with receiver startups and receiver trips.

In tanks constructed from Type 347H, stress relaxation cracking will begin at the start of commercial service. The mechanism will progress at the highest rate at those locations with the largest residual stresses and the highest strains. The regions include the weld zones and the tops of the buckles. At some point, 1) the progression of internal cracks and micro-voids generated by stress relaxation cracking, 2) in combination with cyclic compression / tension loads generated by friction, 3) in combination with a reduction in fatigue life at locations subject to plastic strains, 4) produces a crack at the top of the ridge. The crack is the source of the leak in the tank. This failure mechanism has been observed in a number of commercial projects.

It can be noted that ultrasonic examination of the welds in the floor and in the wall of a commercial tank has shown evidence of voids. The examinations were conducted after the tank had been in service for about 2 years. The formation of voids in 2 years is consistent with expected progression of stress relaxation cracking. If stress relaxation cracking is present, then strain localization at the grain boundaries can accelerate the damage due to low cycle fatigue.

5.14 Solutions

The question then arises: What solutions can be effected to prevent damage to the tank? Two approaches can be considered:

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- A change in the tank design Codes
- Methods to reduce compressive loads on the floor.

5.14.1 Changes in the Tank Design Codes

Current Approach

To date, commercial tanks have been designed to the following combination of Standards and Codes:

- API Standard 650, Welded Steel Tanks for Oil Storage. However, the maximum design temperature in the Standard is 260 °C
- To overcome this limit, allowable material stresses are typically taken from Section II Part D Subpart 1A of the ASME Code for design temperatures above 260 °C
- Certain portions of the tank can operate at stresses greater than those listed in Section II. To ensure that the tank has a suitable fatigue life, the design by analysis requirements specified in Section VIII Division 2 are often adopted.

Stress Categories

The tank is subjected to a range of stress categories, including the following:

- Primary membrane stress, P_m , associated with static fluid pressures, operating pressures, and dead loads
- Local membrane stress, P_L , is a membrane stress typically generated by a discontinuity in the geometry
- Bending stress, P_b , produced, for example, by pressure applied to the central portion of a flat head
- Secondary stress, Q , produced by constrained thermal expansion or the bending stress at a structural discontinuity
- Peak stress, F , which does not cause any noticeable distortion but can be a source of a fatigue crack or a brittle fracture. An example is the thermal stress in the wall of a vessel caused by a rapid change in temperature.

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Allowable Stress Values in Section VIII Division 2

In Section VIII Division 2, the allowable values for various combinations of stress categories is illustrated in Figure 5-21 and include the following:

- Primary membrane stress $P_m = S$, where S is the value listed in Section II Part D Subpart 1. For Type 347H stainless steel at a temperature of 565 °C, a representative value for S is 117 MPa
- Local membrane stress $P_L = 1.5 * S$, or 234 MPa
- Local membrane stress P_L + Bending stress $P_b = 1.5 * S$, or 234 MPa
- Local membrane stress P_L + Bending stress P_b + Secondary stress $Q = S_{PS}$, where S_{PS} is the larger of 1) $3 * S$ from Section II, Part D, Table 5A, and 2) $2 * S_y$ (yield stress) from Section II, Part D, Table Y-1. For Type 347H stainless steel at a temperature of 565 °C, a representative value for S_{PS} is 352 MPa
- Local membrane stress P_L + Bending stress P_b + Secondary stress Q + Peak stress $F = S_a$, where S_a is alternating stress. The limit for the stress is obtained from a fatigue curve for the specified number of operating cycles, with corrections applied to the curve for mean stresses other than zero.

The values for the stresses are generally determined by finite element analyses.

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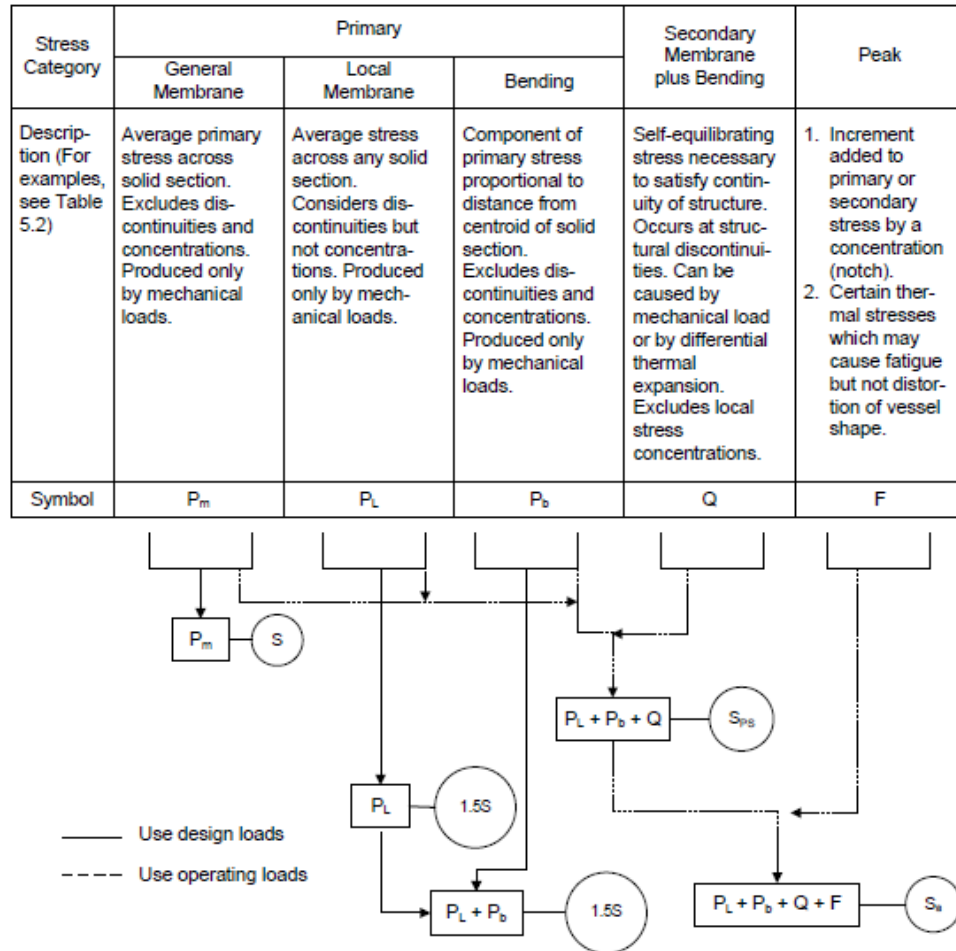


Figure 5-21 Combinations of Stresses and Allowable Values in Section VIII

Limitations of Section VIII Division 2

In the stress categories outlined in Figure 5-21, all of the allowable values are based on time-independent properties of the material. In particular, the effects of creep are not considered in the design.

In this design approach, the equipment is assumed to operate below the creep range by providing design margins against the following failure modes:

- Failure by plastic Instability or necking
- General structural collapse under a single application of limit load

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- Time-independent buckling
- Incremental collapse or ratcheting under cyclic loading
- Fatigue under cyclic loading
- Fast fracture.

Nonetheless, preliminary calculations of the stresses in one commercial tank show values which are high enough (> 300 MPa) to 1) reduce the fatigue life to values less than the 30-year life of the project, and 2) initiate creep at the temperatures characteristic of a hot salt tank. Granted, none of the commercial tanks built to date have failed due to creep, but this may be due to a condition in which other failure mechanisms have reduced the life of the tank to values which are too short to allow significant quantities of creep to develop. In any event, all of the hot salt tanks built to date have failed in one way or another. As such, one could ask if the requirements in the Standards and Code sections noted above is the correct combination to use in the design and the analysis of the tank.

Alternate Design Codes

The requirements of Section III, Rules for Construction of Nuclear Facility Components, apply to materials in service at high temperatures.

Section III Division 1 Subsection NB provides rules for the design and the evaluation of pressure vessels, including the effects of fatigue. Nonetheless, the Subsection largely follows the design approach described for Section VIII Division 2. In addition, the effects of creep are largely avoided by limiting the maximum design temperature to 427°C .

In contrast, Section III Division 1 Subsection NH provides rules for the design and the evaluation of pressure vessels, including the effects of both fatigue and creep. The Subsection applies to design temperatures up to 850°C .

Operation at elevated temperature introduces time-dependent failure modes. The following six time-dependent failure modes are considered in the high-temperature design:

- Creep rupture under sustained primary loading
- Excessive creep deformation under sustained primary loading
- Cyclic creep ratcheting due to steady primary and cyclic secondary loading

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S₀ for Use in the Design Limits Calculation

S₀ is the greater of the following two values:

- The maximum allowable stress value S in Section II Part D Table 1A. For Type 347H stainless steel at a temperature of 565 °C, the value is 110 MPa
- The S_m stress intensity at 300,000 hours in listed in Subsection HBB of the upcoming version of Section III Division 5, High Temperature Reactors. The text has yet to be published; however, publicly available inputs to the Section are available from Oak Ridge National Laboratory, Pacific Northwest Laboratory, Argonne National Laboratory, and NUMARK Associates. The value for S_m is listed as 53 MPa in the ORNL report ²¹.

As such, the value for S₀ is 110 MPa.

Time-Dependent Stress Value, S_t

In the design rules of Subsection NH, the material has assumed to reach a temperature where creep can produce deformations of the structure. S_t is a time-dependent allowable stress, and is defined as the lesser of the following parameters:

- 100 percent of the stress required to produce a permanent creep strain of 1 percent in a defined period. The allowable stress is determined through the use of a Larson Miller curve, which plots the stress as a function of the metal temperature and the logarithm of the operating period. An example for Type 316 stainless steel is shown in Figure 5-23.

The estimated stress to produce a 1 percent strain in 100,000 hours is 105 MPa

- 80 percent of the minimum stress to initiate tertiary creep. The creep characteristics are often divided into three phases: primary; secondary; and tertiary. The primary phase is generally short, and shows a relatively rapid change in dimension with time. The secondary phase is typically much longer. As the material deforms, its cross sectional area will reduce. However, the material also strain hardens. The increase in the strength generally offsets the reduction in area, and the true stress value remains nominally constant. At some point in the process, the effects of strain hardening cease and the material enters the tertiary phase. Gross deformations appear, which is generally followed soon after by rupture. Using a separate Larson Miller plot, the allowable stress to initiate tertiary creep is 84 MPa for a time span of 100,000 hours

²¹ Ren, Weiju, et. al. (Oak Ridge National Laboratory, Oak Ridge, Tennessee), 'Technical Input for the Nuclear Regulatory Commission Review of the 2017 Edition of the ASME Boiler and Pressure Vessel Code, Section III, Division 5, High Temperature Reactors', Technical Letter Report ORNL/SPR-2020/1653, August 2020

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- 67 percent of the minimum stress to cause rupture. Using a separate Larson Miller plots, the allowable stress is 106 MPa for a time span of 100,000 hours.

As such, the value for S_t is 84 MPa.

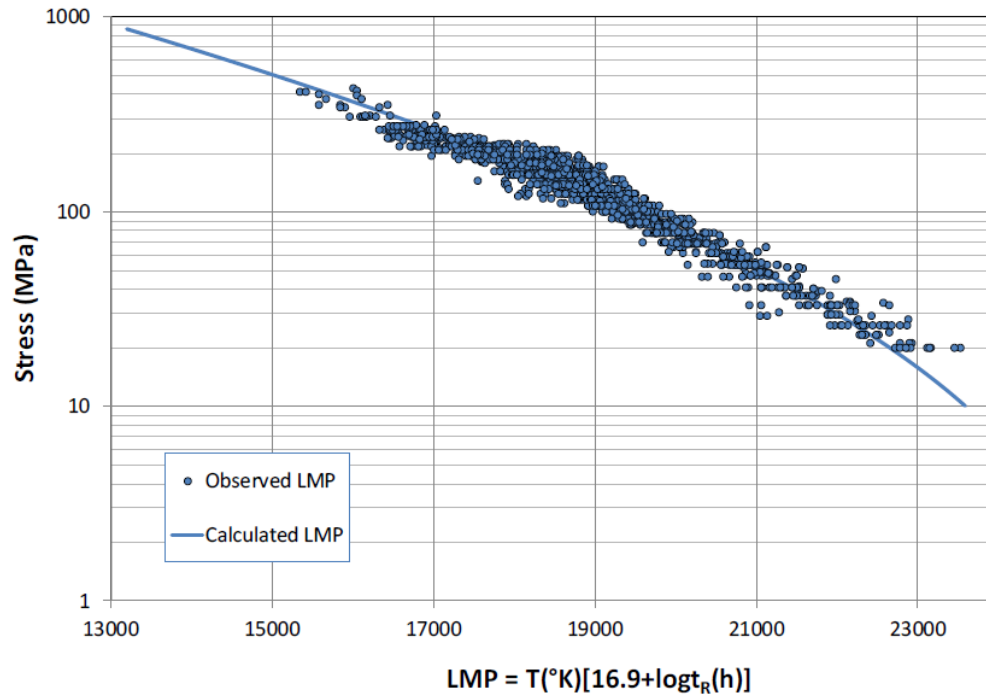


Figure 5-23 Larson Miller Diagram for the Stress to Develop 1 Percent Strain

Time-Independent Stress Value, S_m

S_m is the lesser of the following two values:

- 2/3 of the yield stress. For Type 347H stainless steel at a temperature of 565 °C, the value is 2/3 * 155 MPa = 103 MPa
- 1/3 of the tensile strength. For Type 347H stainless steel at a temperature of 565 °C, the value is 1/3 * 396 MPa = 132 MPa.

As such, the value for S_m is 103 MPa.

Lastly, S_{mt} is defined as the lesser of S_m (103 MPa) and S_t (84 MPa). As such, the value for S_{mt} is 84 MPa.

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K has a value of 1.5 for shells and solid sections, and K_t is defined as $(K + 1) / 2 = 1.25$.

From the Design Limits portion of the flow chart in **Error! Reference source not found.**, the allowable primary membrane stress, P_m , is equal to $S_o = 110$ MPa. The allowable sum of the local membrane stress, P_L , and the bending load, P_b , is equal to $1.5 * S_o = 165$ MPa.

For the Level A and B Service Limits, the allowable primary membrane stress, P_m , is equal to $S_{mt} = 84$ MPa. The allowable sum of the local membrane stress, P_L , and the bending load, P_b , is equal to $1.5 * S_m = 154$ MPa. The allowable sum of the local membrane stress, P_L , and the bending load, P_b , modified by the K_t factor is equal to $S_t = 84$ MPa.

Comparison of Section VIII Division 2 with Section III Subsection NH

Effects of Creep

The effects of creep are associated with operation at high temperatures, and the rate of creep is influenced by the stress. One of the contributions to the total stress can be a thermal stress produced by temperature gradients in the equipment.

In Section VIII Division 2, thermal stresses are treated as secondary stresses. As shown in Figure 5-21, secondary stresses are not components in the design limit calculations involving the primary membrane stress, the local membrane stress, and the bending loads. As such, any creep generated by thermal stresses is not considered in the design calculations.

In contrast, in Section III Subsection NH, the flow diagram in Figure 5-22 requires an analysis of the following:

- Deformations, based on either 1) a maximum allowable strain of 1 percent, or 2) the results from an inelastic analysis
- A creep-fatigue evaluation based on summations of the damages associated with the number of fatigue cycles and creep durations
- Buckling and instability criteria.

Allowable Stresses

In Section VIII Division 2, the highest numerical value for the allowable stress is S_{PS} , with a value of 352 MPa. In contrast, in Section III Subsection NH, the highest numerical value for the allowable stress is $1.5 * S_o$ (165 MPa). This value is about one-half of the design value from Section VIII Division 2.

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To date, no commercial tanks have been designed and built to the requirements of Section III Subsection NH, and direct comparisons to commercial tanks are, to some degree, conjectural. Nonetheless, differences are likely to be present in the following areas:

- Design limit and fatigue calculations from a representative commercial project show peak stresses on the order of 220 MPa at the connection between the floor and the wall. The calculations are based on a coefficient of friction of 0.2 between the floor and the foundation. Evidence from other studies indicate that the coefficient is more likely to be in the range of 0.5 to 1.0. If the tank stress calculations were repeated with a higher coefficient of friction, it's not clear that the design could satisfy the requirements of Section III Subsection NH
- As discussed in Section 5.7, steady state radial temperature gradients in the floor on the order of 5 °C can produce thermal stresses in the floor approaching the allowable value listed in Section II Part D Table 1A. In the design approach listed in Section VIII Division 2, these stresses are treated as secondary stresses, and do not influence the design limit calculations. In Section III Subsection NH, the thermal stresses in the floor will influence the creep and the fatigue assessments. This, in turn, may have an influence on the required design for the floor, which, in turn, would influence the buckling characteristics of the floor
- Based on the operating experience from a number of commercial tanks built to the requirements of API Standard 650, the vendor will often specify an allowable rate of temperature change. A representative value is 1 °C/min. However, operating data from the Crescent Dunes project has shown rates of change at least an order of magnitude higher. The fatigue damage associated with these rates of change has yet to be calculated. But, in future projects, the operating procedures must be specified in such a manner that the 30-year fatigue life of the tank can be satisfied.

Observations

Commercial tanks built to date have been designed using API Standard 650, Section II, and Section VIII Division 2. Essentially all of these tanks have developed leaks or other failures. Nonetheless, no studies have been conducted which demonstrate that this combination of design standards and codes, in combination of historical operating data on the tanks during transient conditions, has resulted in the tank failures. Similarly:

- Section III Subsection NH offers a more conservative set of design stresses and rules than Section VIII Division 2. However, no studies have been conducted which demonstrate that a tank, designed to the rules of Section III, would survive the life of the project
- The stresses in some portions of the tank, such as the wall and the roof, are generally predictable. As a result, it should be possible to estimate the creep and the fatigue life of these sections with

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an accuracy sufficient for a commercial project. In contrast, the stresses in the floor are not well understood, and the accuracies of the creep and fatigue calculations are generally unknown. As such, it may be possible to design the wall and the roof to the Codes typically used in previous commercial projects (i.e., API 650 and Section VIII Division 2), and adopt the more conservative design rules in Section III Division 5 for the floor.

Whichever combination of design rules is selected for the tank, the effect on the capital cost of the tank is likely to be modest. For example, in API Standard 650, the floor is essentially treated as a liquid boundary, and the thickness of the floor can be thin (i.e. 8 mm). However, following the rules of Section III Division 5, the allowable stresses associated with a creep and fatigue life of 30 years are relatively modest (~85 MPa), in which case the thickness of the floor may need to be double that of an API-style tank. Doubling the thickness of the floor increases the mass of steel in the tank by about 15 percent, which means, to a zeroth-order, that the capital cost of the tank also increases by 15 percent. A cost increase of this order will have essentially no effect on the financial status of the project.

To some extent, the industry is faced with the following conundrum:

- Can a set of analyses be conducted that convinces the financial community that a tank designed to the requirements of Section III Division 5, or a comparable international Code, will have an availability of 99.99 percent?
- In an ideal case, a flat bottom tank experiences uniform friction forces around the circumference of the tank. However, industry experience indicates that this may not be the case, as lateral movement of a tank in a commercial project has been observed. If it is not possible to determine the actual distribution of friction forces on the floor, then it may not be possible to determine the actual stresses in the floor. If so, an alternate tank concept, one that avoids a flat floor on a flat foundation, may be required to reach an availability of 99.99 percent. However, there will be no industry experience with an alternate tank design, and this will necessarily incur some form of technical and financial risk to the Owner
- The best, and perhaps the only, method to prove which design codes are the correct codes, and which tank concept is the correct tank concept, is to design, fabricate, and test a full-size tank at a commercial project over a period of several years. However, it's not clear who would be prepared to fund such a development program.

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5.14.2 Methods to Reduce Loads on the Floor

Multiple Tanks

The peak loads in the floor are, to some degree, a function of the tank diameter. Substituting two 50-percent capacity tanks, or three 33-percent capacity tanks, for one 100-percent capacity tank could reduce the loads in the floor to values below a yet-to-be-defined damaging threshold. The relationship between floor stresses due to friction and tank diameter are discussed in Section 5.10.4.

In addition, multiple tanks provide a degree of redundancy, and eliminate the single point of failure problem with one 100-percent capacity tank.

Nonetheless, the failure mechanisms in commercial tanks are not yet fully understood. Adopting smaller tanks may be a step toward a reliable design approach, but the maximum allowable tank dimensions have yet to be defined and demonstrated.

Radial Gradients

A few methods for controlling the radial temperature gradient in the floor include the following:

Insulation Design The radial temperature gradients can be reduced by changing the insulation design at the perimeter of the tank. This may consist of greater insulation thicknesses at the base of the tank, and the addition of insulation on the refractory ring wall beneath the wall. Nonetheless, a radial temperature gradient will still develop, and if the inventory reaches, or approaches, a stagnation condition, then damaging stresses in the floor will result.

Perimeter Heat Addition The radial gradient can also be reduced by some form of heat addition at the tank perimeter. The heat addition might take the form of radiant electric heaters, installed beneath the wall insulation. The principal liability would be a slow response time due to limited radiation heat fluxes from radiant heaters to the tank wall. Alternately, electric heaters could be embedded in the refractory ring wall. As above, the principal liability would be a slow response time due to the combination of a modest thermal conductivity, and a high thermal inertia, of refractory materials.

Multiple Distribution Headers An alternate approach to controlling the radial gradients is a dedicated distribution ring at the perimeter of the tank. Salt flow to the distribution ring would be supplied by recirculation from one of the hot salt pumps. The goal would be to move salt from the bulk inventory to the tank perimeter, and thereby provide forced convection heat transfer to compensate for the higher heat losses at the perimeter. The ability to move heat by forced convection is generally large relative to conduction heat transfer. As such, it should be a straightforward exercise to reduce the magnitude of the radial gradient.

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On a related point, the hot salt tank at the 10 MWe Solar Two demonstration project did not use a distribution ring header near the floor. Instead, a series of spray nozzles, supported from the inside of the roof, introduced salt into the tank. The (flawed) goal of the nozzles was to prevent the mixing of salt from the receiver, at a temperature of 565 °C, with the bulk inventory in the tank, which would normally be at a temperature less than 565 °C. The intent was to intentionally stratify the inventory, and only send salt, at a temperature of 565 °C, to the steam generator. This approach, of course, reduced mixing within the inventory, which promoted undesirable vertical radial temperature gradients in the floor. However, adding salt to the top of the inventory, rather than introducing salt near the floor by a distribution header, effectively isolated the floor from any transient conditions in the downcomer. To what extent this reduced the local transient stresses in the floor is unknown. However, the hot tank operated safely, without leaks, during the 3-year test and evaluation period.

In principle, a commercial project could also replace the conventional distribution header near the floor with a series of spray nozzles, or some other flow distribution arrangement, located inside the roof. Thermal stratification will inevitably occur. However, the methods discussed above, and below, should be effective in limiting both the vertical and the horizontal stratification to acceptable levels.

Tank Scale Mixing Device An alternate method to control the radial and the circumferential temperature gradients is to provide some form of a mixing device that spans the full dimensions of the tank. This might consist of a vertical shaft located at the center of the tank. Two to four vertical perforated plates would be attached to the shaft. The horizontal dimension of the plate would be somewhat less than the radius of the tank, and the height of the plate would be perhaps one half of the maximum liquid level. Rotating the shaft at a low speed, say 1 rpm, would keep the inventory mixed.

A second approach might be a perforated plate, with a diameter slightly less than the diameter of the tank. The plate would vertically traverse the depth of the inventory once every few minutes. Forcing a portion of the inventory to pass from the top of the plate to the bottom, and vice versa, would promote convection mixing at the perimeter of the tank.

It can be noted that the two approaches to whole-tank mixing are only conceptual, and additional design work is required to demonstrate that the concepts will work as intended. Nonetheless, the calculation of the floor stresses produced by radial temperature gradients (Section 5.7) and by circumferential temperature gradients (Section 5.11.2) indicate that intra-tank temperature differentials should be held to values no higher than perhaps 5 °C. It is unlikely that the static mixing generated by either one or two inlet distribution headers will satisfy a 5 °C limit, particularly at low flow rates or following a trip of the receiver. As a result, some form of active mixing device may be mandatory to protect the tank.

Friction Forces with a Tank Scale Mixing Device It can be noted that friction forces can develop in the floor, even with if the inventory temperatures are always isotropic. The effect occurs during startup of the receiver, as discussed in *Friction Forces*. As such, the use of a tank scale mixing device can help to control, but not fully eliminate, transient thermal stresses in the floor.

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Cold Tank Diversion Valve Failure and Tank Scale Mixing Device On a related topic, the temperature distribution in the hot salt tank is determined by 1) conditions where the equipment (valves, instruments) operates as intended, and 2) conditions where the equipment does not operate as intended. One example of the latter is the following scenario:

- At the start an operating day, the hot tank is at the minimum level
- The receiver, after reaching normal operating conditions, trips. The receiver pumps are tripped. The circuit inlet control valves move to the 100 percent open position, and cold salt is supplied to the receiver from the pressurized inlet vessel
- The heliostat field, except for a limited number of heliostats providing a preheat flux, is defocused. The defocusing period requires about 60 seconds. During this period, the air pressure in the inlet vessel, and the salt flow rate to the receiver, follow an exponential decay
- The diversion valve in the downcomer, which directs flow to the cold salt tank, then malfunctions (fails to open). The temperature of the salt entering the hot tank decays from an initial value of 565 °C to the cold salt temperature of 295 °C at the projected rate of 360 °C/min.

The flow through the receiver is then stopped, and the receiver is drained. The temperature of the inventory in the hot tank, assuming perfect mixing, is shown in Figure 5-24.

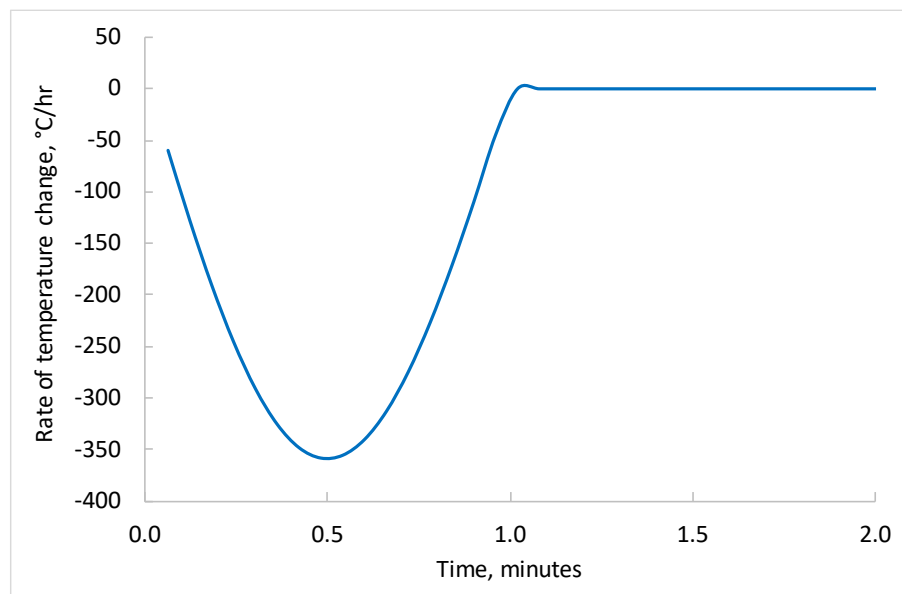


Figure 5-24 Hot Tank Temperature Following a Failure of the Cold Tank Diversion Valve

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The peak cooling rate of $-350\text{ }^{\circ}\text{C/hr}$ is approximately 6 times the typical maximum rate allowed by the tank supplier. The stresses associated with this scenario have not been calculated, but local stresses above the yield value, and some loss in the fatigue life, would not be unexpected.

The failure of the cold tank diversion valve represents a single point failure, and would not be considered an improbable condition. As such, the design basis for the project should include this scenario, together with other potential single point failures, in an analysis of the fatigue life. If the analysis shows that damage to the tank can only be prevented (or meaningfully reduced) by some form of active mixing, then a mechanical mixer must be a feature of the tank design.

Friction Forces

During overnight hold periods, the temperature of the salt inventory in the tank decays. The tank thermally contracts, which places the floor in tension. Following the startup of the receiver, the temperature of the salt inventory increases. The tank starts to thermally expand, which 1) reduces the tensile stresses in the floor, and then 2) places the floor into compression. As discussed in Section 5.10.3, an increase in the inventory temperature of about $30\text{ }^{\circ}\text{C}$ is sufficient to change the friction in the floor from the maximum tensile stress to the maximum compressive stress. The magnitude of the tensile or the compressive stress is directly proportional to the height of the salt in the tank.

During the plant design, finite element calculations of the stress in the floor should be developed for various combinations of temperature change, both positive and negative, and salt height. Allowable combinations are defined, such that the floor stresses satisfy the creep and the fatigue requirements. The allowable combinations are programmed into the distributed control system in the form of permissives. The permissives provide two useful functions:

- The operator is not allowed to make ad hoc decisions regarding inventory temperature and level that could have a detrimental effect on the creep and the fatigue life of the floor
- Over the course of a day, the inventory temperatures increase to values as high as possible, consistent with the allowable stresses in the floor. This, in turn, maximizes the daily electric output of the Rankine cycle, without compromising the long-term availability of the storage system.

As discussed in Section 5.10.3, a static coefficient of friction of 0.3 between the floor and the foundation may be high enough to produce yield stresses near the center of the tank. The most common lubricant for commercial tanks is sand. One reference for data on friction coefficients between steel and sand is a

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paper Uesugi²². However, in general, there are little data on the coefficients for sand at high temperatures. If experiments show that the coefficients are greater than about 0.4, then sand may not be a suitable solid lubricant.

An alternative high temperature lubricant may be boron nitride. During construction, the boron nitride would be applied to the top of the foundation material (presumably refractory bricks) as a water/boron nitride suspension. However, there are no data on how long the boron nitride layer might survive. If the layer becomes eroded, it may be possible to renew the lubricant by injecting a water/boron nitride suspension through an array of pipes installed at the top of the foundation. A series of experiments, at the full design temperature of the tank, would need to be conducted to demonstrate the feasibility of this approach.

An alternate solid lubricant is graphite. However, the galvanic potential between carbon and stainless steel is large. Pitting corrosion of the stainless steel is likely, which makes graphite a poor candidate.

Tanks Without Flat Floors

Flat bottom tanks, with self-supporting dome roofs, have been universally adopted in the solar industry due to their low cost. However, this design is not the only option for the industry.

Elevated Tanks

Elevated water storage tanks, in capacities up to 11,300 m³ (3,000,000 gallons), are commercially available from a number of suppliers. To replicate the storage volume at Crescent Dunes (17,500 m³) two tanks, similar to that shown in Figure 5-25, would be required. For a solar project, the tanks would be elevated by only a small amount, perhaps 1 m, above grade.

The tanks are supported either on single column, as shown above, or on a series of legs, attached to the side of the tank. With this arrangement, there are no friction loads on the floor, and many of the structural problems with flat bottom tanks can be avoided.

Naturally, this type of tank has a number of characteristics which will increase the cost relative to a flat bottom tank, as follows:

²² Uesugi, Morimichi, et. al., (Department of Architectural Engineering, Chiba University, Japan), 'Friction Between Sand and Steel under Repeated Loading', Soils and Foundation Volume 29, No. 3, 127-137, September 1989, Japanese Society of Soils Mechanics and Foundation Engineering

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Figure 5-25 11,300 m³ (3,000,000 gallon) Elevated Water Storage Tank

- The vertical loads are taken either at a central location or by a series of columns located around the circumference of the tank. The concentrated loads, relative to a flat bottom tank, will incur a significant increase in the cost of the foundation
- The bottom of the curved tank is exposed to hydrostatic loads which are higher than those seen in a flat bottom tank. This will lead to an increase in the thickness, and the cost, of the bottom of the tank.

Nonetheless, the cost of the hot salt tank in a solar project is on the order of 2 to 3 percent of the cost of the project. If switching from a flat bottom tank to an elevated tank doubled the cost of the tank, then the net change to the project is perhaps a 2 to 3 percent increase in the capital cost. However, if the availability of the project improves by, say, even 5 percent due to the avoidance of a tank failure, then the increase in the cost of the tank is fully justified.

Horizontal Tanks

The lowest risk approach is, perhaps, a field of horizontal cylindrical tanks. The tanks are supported on saddles, which avoids any problems with friction. Further, the geometry is favorable in terms of accommodating both high rates of temperature change and non-uniform temperature distributions in the inventory.

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Horizontal tanks are available from a number of commercial suppliers. The maximum practical diameter and length are on the order of 3.5 m (11 ft.) and 40 m (130 ft.), respectively, based on shipping considerations. To replicate the storage volume at Crescent Dunes (17,500 m³), some 44 tanks would be required.

On a related point, it may be possible to reduce the number of tanks to 24 using an N + 1 arrangement. Specifically, one of the tanks would always be a cold tank, one of the tanks would always be a hot tank, and the balance of 22 tanks would operate either as a cold tank or a hot tank, depending on the state of charge in the storage system. Naturally, switching from cold tank status to hot tank status would result in rapid changes in the metal temperature. Finite element calculations of the transient stresses in the tank would first need to be conducted to verify the practicality of this approach.

Returning to the original estimate of 44 tanks, the mass of steel in 44 horizontal tanks is about 2.8 times the mass of steel in a single, flat bottom tank, and a cost comparison of the two options would reflect this difference. The surface area of 44 horizontal tanks is slightly more than 6 times the surface area of a single, flat bottom tank. It would be prohibitively expensive to insulate 44 horizontal vessels, particularly if the cost of the heat loss through the insulation was considered in the analysis. However, it may be possible to locate the 44 vessels in a common enclosure, and then insulate the outside of the enclosure.

To a first order, replacing a single, flat bottom hot tank with 44 horizontal tanks is likely to triple the cost of the hot tank. However, as noted above, the cost of the hot tank is 2 to 3 percent of the cost of the project. If the cost of the tank triples, then the cost of the project increases by something approaching 6 percent. However, if the flat bottom tanks are judged to be unsuitable for service as a hot tank, then the maximum penalty incurred by switching to a low risk design is on the order of 6 percent.

5.15 Lifetime Assessment

The material in a tank is subjected to cyclic stresses, environmental effects (i.e., corrosion), and continuous stresses. The life of the material, N_f , is a function of the accumulated damage due to fatigue, corrosion, and creep, and can be represented as follows:

$$\frac{1}{N_f} = \frac{1}{N_f^{Fatigue}} + \frac{1}{N_f^{Oxidation}} + \frac{1}{N_f^{Creep}}$$

A range of considerations in evaluating the fatigue, the corrosion, and the creep lives of the tank are outlined in the sections below.

5.15.1 Fatigue Assessment

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The number of stress (or strain) cycles that a material can tolerate before fracture is a function of the material, the temperature, and the range of the stress (or strain). Fatigue data are typically generated for smooth bar samples, with a nominal cyclic strain rate of 0.001 in/in-sec.

Representative S-N (Stress-Number of cycles to failure) curves for Type 316 stainless steel are shown in Figure 5-26.

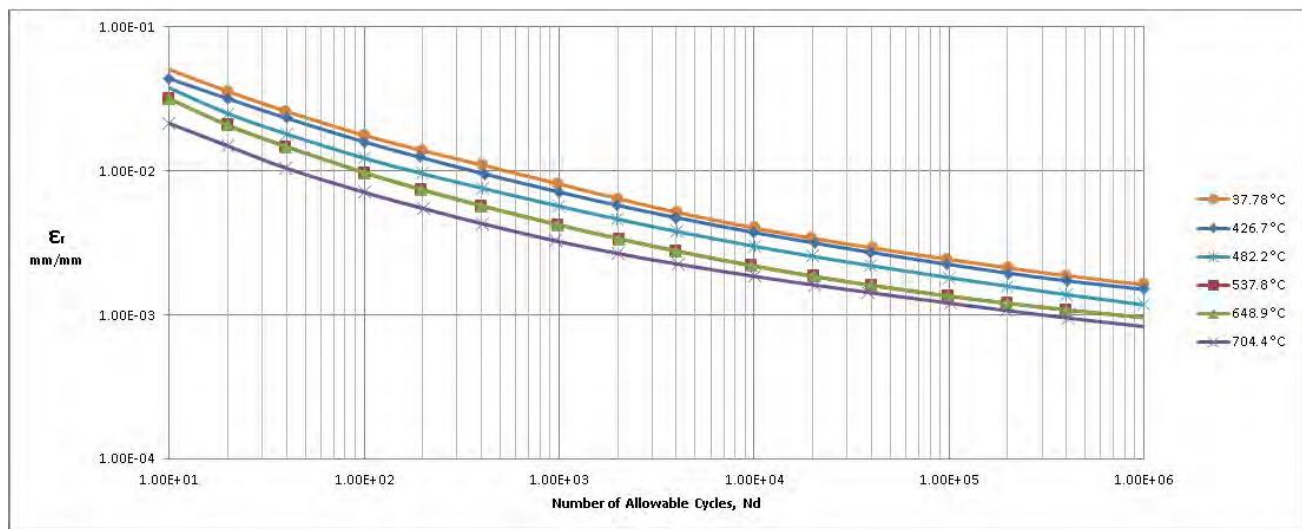


Figure 5-26 S-N Curves for Type 316 Stainless Steel

Nonetheless, and as might be expected, the material in a salt tank, particularly after welding, has markedly different fatigue characteristics than a smooth bar sample. The factors which influence the fatigue life include weld surface fatigue strength reduction factors, weld misalignment fatigue strength reduction factors, and fatigue life knockdown factors.

Weld Surface Fatigue Strength Reduction Factors

When a metal is welded, changes occur in the material properties, the grain structure, and state of stress. As might be expected, the fatigue properties in the weld region are generally inferior to the fatigue properties of the parent material. The fatigue properties of a weld, in turn, are influenced by the type of the weld, the surface condition, and the non-destructive examination methods. The effects are incorporated in the fatigue analysis by a fatigue strength reduction factor. The factor is a stress intensification value which accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. The factor is the ratio of the fatigue strength of a component without a discontinuity or weld joint to the fatigue strength of that same component with a discontinuity

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or weld joint. The factor is a multiplier on the alternating stress value used in the S-N curves; it is not a reduction factor on the fatigue life.

Fatigue strength reduction factors for a range of weld surface conditions are presented in Table 5-2. The weld quality levels are defined in Table 5-3.

Table 5-2 Weld Surface Fatigue Strength Reduction Factors - Table 1 of 2

Weld Condition	Surface Condition	Quality Levels (see Table 5.12)						
		1	2	3	4	5	6	7
Full penetration	Machined	1.0	1.5	1.5	2.0	2.5	3.0	4.0
	As-welded	1.2	1.6	1.7	2.0	2.5	3.0	4.0
Partial Penetration	Final Surface Machined	NA	1.5	1.5	2.0	2.5	3.0	4.0
	Final Surface As-welded	NA	1.6	1.7	2.0	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0
Fillet	Toe machined	NA	NA	1.5	NA	2.5	3.0	4.0
	Toe as-welded	NA	NA	1.7	NA	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0

In commercial tanks, the floor plates are typically welded using backing strips. The backing strips aid in the alignment of the welds between plates, which helps to reduce the fatigue strength reduction factors associated with weld misalignments (see Section *Weld Misalignment* below).

Code Guidelines

Section NH-3352(c) in Section III states *When Category B joints with backing strips not later removed are used, the suitability for cyclic operation shall be analyzed using a fatigue strength reduction factor of not less than 2 under conditions where creep effects are insignificant [NH-3211(c)]. For conditions where creep effects are significant, see NH-3353(b).* Category B joints are butt joints, which is the typical weld joint in a floor.

Further, Section NH-3352(e) states *Where partial penetration welds are used, a fatigue strength reduction factor of not less than 4 shall be used for any related fatigue and analysis under conditions where creep effects are insignificant [NH-3211(c)]. For conditions where creep effects are significant, see NH-3353(b).* Anecdotal evidence from the postmortem examinations of floors in commercial projects show evidence of misaligned welds with incomplete penetration.

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As discussed in the section of this report titled *Comparison of Section VIII Division 2 with Section III Subsection NH*, creep effects in a hot salt tank are not likely to be insignificant. Along these lines, Section NH-3353(b) states *For meeting the analysis requirements of NH-3251 at elevated temperature weld regions, the assumed weld surface should model the most severe strain concentrations expected in the actual weld placed in service. This geometry may be prescribed on a drawing or may be recorded by prior observation. Prior observations of weld surface geometry can be visual; remote visual (e.g., using a borescope device or making a surface replica); or ultrasonic, based on a weld mockup test in which the same weld procedures are used on the same nominal pipe diameter and wall thickness, or based on a radiographic technique that is suitable for inspection of internal surfaces.* (Section NH-33353(b) references *Welded Joints*, and the Section is likely applicable to joints other than piping connections.)

Table 5-3 Weld Surface Fatigue Strength Reduction Factors - Table 2 of 2

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MT/PT examination and a VT examination.
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MT/PT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MT/PT examination and VT examination
1.6	2	As-welded weld that receives a partial volumetric examination, and a surface that receives MT/PT and VT examination
1.5	3	Machined or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
1.7	3	As-welded or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
2.0	4	Weld has received a partial or full volumetric examination, and the surface has received VT examination, but no MT/PT examination
2.5	5	VT examination only of the surface; no volumetric examination nor MT/PT examination.
3.0	6	Volumetric examination only
4.0	7	Weld backsides that are non-definable and/or receive no examination.
Notes: 1. Volumetric examination is RT or UT in accordance with Part 7. 2. MT/PT examination is magnetic particle or liquid penetrant examination in accordance with Part 7 3. VT examination is visual examination in accordance with Part 7. 4. See WRC Bulletin 432 for further information.		

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With floor plate thicknesses in the range of 6 to 8 mm, two weld passes would be typical. In terms of weld examination, the most problematic location would be the bottom of the root pass; i.e., the weld surface which is in contact with the top of the weld backing plate. Two problems can occur:

- Currently, there is no mechanism to visually examine the surface of this weld, looking for signs of the following:
 - Incomplete penetration or fusion
 - Oxidation. The quality of the weld will depend, in part, on the ability of the weld backing strip to capture and retain the cover gas (argon) in the weld zone. Any oxygen present in the weld zone can cause the filler material and the edges of the heat affected zone to oxidize. The result can be a weld with a high porosity and a low strength.
- There is a long total length of the welds to examine in a floor. Common non-destructive examinations typically include dye penetrant and vacuum box tests due to their speed and low cost. These examination methods can result in a low value for the weld quality presented in Table 5-3.

Based on these assessments, a weld surface fatigue strength reduction factor in the range of 2.5 to 4.0 can likely be justified.

Weld Misalignment Fatigue Strength Reduction Factors

Steel plates for the tank floor are typically fabricated to nominal dimensions for length, width, thickness, and flatness. Allowable tolerances on the dimensions are defined in ASTM A480, Standard Specification for General Requirements for Flat-Rolled Stainless and Heat-Resisting Steel Plate, Sheet, and Strip. For a floor plate which has a width of 2.5 m, a length of 12.0 m, and a thickness of 8 mm, the allowable tolerances include the following:

- Variation in thickness of -0.00 mm to +1.30 mm
- Panels cut by shearing
 - Variation in width of -0 mm to +20 mm
 - Variation in length of -0 mm to +75 mm
- Panels cut by water jet
 - Variation in width of -0 mm to +3.2 mm
 - Variation in length of -0 mm to +3.2 mm
- Variation in flatness of 14 mm.

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When the panels are arranged on the foundation prior to welding, the tolerances will result in the following effects:

- The tops of all of the plates will not all be in the same horizontal plane
- The bottoms of all of the plates will not all be in the same horizontal plane
- The edges of the plates will not all be parallel.

As a consequence, some level of weld misalignment is inevitable.

The effects of weld misalignment on the fatigue life of a structure are discussed in Part 8, Assessment of Weld Misalignment and Shell Distortions, in API 579-1 / ASME FFS-1. The effects are calculated using the following expression:

$$\Delta S_p = \sigma_{ms} (1 + R_b^{pc} + R_b^{pa}) K_f$$

where ΔS_p is the range of primary plus secondary plus peak equivalent stress, MPa
 σ_{ms} is the membrane stress from supplemental loads, MPa
 R_b^{pc} is the adjustment factor for centerline misalignment, []
 R_b^{pa} is the adjustment factor for angular misalignment, []
 K_f is weld fatigue strength reduction factor, [].

Supplemental loads include, but are not limited to, the weight of the component, contained fluid, insulation or refractory; loads resulting from the constraint of free thermal expansion, thermal gradients or differences in thermal expansion characteristics; occasional loads due to wind, earthquake, snow, and ice; loads due both to environmental and operating conditions; reaction forces from fluid discharges; loads resulting from support displacements; and loads due to process upset conditions.

Equations for calculating the R_b factors are listed in Table 5-4.

Centerline Misalignment

Centerline misalignment in flat plates with a butt weld is defined in Figure 5-27. 'P' is the applied load, 't' is the plate thickness, and 'e' is the weld offset.

Angular Misalignment

Angular misalignment in flat plates with a butt weld is defined in Figure 5-28. 'P' is the applied load, and Θ_p is the angular misalignment. The dimension 'L' is defined in Figure 5-29.

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Table 5-4 Equations for Calculating the R Factors in the Equation for ΔS_p

Type Of Misalignment	Equations For R_{bs}
Flat Plate – Centerline Offset (see Figure 8.2) (1)	$R_{bs}^{pc} = 6 \left(\frac{e}{t_c} \right) \left(1 + \left(\frac{t_{2c}}{t_c} \right)^{1.5} \right)^{-1}$ Limitations: $0.0 \leq e \leq t_{1c}, t_{2c} \geq t_{1c}$
Flat Plate – Angular Misalignment (see Figure 8.4) (1)	$R_{bs}^{pa} = 3 \left(\frac{\delta}{t_c} \right) C_f$ where: $\delta = \frac{L\theta_p}{4} \quad (\theta_p \text{ in radians})$ $C_f = \left(\frac{2}{\beta} \right) \cdot \tanh \left[\frac{\beta}{2} \right] \quad (\text{for fixed ends})$ $C_f = \left(\frac{2}{\beta} \right) \cdot \tanh [\beta] \quad (\text{for pinned ends})$ $\beta = \frac{L}{t_c} \sqrt{\frac{3\sigma_m}{E_y}} \quad (\text{in radians})$ Limitations: $\theta_p \geq 0.0$
Note: The equation for R_b and R_{bs} are dimensionless.	

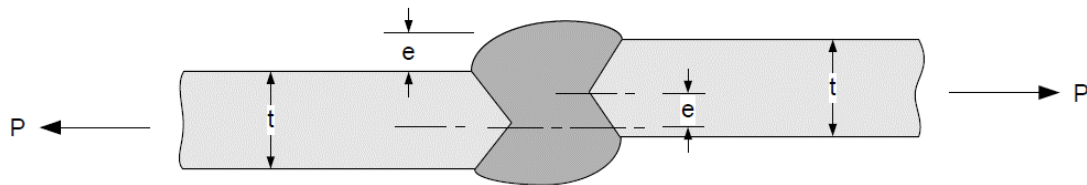


Figure 5-27 Centerline Offset Misalignment of Butt Welds in Flat Plates

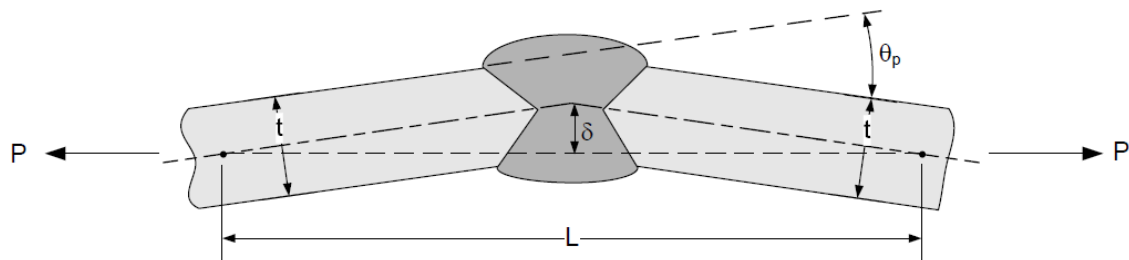


Figure 5-28 Angular Misalignment of Butt Weld Joints in Flat Plates

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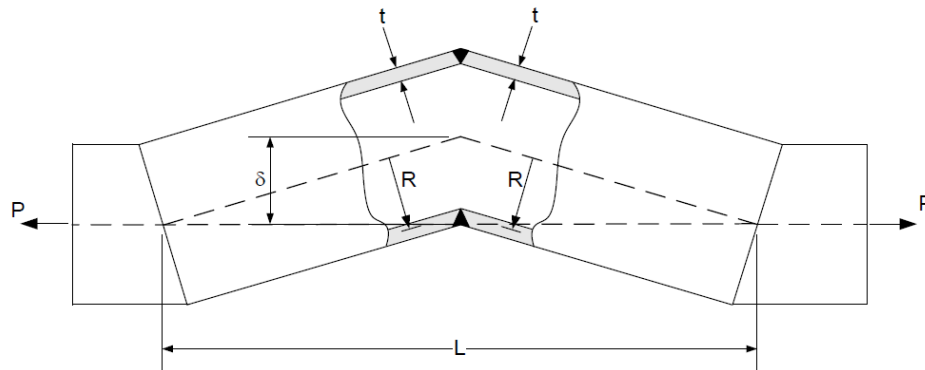


Figure 5-29 Definition of Dimension L in Figure 5-28

The calculated value for R_b^{pc} , the adjustment factor for centerline misalignment, for a range of offsets is shown in Figure 5-30.

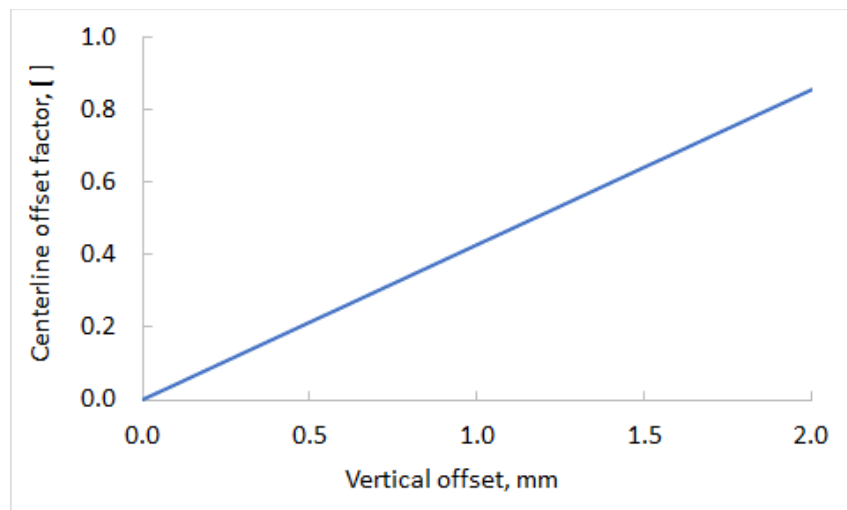


Figure 5-30 Adjustment Factor for Centerline Misalignment

The calculated value for R_b^{pa} , the adjustment factor for angular misalignment, for a range of angles is shown in Figure 5-31. For the purposes of the example, the sum of the supplemental loads is taken to be 200 MPa.

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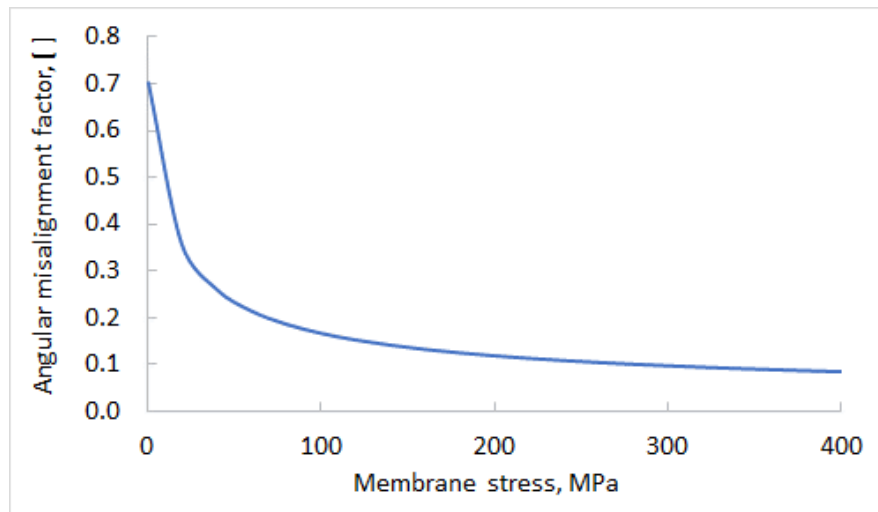


Figure 5-31 Adjustment Factor for Angular Misalignment

Representative Misalignment Factors

The centerline offsets and the angular misalignments in the tank in a commercial project likely span a fairly broad, and perhaps random, set of values. For the purposes of this discussion, the minimum centerline offset between adjacent plates is judged to be not less than 0.5 mm (1/64 in.). This is particularly so if the plate, as delivered, 1) deviates from flat by the maximum allowable value of 14 mm, and 2) the plate must be mechanically pushed flat against the foundation prior to welding. If the centerline offset is 0.5 mm, then the fatigue adjustment factor is on the order of 0.2.

Defining an angular misalignment is somewhat more problematic. With the plates arranged on the foundation, prior to welding, the angular misalignment should be zero. To start the welding process, the plates are tack-welded to one another, but this should not influence the angular misalignment. However, when Plate 1 is welded to Plate 2, the two plates start to form a domed shape. If the height of the dome, after the welding is complete, is 100 mm (see Section 5.5.2), then the out-of-plane angle at the edge of the plate is approximately $\tan^{-1}(100 \text{ mm}/1,250 \text{ mm}) = 4.5$ degrees, where 1,250 mm represents one-half of the width of the plate. When the tank is initially filled with salt, hydrostatic loads push the plates flat against the foundation. The bending stresses at the edges of the plates are equal to, or somewhat higher, than the yield stress. As such, most of the 4.5 degree angular misalignment disappears. However, some residual value of the stress will remain, which will retain some level of bending displacement at the edge. For purposes of the discussion, the residual bending misalignment is taken to be 0.25 degrees. If the stresses in the floor are in the range of 100 to 200 MPa, then the value for the angular misalignment factor is approximately 0.15.

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With these representative factors, the weld fatigue strength reduction factor selected above in the Section titled *Weld Surface Fatigue* Strength Reduction Factors is further multiplied by a factor of $(1 + 0.2 + 0.15) = 1.35$. This would result in composite reduction factors in the range of $1.35 * 2.5 = 3.4$ to $1.35 * 4.0 = 5.4$.

It can be noted that increasing the thickness of the floor reduces the centerline offset misalignment factor (assuming that the vertical offset is maintained at 0.5 mm). However, the angular misalignment factor remains unchanged with a thicker floor.

Fatigue Strength Knockdown Factors

For areas not adjacent to welds, the fatigue life of commercial equipment is likely to be shorter than the fatigue life measured in smooth bar tests. The effect on the fatigue life, in areas not adjacent to welds, can be estimated using fatigue life knockdown factors. Several of these factors are outlined in Table 5-5. Unlike the fatigue strength reduction factors discussed in the sections above, the knockdown factor is a divisor on the smooth bar fatigue life, rather than a multiplier on the alternating stress.

The size effect factor accounts for equipment which has larger dimensions, and therefore has a shorter fatigue life, than the small samples typically used to develop fatigue data. Similarly, the surface finish effect accounts for equipment surfaces which are rougher, and therefore have a shorter fatigue life, than smooth bar samples. The data scatter factor accounts for the statistical nature of the smooth bar fatigue curves, and provides a degree of conservatism in the estimated life of the commercial equipment. Environmental effects are discussed below in the Section 5.15.2.

Using the factors in the column labeled ASME results, and an allowance for a surface finish factor of 1.5, results in an overall knockdown factor of $2.5 * 1.5 * 2 = 7.5$.

5.15.2 Corrosion Assessment

Stainless steel, when exposed to salt, develops an outer corrosion layer. The layer consists of various nickel and iron oxides. The corrosion layer controls the principal corrosion mechanism for stainless steel, which is the diffusion of chromium from the metal into the salt.

In corrosion studies conducted at Sandia and at other laboratories, the corrosion coupons are typically free of internal and external stresses. However, under cyclic conditions, corrosion layers can develop when the material is at a high temperature and under compressive stress. During the balance of the cycle, the material can cool and the compressive stress can switch to a tensile stress. This can form microcracks in the oxide layer, which expose the parent material to an accelerated corrosion rate due to thinning, or the local removal, of the oxide layer.

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Table 5-5 Fatigue Life Knockdown Factors

Item	ASME	PVRC Working Group on S-N Data		PVRC Working Group on Evaluation Methods	
		Best Estimate Of Margin Needed	Conservative Estimate Of Margin Needed	Best Estimate Of Margin Needed	Conservative Estimate Of Margin Needed
Size Effect	2.5	1	1.2	1.0	1.2
Surface Finish	4 (1)	1.5	2	1.0	1.5
Data Scatter	2	2	2.5	2.0	2.5
Environmental Effects	4 (1)	2.5	4	---	---
Loading time history Effects	NC	1	1	1.0	2.5
Multi-Axial Stress Field	---	---	---	1.0	1.3
Sum	20	7.5	24	2.0	14.6
Notes: 1. Surface finish and environmental combined with a margin of 4. 2. From WRC 487, Table 1.					

This corrosion mechanism could occur in the floor of a hot tank due to a combination of 1) receiver start and stop cycles, 2) daily reversals in friction direction, and 3) changes in the stresses due to changes in the radial temperature gradient. If the floor is operating at stresses above than the yield stress, then the associated strains could be large enough to damage the outer oxide layer. This could change the shape of the corrosion rate-versus-time curve from parabolic (desired) to linear (not desired).

In general, the magnitude of this potential effect is largely unknown. Only after one or more commercial tanks have been in service for several years, and the inner surfaces examined for signs of non-uniform corrosion layers, can some form of an assessment be made.

5.15.3 Creep Assessment

WRC Bulletin 541

The Welding Research Council publishes Bulletin 541, Evaluation of Material Strength Data for Use in API Standard 530. Standard 530, in turn, is the Calculation of Heater-tube Thickness in Petroleum Refineries. The methods described in Bulletin 541 optimize the Larson-Miller Parameter (LMP) stress-rupture coefficients of the materials listed in API Standard 530.

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The time to rupture, L [hours], is estimated by the following equation:

$$\log_{10}[L] = \frac{LMP(\sigma)}{T} - C_{LMP}$$

where LMP is the Larson Miller parameter corresponding to the applied stress

σ is the applied stress, MPa

T is the absolute temperature, °K

C_{LMP} is a constant for the material, []

The stresses associated with times to rupture in the range of 1,000 hours to 1,000,000 hours are shown in Figure 5-32. The nominal life of the hot salt tank is 30 years, or 300,000 hours. The associated stress for both Type 316 / 316H and Type 347 H is a nominal value of 134 MPa.

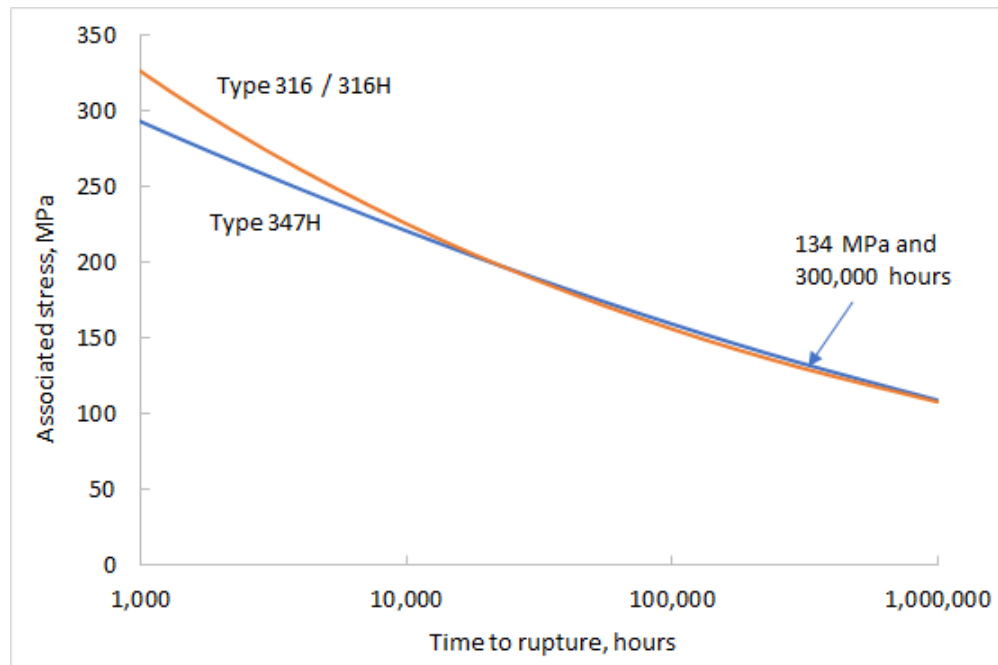


Figure 5-32 Bulletin 541 Time to Rupture and Stress For Type 316/316H and Type 347H

To a first order, any permanent stress in the tank cannot exceed a value of 134 MPa; otherwise, that portion of the tank can fail due to rupture.

Project Omega

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The assessment techniques developed under the Materials Properties Council (MPC) Project Omega program provide a method for estimating the remaining life of a component operating in the creep regime. A strain-rate parameter and multi-axial damage parameter (Omega) are used to predict the rate of strain accumulation, creep damage accumulation, and remaining time to failure as a function of stress state and temperature.

The equations are presented in Annex 10B - Material Data for Creep Analysis (Normative) in API 579-1 / ASME FFS-1. Equation coefficients for Type 347H are listed in Table 10B.1M, MPC Project Omega Creep Data (MPa, °C).

The stresses associated with times to rupture in the range of 1,000 hours to 1,000,000 hours are shown in Figure 5-33. With a creep rupture life of 300,000 hours, the associated stress for Type 347 H is a nominal value of 127 MPa.

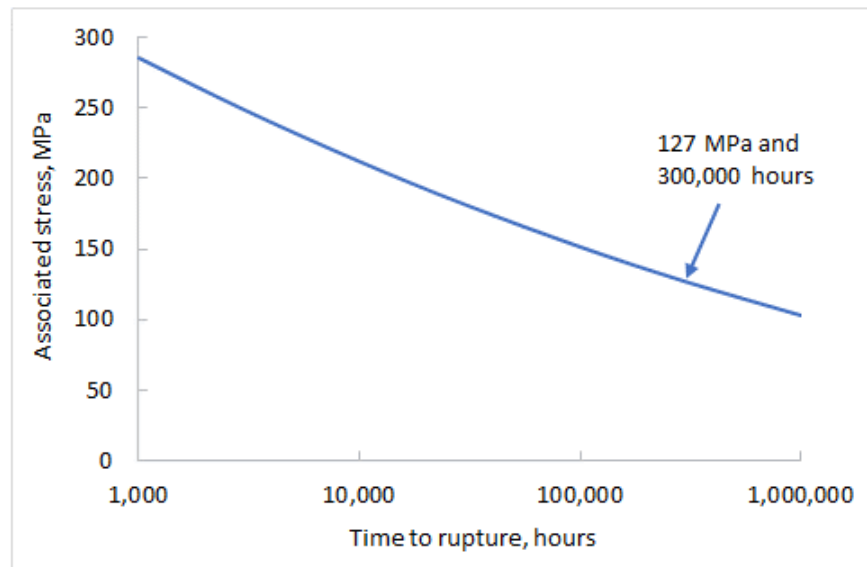


Figure 5-33 Project Omega Time to Rupture and Associated Stress For Type 347H

The permanent stress in the tank cannot exceed a value of 127 MPa if the creep rupture life is to exceed 30 years.

5.15.4 Subset of Creep Assessment in Weld Regions

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The coefficients for the Larson Miller equation ($LMP(\alpha)$, C_{LMP}) in WRC Bulletin 541 and in Project Omega are based on materials in the as-received condition; i.e., prior to welding. The value for C_{LMP} is typically taken as a constant for the material.

The value for C_{LMP} is obtained by plotting experimental creep data in $\log(\text{time [hours]})$ versus $1/(\text{Absolute Temperature[K]})$. An example for an austenitic stainless steel is shown in Figure 5-34²³.

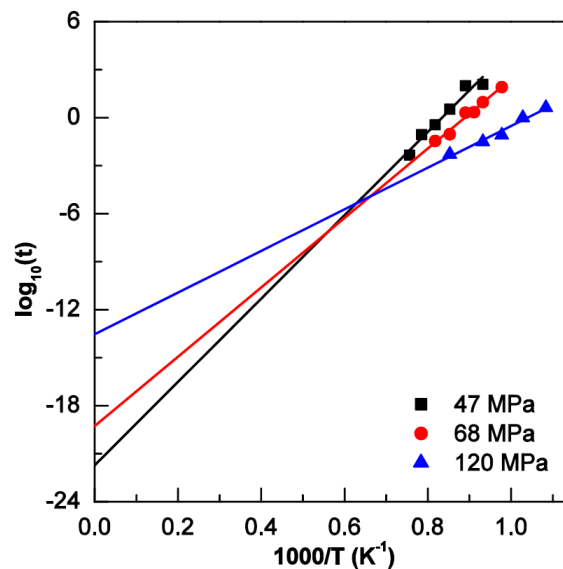


Figure 5-34 Graphical Calculation of Larson Miller Coefficient C_{LMP}

As noted in the Figure, the value for C_{LMP} is not a constant; it is a function of the applied stress.

During welding, residual stresses are developed in the weld region. The stresses are a function of the weld filler material, the heat input to the weld, the number of passes, and the location in the weld zone. The last item incorporates the fusion zone, the partially melted zone, and the heat affected zone.

The creep life will be a function of the sum of the operating stresses and the residual welding stresses. This implies that the Larson Miller coefficients will vary among the fusion zone, the partially melted zone, the heat affected zone, and the zones which are not welded. Further complicating the calculation of the creep life is the expected relaxation of the residual welding stresses over time, as shown in Table 5-1 Stress Relief as a Function of Temperature.

²³ Ghatak, A., and Robi, P. S. (Department of Mechanical Engineering, North Eastern Regional Institute of Science and Technology, Nirjuli, India), Modification of Larson–Miller Parameter Technique for Predicting Creep Life of Materials, Transactions of the Indian Institute of Metals, DOI 10.1007/s12666-015-0803-6, December 2015

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To account for the variation in C_{LMP} with stress, the authors in the referenced paper propose the following modified equation to estimate the creep lifetime:

$$\log_{10}[L] = \frac{LMP(\sigma)}{T} - C_{LMP1} - C_{LMP2}\sigma$$

where LMP is the Larson Miller parameter corresponding to the applied stress

σ is the applied stress, MPa

T is the absolute temperature, °K

C_{LMP1} and C_{LMP2} are constants for the material, []

Ideally, different values for $LMP(\sigma)$, C_{LMP1} , and C_{LMP2} would be developed for the fusion zone, the partially melted zone, the heat affected zone, and the zones which are not welded. However, the degree to which the ideal can be approached has yet to be determined. Specifically, the various coefficients are developed experimentally. Further, the accuracies of the coefficients are improved if the experiments are conducted at temperature representative of a commercial project; i.e., 600 °C. However, at these temperatures, creep lifetimes can be the order of tens of years if the stresses are modest (< 150 MPa). As a result, the uncertainties in the estimates of creep lifetime, particularly in the weld zones, are likely to remain high for at least the next generation of commercial projects.

5.15.5 Subset of Creep Assessment in Buckled Regions

In several commercial projects, the tank floors have developed buckles, with the buckles largely spanning the diameter of the tank. For the purposes of discussion, the thickness of the floor is taken to be 7 mm, and the inner radius of the plate at the top of the buckle is assumed to be 50 mm. Bending the floor in this shape generates a tensile strain of 0.065 on the top surface of the buckle, and a compressive strain of - 0.065 on the bottom surface of the buckle.

Using information from Section VIII, Division 2, Annex 3.D, for 300-series stainless steel at a temperature of 565 °C, a strain of 0.065 represents a true stress of approximately 260 MPa, as shown in Figure 5-35. The true stress and the true strain values include the effect of strain hardening.

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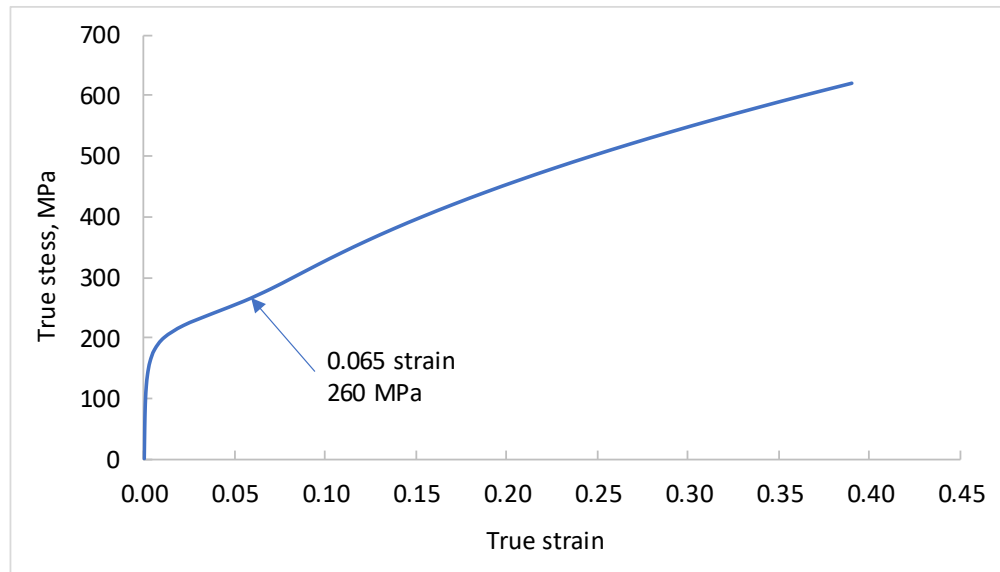


Figure 5-35 True Stress as a Function of True Strain for Type 347H at 565 °C

It should be noted that the equations in Annex 3.D are applicable only for temperatures up to 482 °C. As such, the calculated stress of 260 MPa likely incurs some error. However, for the purposes of the discussion, the estimated stress should be the correct order of magnitude.

The inventory temperatures during commercial operation are well below the temperatures at which significant stress relaxation occurs (800 to 900 °C). As such, the calculated stress of 260 MPa is likely to persist for the life of the tank.

Using information for Larson Miller parameters from API 579-1 / ASME FFS-1, Table 10B.4M, the creep life as a function of stress for Type 347H at 565 °C can be calculated, as shown in Figure 5-36.

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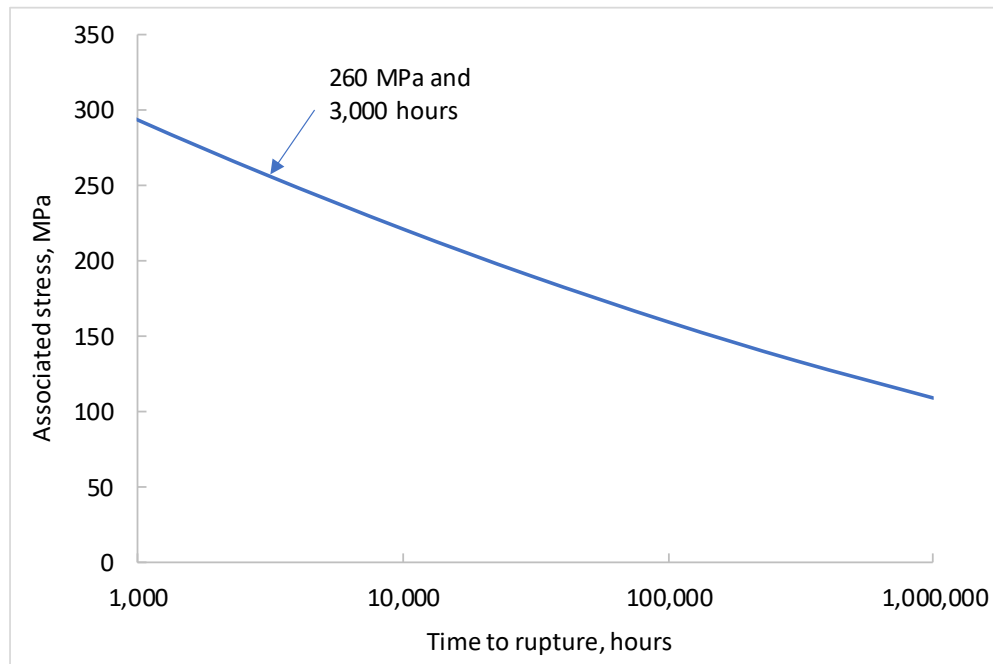


Figure 5-36 Time to Rupture as a Function of Stress for Type 347H at 565 °C

The calculated time to rupture is only several months if the tank is always operating at a temperature of 565 °C. However, in commercial projects, the temperature of the inventory is typically less than 565 °C due to the following effects:

- During overnight hold periods, the inventory level is typically low, and the temperature of the tank can decay by values in the range of 5 to 10 °C
- During receiver startup, salt is introduced into the tank at temperatures in the range of 500 to 565 °C.

If the annual average inventory temperature is 520 °C, rather than 565 °C, then the estimated creep life increases to some 26,000 hours. This period is a nominal 3 years, which is roughly consistent with the crack development times experienced at commercial projects.

Clearly, the above calculations are only a zeroth-order analysis. The shape and the dimensions of the floor buckles in commercial projects, if present, are unique to each project. Further, the stresses in the buckled regions are also influenced by friction forces, thermal stresses generated by radial temperature gradients, and bending loads produced by hydrostatic loads. As such, it is difficult to accurately predict the creep life of a buckled region in the floor. Nonetheless, if the required creep life of the floor is 300,000 hours (30 years), then the stresses in the floor must be limited to about 127 MPa, as noted above in Section 5.15.3. A stress of 127 MPa is produced by a strain of 0.0015, which would be produced by a

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‘buckle’ with a radius of about 2,400 mm. This represents a rather gentle curvature in the floor, which implies that even small permanent deformations of the floor could result in a creep life less than required.

5.15.6 Creep Assessment with Consideration of Residual Welding Stresses

Estimates of creep lifetimes can be developed using the methods outlined in Section 5.15.3; i.e., WRC Bulletin 541 and Project Omega. The coefficients in the equations are typically generated for materials as supplied from the mills; i.e., prior to cutting, fabrication, or welding.

When the floor plates are welded using the standard 347 filler material (E347-16 for SMAW, or ER347 for GTAW), residual welding stresses can reach peak values in the range of 350 to 500 MPa²⁴. These values fall between the yield stress and the rupture limit at ambient temperature.

It can be noted that different weld fillers are being investigated as an alternative to 347 to reduce residual stresses in this application. A common choice is ER16-8-2, which is often termed a lean austenitic. The principal benefit to ER16-8-2, compared with 347, is an increase in ductility, which allows a portion of the residual welding stresses to relax. However, qualifying filler materials can involve months to years of laboratory testing to verify the long term mechanical, corrosion, and chemical properties.

With a stress of 500 MPa, the estimated time to rupture for Type 347H stainless steel at a temperature of 25 °C is billions of years.

During preheating, the temperature of the tank is raised to approximately 300 °C. With a stress of 500 MPa, the creep life is reduced, but it is still in the range of thousands of years.

During commissioning, the temperature of the tank first reaches its design value of 565°C. With a stress of 500 MPa, the creep life is predicted to be only 5 hours. Clearly, tanks in commercial service have creep lifetimes longer than 5 hours, which means that a significant level of stress relaxation is occurring between the end of fabrication and the start of commercial service.

5.15.7 Relaxation of Residual Stresses Prior to Tank Commissioning

Tanks in commercial service are exhibiting lifetimes of at least 3 years. The sources of the failures are many and varied. Nonetheless, it is apparent that the creep life must exceed 3 years, as the tank failures to date are showing failure mechanisms other than creep.

²⁴ Osorio, J. (National Renewable Energy Laboratory, Golden, Colorado), Failure Analysis for Molten Salt Thermal Energy Tanks for In-Service CSP Plants, DOE Award Number DE-EE00038475

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As a bounding exercise, for Type 347H stainless steel at a temperature of 565 °C, the stress cannot exceed 200 MPa if the material is to reach a creep life of 26,000 hours (3 years). This implies that some form of stress relaxation is occurring, early in the commercial life of the tank, which reduces the peak stress by a nominal 60 percent; i.e., from 500 MPa to 200 MPa.

Residual stresses can be relaxed by thermal mechanisms, mechanical mechanisms, or a combination of the two.

Thermal Mechanisms

An increase in the metal temperature results in a decrease in the yield stress. When the yield stress decreases to the residual stress, the material plastically deforms. This, in turn, relieves a portion of the residual strain and relieves a portion of the associated stress. The degree of stress relaxation is a function of temperature and time, with the former the principal determinant.

The largest degree of relaxation occurs during solution annealing, when the metal is heated to about 1,050 °C. The residual stresses are reduced by some 95 percent; i.e., from 500 MPa to 25 MPa²⁵. However, prior to the start of commercial service, tank temperatures are limited to a nominal 300 °C during preheating and initial filling. At 300 °C, the material undergoes only stress relief, and relaxation of the residual stresses is limited to about 30 percent (from 500 MPa to 350 MPa).

At a temperature of 300 °C, and at a stress of 350 MPa, the creep life is measured in billions of hours. As such, any creep damage which occurs during preheating can safely be ignored.

During commissioning, the temperature of the tank is raised from the preheat value of 300 °C to the design value of 565 °C. At a temperature of 565 °C, and at a stress of 350 MPa, the creep life is only 200 hours. Since the tanks in commercial projects are not suffering creep failures at 200 hours, additional relaxation of the residual stresses must be occurring during the heating period from 300 °C to 565 °C.

At temperatures in the range of 550 to 650 °C, the degree of stress relief is on the order of 35 percent. This implies that the peak residual stress is reduced to approximately 230 MPa (i.e., $0.65 * 350$ MPa) at the start of commercial service. At a temperature of 565 °C, and at a stress of 230 MPa, the creep life is on the order of 7,000 hours. As above, the tanks are not suffering failures due to creep in the first year of service, and additional relaxation of residual stresses must be occurring before the start of commercial service.

Mechanical Mechanisms

²⁵ Peckner, D. and Bernstein, I., Handbook of Stainless Steel, Chapter 4, Table 25, ISBN 0-07049147-X, 1977

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Cyclic loading is a known mechanism for reducing residual stresses.

Late in the tank fabrication process, the floor is welded to the wall. The wall has a radial and a circumferential stiffness that is several orders of magnitude greater than the radial and tangential stiffness of the floor.

The tank, after fabrication but prior to preheating, undergoes some number of daily temperature cycles due to solar heating and nighttime cooling. The temperature of the wall will be out of phase with the temperature of the floor because 1) the heat transfer rate from the wall to the floor is low, and 2) the thermal inertia of the foundation, relative to thermal inertia of the wall, is essentially infinite. As such, the wall will place the floor into a number of compression and tension cycles, with a total of perhaps 100 to 200 cycles, depending on the project schedule. Some level of stress relief can be expected, but developing an accurate estimate is still likely to involve large uncertainties.

Summary

As noted above, commercial tanks have a creep life in excess of 26,000 hours. To achieve this life, the sum of the operating stresses and the residual stresses must be reduced from 230 MPa at the start of commercial service to values no higher than 200 MPa soon after the start of commercial service.

There is some combination of thermal and mechanical mechanisms, as yet undefined, which are at work to reduce the stress. Deciphering these mechanisms, and quantifying their effects, will be an important element in predicting the creep lifetime of the tank.

5.15.8 Required Relaxation of Residual Stresses to Achieve a 30-Year Creep Life

The tank has a design life of 25 to 30 years, which corresponds to 220,000 to 260,000 hours. Due to a range of uncertainties in the estimates of the operating and the residual stresses, a case could be made for some level of conservatism in the calculation of the creep life. As a point of departure, a design life of 300,000 hours could be adopted. At 565 °C, the corresponding allowable stress is 135 MPa.

For a commercial-size tank (35+ m diameter), the operating stresses, exclusive of residual stresses, are already in the range of 100 to 125 MPa. This implies that the residual stresses must be reduced by 75 to 90 percent to reach a creep life of 300,000 hours. However, as noted above, the degree of stress relief early in commercial service is only about 50 percent. Potential, and in some cases only partial, solutions to the problem of stress relief are outlined below.

Comprehensive Approach to Stress Relief

A post weld heat treatment on the tank, at the solution annealing temperature of 1,050 C, will almost completely remove the residual stresses (approximately 25 MPa remaining).

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Practically speaking, solution annealing at a temperature of 1,050 °C, while maintaining the integrity of the tank shell, will be a challenging task. As outlined in the Code, it is preferable to heat the entire vessel in a closed furnace. However, this is not feasible for a large structure such as a tank, and alternative methods must be considered. The tank must either be internally heated, or different parts heated separately before joining, followed by local heat treatment of the remaining joints. In either case, the shell must be protected from damaging thermal gradients throughout the heating and cooling process.

Heat treatment of the wall may be relatively straightforward, as there are specialty contractors for this work. Nonetheless, the strength of the material at the annealing temperature will be quite low, and the wall may need to be temporarily reinforced to prevent plastic deformations.

The more problematic element is the floor. Achieving the required heating rates, the hold times, and the cooling rates may not be feasible due to the presence of the foundation. However, if the floor is elevated during fabrication, the heat treatment process is considerably more practical. Also, with the floor elevated, welding can be conducted from both above and below, which 1) eliminates the need for backing strips, and 2) reduces the heat input to the welds and the associated residual stresses.

It can be noted that in tanks designed to API 650 requirements, the principal function of the floor is to form a liquid boundary; i.e., it is not intended as a structural element. If welding results in changes to the metal chemistry, or the development of residual stresses, these conditions are unlikely to affect the primary function of the floor. As such, post weld heat treating of the floor is not normally performed.

Conceptually, the post weld heat treatment of the wall and the floor might consist of the following steps:

1. The floor, in an elevated position, is welded, and local post weld heat treatments are conducted as the welds are completed. The process would be similar to the post weld heat treatment of pipe welds. (It should be noted that a successful local or global post weld heat treatment is not guaranteed. Stainless steels typically have sufficient uniaxial ductility to accommodate the plastic strains associated with stress relaxation. However, if the material is constrained, as it would be at the edges of adjoining plates, then strains can develop in multiple axes of the heat affected zones. Stainless steels with a high carbon content have a multiaxial failure ductility as low as 1 percent. The creep strain that occurs during the relaxation of the weld residual stresses could be sufficiently large to exceed the multiaxial failure ductility, and cracks could develop in the weld zone.)
2. The bottom course of the wall is assembled and welded.
3. The perimeter of the floor is welded to the bottom of the wall.

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4. A post weld heat treatment is conducted on the bottom course. The heat treatment would incorporate the thick fillet weld connecting the floor to the bottom of the wall.

Heat treating the bottom course thermally expands the wall, which places the floor into tension. As long as the wall and the floor remain in the elastic range, then no new residual stresses will be imposed on the floor.

Approach to Partial Stress Relief

If solution annealing is impractical, it may be possible to achieve a sufficient degree of stress relief by heating the tank to an intermediate temperature, and then holding the tank at this temperature for an extended period.

The intermediate temperature should be one which reduces the yield strength as far as possible, but does not result in fundamental changes to the metal chemistry. For Type 347H stainless steel, a representative value might be 625 °C.

The preheating period for an empty tank (ambient to 300 °C) is nominally 30 days. In principle, the preheating temperature could be extended to 625 °C, and the preheating period could be extended to, as an example, 60 days.

An engineering analysis could be conducted to determine the optimum combination of temperature, time, degree of stress relief, and preheating cost.

Methods to Increase the Creep Life

One method for increasing the creep life is to reduce the tank temperature. For the purpose of the discussion, the tank design temperature is assumed to be reduced by 50 °C from the typical commercial value of 565 °C to a new value of 515 °C. With a creep life of 300,000 hours, the allowable stress increases to 200 MPa. If the operating stress, through careful tank design and process control, can be limited to 100 MPa, then the allowable residual stress can be $200 \text{ MPa} - 100 \text{ MPa} = 100 \text{ MPa}$. This, in turn, requires a stress relief of $(230 \text{ MPa} - 100 \text{ MPa}) / 230 \text{ MPa} = 55$ percent. However, at a temperature of 515 °C, a stress relief of 55 percent is not assured, and the approach constitutes only a partial solution.

An alternate approach for increasing the creep life is to reduce both the tank design temperature and the design life. This necessarily requires the fabrication of replacement tanks during the project. For the purposes of the discussion, the design temperature is set to 515 °C and the design life is set to 10 years (100,000 hours). The allowable stress increases to 230 MPa. If the operating stress can be limited to 100 MPa, then the allowable residual stress can be $230 \text{ MPa} - 100 \text{ MPa} = 130 \text{ MPa}$. This, in turn,

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requires a stress relief of $(230 \text{ MPa} - 130 \text{ MPa}) / 230 \text{ MPa} = 40$ percent. At a temperature of 515°C , a stress relief of 40 percent cannot be guaranteed, and this approach only constitutes a partial solution.

Summary Observations

The estimates above of allowable residual stresses is only approximate because a full understanding of the operating stresses, and the degree of stress relief, in commercial tanks are not fully understood. Nonetheless, it is clear that sum of the operating stresses and the residual stresses could well exceed the allowable stresses associated with a creep life of 30 years. At some point, the solar industry may decide that some form of stress relief must be conducted if the tank is to achieve a creep life of 30 years.

5.15.9 Code Interpretation of Residual Stresses

The creep life is a function of the material, the temperature, and the stress. The last item includes operating stresses due to temperature gradients (self-restraint) and due to friction. However, the Code is somewhat ambiguous regarding the use of residual stresses in the creep analyses.

A review of Fitness-for-Service API 579-1 / ASME FFS-1 shows the following:

PART 9 - ASSESSMENT OF CRACK-LIKE FLAWS

ANNEX 9D - RESIDUAL STRESSES IN A FITNESS-FOR-SERVICE EVALUATION

9D.4.1 Post Weld Heat Treatment

If the type of PWHT cannot be established, the residual stress distribution for the as-welded condition shall be used in the assessment. If the weld joint is in a component operating in the creep range (i.e. long-term operation), then the residual stress based on the PWHT condition may be used in the assessment.

PART 10 – ASSESSMENT OF COMPONENTS OPERATING IN THE CREEP RANGE

10.4.2.1 The Level 1 assessment for a component subject to a single design or operating condition in the creep range is provided below.

a) STEP 1 – Determine the maximum operating temperature, pressure, and service time the component is subject to. If the component contains a weld joint that is loaded in the stress direction that governs the minimum required wall thickness calculation, then 14°C (25°F) shall be added to the maximum operating temperature to determine the assessment temperature. Otherwise, the assessment temperature is the maximum operating temperature.

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14.8.3 Loading Time History

The loading time history showing the historical and assumed future start-up, normal, upset, and shut-down conditions should be reported. In addition, any additional loads and stresses considered in the assessment, e.g. stresses from supplement loads, thermal gradients and residual stresses.

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10.5.1.4 A sensitivity analysis should be performed as part of the assessment regardless of the method chosen.

b) Remaining life estimates in the creep range are sensitive to materials data, applied stresses, and corresponding temperature. Therefore, sensitivity studies should be performed to evaluate the interaction of these variables in regards to the remaining life prediction.

Summary Observations

The Fitness-for-Service Code provides some guidance regarding the use of residual welding stresses in the calculation of creep life. However, the texts in Parts 9 and 14 use terms such as ‘may’ and ‘should’, and as such, are not mandatory requirements. However, in Part 10, the term ‘shall’ is used. Nonetheless, adding 14 °C to the design temperature to account for creep damage may, or may not, result in an accurate lifetime assessment.

Operating stresses in the floor, due to temperature gradients and due to friction, are likely in the range of 100 to 125 MPa. For a 30-year tank life, the allowable creep stress is on the order of 130 MPa. This implies that residual stresses associated with welding must be quite low.

The Code is not specific regarding a mandatory requirement to include residual stresses in the assessment of creep. Nonetheless, it is clear that residual stresses must be included if the creep evaluation is to be accurate. As a minimum, the allowable residual stress will point to the degree of stress relief required to reach a 30-year tank life. To this end, a requirement to include residual stresses, even if not mandated by the Code, should be included in the tank procurement specification.

5.15.10 Combined Fatigue and Creep Damage

A common approach to estimating the cumulative effect of successive fatigue cycles is Miner’s Rule ²⁶. The approach is based on the following assumptions:

- The damage accumulated by the material during each cycle is a function only of the stress, σ

²⁶ Lalanne, Christian, “Fatigue Damage, Mechanical Vibration and Shock Analysis, Third edition - Volume 4”, Library of Congress Control Number 2014933740, ISTE Ltd and John Wiley & Sons, 2014

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- For n cycles, the damage (or fraction of fatigue life) at the specified stress is

$$d = \frac{n}{N}$$

where N is the number of cycles to the failure at the stress level σ .

- The damages, d_i , are added linearly, as follows:

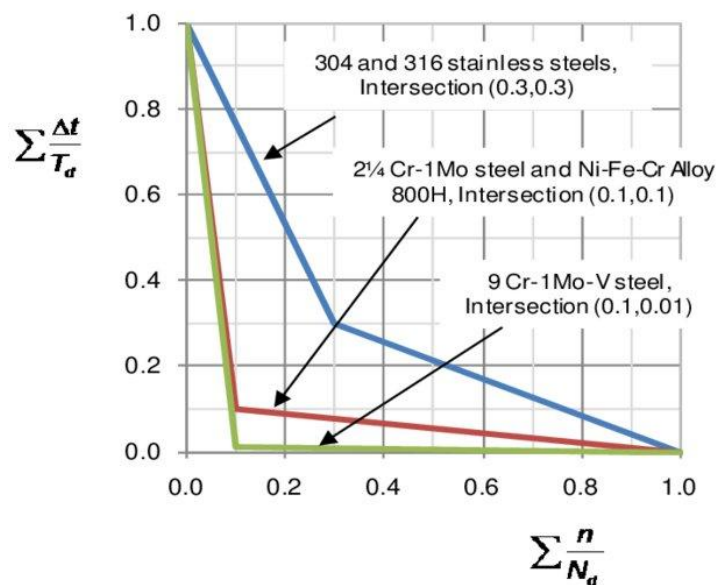
$$D = \sum_{i=1}^k d_i = \sum_i \frac{n_i}{N_i}$$

- It is assumed that damage accumulates without the influence of one level on another. In this case, failure occurs when:

$$D = \sum_i \frac{n_i}{N_i} = 1$$

A similar approach can be used to estimate the cumulative effect of creep damage, with cycles (n) replaced by time (t).

As might be expected, the accumulation of fatigue damage reduces the ability of the material to accommodate creep damage. Allowable combinations of fatigue damage in conjunction with creep damage, for stainless steel and two ferritic steels, are shown in Figure 5-37.



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Figure 5-37 Allowable Combinations of Fatigue and Creep Damage

5.15.11 Preliminary Lifetime Assessment

The sections above discuss methods for assessing the fatigue, the corrosion, and the creep lifetimes of the tank. However, quantitative assessments are not presented, as the calculation of stresses (steady state, transient, and cyclic) are beyond the scope of the Design Basis study.

As a surrogate, representative values for the stresses have been developed from a Federal laboratory study evaluating the low cycle fatigue life of hot salt tanks ²⁷. Inputs to the computational fluid dynamics model are based on operating data from Crescent Dunes on the following parameters, each as a function of time:

- Downcomer flow rate and temperature
- Hot salt flow rate to the steam generator
- Inventory temperature and level.

Calculations of the floor temperatures and the floor stresses were developed for 3 representative days: clear; partly-cloudy; and intermittent clouds. The cycle of days are assumed to repeat, until a period of 30 years is reached.

Estimates of the cumulative fatigue damage, and the cumulative creep damage, were developed using the methods presented in ASME Section III, Division 1, Subsection NH and in API 579-1 / ASME FFS-1.

For floor radii between 0 and 75 percent of the tank radius, the creep and fatigue damage results are shown in Figure 5-38. For floor radii between 93 and 100 percent of the tank radius, the results are shown in Figure 5-39. In both figures, the combined fatigue and creep damage should remain beneath the limits defined by the red and the blue lines.

For floor locations between 0 and 75 percent of the tank radius, the fatigue damage is minor; perhaps 1 percent of the allowable damage. Creep damage is in the range of 90 to 105 percent of the allowable damage, and lifetimes of the floor at these locations are projected to last the duration of the project.

In contrast, for floor locations between 93 and 100 percent of the tank radius, the cumulative fatigue damage is in the range of 2 to 15 percent of the allowable value, while the cumulative creep damage is a

²⁷ Osorio, J. (National Renewable Energy Laboratory, Golden, Colorado), Failure Analysis for Molten Salt Thermal Energy Tanks for In-Service CSP Plants, DOE Award Number DE-EE00038475

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factor of 10 to 20 times the allowable value. A review of the calculated floor stresses from the CFD / FEA model show values that are routinely above the yield value. In general, stresses in this range are inconsistent with a multi-decade creep lifetime.

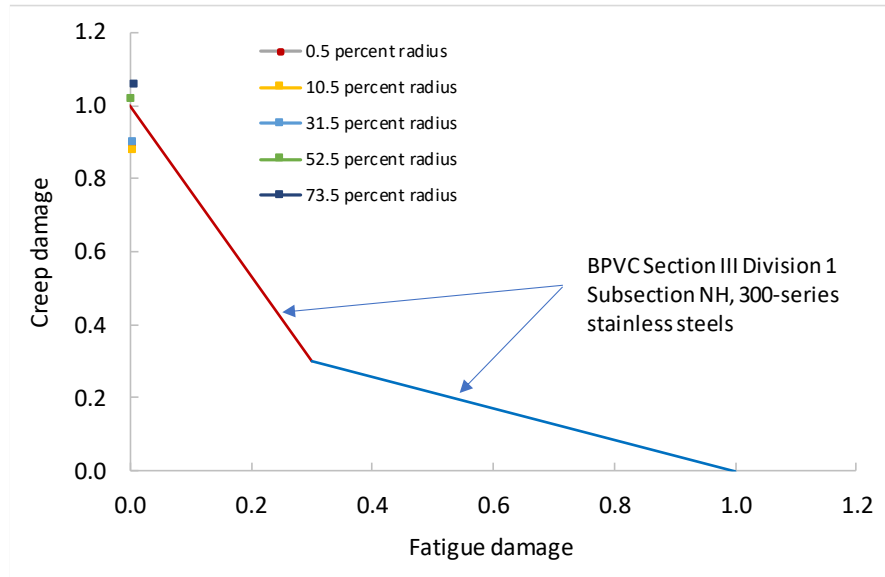


Figure 5-38 Combined Creep and Fatigue Damage for Tank Radii Less than 75 Percent

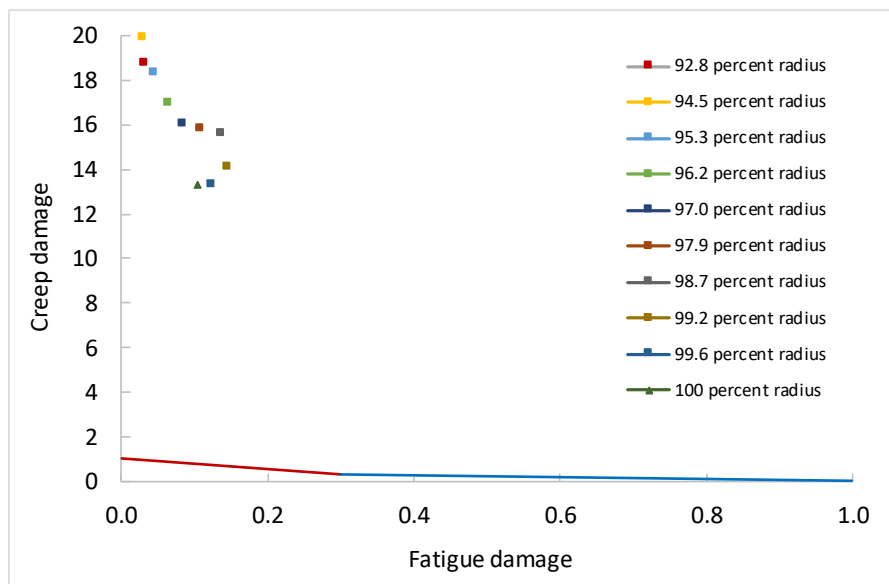


Figure 5-39 Combined Creep and Fatigue Damage for Tank Radii Greater than 93 Percent

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The floor stresses are likely the result of a combination of 1) friction loads, and 2) self-restraint due to thermal stresses generated by radial temperature gradients. The relative contributions of the two effects are not yet known. Methods to reduce the stress values may involve some combination of the following:

- Limits on the maximum coefficient of friction between the floor and the foundation; i.e., certain materials in contact with the floor may be excluded from design consideration
- Constraints on tank operating procedures; i.e., if the tank level is W m, if the inventory temperature is X °C, and if the downcomer flow rate is Y kg/sec, then the receiver outlet temperature cannot exceed Z °C.

5.16 Material Selection

5.16.1 Stabilized and Non-stabilized Stainless Steel

Non-stabilized Stainless Steel

The hot salt tank at the 10 MWe Solar Two demonstration project was fabricated from Type 304H stainless steel. The tank operated at temperatures above 538 °C (1,000 °F). As such, the Code required the use of an 'H' grade of stainless steel, with a carbon content of at least 0.04 percent, to provide the required resistance to creep. However, the H grades of stainless steel are susceptible to a failure mechanism known as intergranular stress corrosion cracking if the following four conditions are met simultaneously:

1. Sensitization. The stainless steel becomes sensitized when the carbon in the steel reacts with chromium to form chromium carbide. At temperatures above 500 °C, the time required to form the chromium carbide is measured in hours. The chromium carbide preferentially forms at the grain boundaries, as there is space in the matrix to do so. With the chromium depleted at the grain boundaries, the material is susceptible to corrosion at the grain boundaries.
2. Tensile stresses. Tensile stresses can arise due to process conditions, or due to residual welding stresses.
3. Chlorides. The nitrate salt, depending on the purity, contains chlorides in the range of 0.05 to 0.4 percent.
4. Liquid water. The water acts both as a transport mechanism for the chlorides and as an electrolyte.

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If all of these conditions are present, then corrosion-induced cracks can develop at the junctions between grain boundaries, and the cracks can then progress along the grain boundaries due to the tensile stresses. Interestingly, once the cracks start to appear, removing the liquid water does not halt the progression of the cracking.

Project Locations Near the Coast

Coastal locations are subject to salt spray from the ocean. Fabricating a tank from an H-grade of a non-stabilized stainless steel carries the risk of the tank developing intergranular stress corrosion cracking during the fabrication process. To minimize this risk, the Owner may elect to fabricate the tank from a stabilized stainless steel, and then accept the risk of stress corrosion cracking developing at some future date. In all likelihood, the Owner would need to make regular payments to a fund, which would be used to finance the construction of a second tank at some point in the future. The cost of the periodic payments would be reflected in the financial model for the project.

Project Locations in the Desert

For plants in a desert location, the risk of intergranular stress corrosion cracking developing during the fabrication stage of a tank constructed from a non-stabilized stainless steel is small. In addition, once the tank enters normal operation, the potential for intergranular stress corrosion cracking is nil. The first three of the four conditions noted above are present, but the operating temperatures of the tank preclude the presence of liquid water.

Nonetheless, if the tank must be drained for maintenance, and if the tank cools to ambient temperature, then there is the possibility of liquid water forming inside the tank due to moisture condensing from the air. Were intergranular stress corrosion cracking to begin, there would be no mechanism to halt the progression. In the worst case condition, the tank would need to be condemned.

Stabilized Stainless Steel

One method for reducing the potential for intergranular stress corrosion cracking is to use a stabilized stainless steel. A stabilized steel adds niobium (Type 347) or titanium (Type 321) as an alloying element. In Type 347, the niobium bonds with the carbon to form niobium carbide; in Type 321, the titanium bonds with the carbon to form titanium carbide. Both carbides reduce the free carbon in the matrix, which reduces the potential for forming chromium carbide. This, in turn, reduces the potential for intergranular stress corrosion cracking.

The solar industry has essentially adopted Type 347H for the fabrication material for the tanks, piping, pumps, and heat exchangers on the hot side ($> 450\text{ }^{\circ}\text{C}$) of central receiver projects. Type 347H also offers higher strength at elevated temperatures than Types 304H and 316H. The higher strength

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translates to lower fabrication masses, and potentially reduced costs. Nonetheless, Type 347H is not a complete solution to the problem of intergranular stress corrosion cracking, as follows:

- During welding, the metal temperatures are high enough to dissolve the niobium carbide. This releases free carbon back into the matrix. As the weld cools, equilibrium considerations lead to a combination of niobium carbide, chromium carbide, and free carbon, depending on time-at-temperature and cooling rates. A common result is the preferential formation of chromium carbide at the edges of the heat affected zone. This leaves two thin zones, on either side of the weld, which become susceptible to intergranular stress corrosion cracking. The phenomenon is known as knife line attack.
- A post weld heat treatment can be performed, which raises the metal temperature high enough to dissolve both the chromium carbide and the niobium carbide. The metal is then held for a defined period at a defined temperature, which leads to the preferential formation of niobium carbide. The weld is then cooled rapidly through the temperature which preferentially forms chromium carbide. The high cooling rates can be achieved on piping, but cannot be effected on the thick metal sections in tanks, pumps, and heat exchangers.
- Welds with high residual stresses can be susceptible to a failure mechanism known as stress relaxation cracking. High residual stresses can result from thick metal sections with multiple welding passes, or from thin metal sections with long continuous weld lengths. This topic, discussed below in Section 5.16.2, is inherent characteristic of using niobium as an alloying element.

As such, Type 347H can reduce the potential for, but cannot eliminate the possibility of, intergranular stress corrosion cracking. Further, Type 347H introduces potential failure mechanisms not inherent in Type 304H and Type 316H.

Alternative Material Options

An alternative material option recently identified in research studies is found in the European standard for pressurized vessels EN10028-7. This material is stainless steel 1.4910 (X3CrNiMoBN17-13-3), which has a maximum service temperature of 600 °C.

The closest material to 1.4910 in the ASME Boiler and Pressure Vessel Code is Type 316LN (16Cr–12Ni–2Mo–N), which has a maximum service temperature of 427 °C.

Compared to 316LN, 1.4910 includes a 20 percent higher nickel content and the addition of small amounts of boron, and as such, is referred to as Type 316LNB.

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Boron is added to improve the creep properties of the material. Recent studies indicate Type 316LNB has higher allowable 100,000-hour creep rupture stresses than Type 347H at temperatures in the range of 500 to 600 °C²⁸, and it has similar corrosion properties in nitrate salts²⁹. More importantly, there is no niobium in the alloy, which reduces the potential for stress relaxation cracking, as discussed in Section 5.16.2.

For these reasons Type 316LNB is considered a potential material candidate to replace Type 347H; however, alternative failure mechanisms for the new material must be considered. The addition of nitrogen and boron to the alloy must be fully evaluated before considering Type 316LNB as the solution to all metallurgical challenges in the hot tank. Further tests should be conducted on weld samples to determine if the material is susceptible to stress relaxation cracking or other cracking phenomena.

Material Selection for Commercial Projects

The Type 304H stainless steel hot tank at Solar Two operated without failure during its brief 3-year test and evaluation period. However, the tank was never drained, and was never cooled to ambient temperature, for maintenance or repair. Similarly, an internal inspection of the Type 347H stainless steel hot tank at Crescent Dunes showed no signs of cracking, corrosion, or weld failures. However, evidence of stress relaxation cracking has been observed in other commercial projects.

The solar industry, in effect, must choose among potential tank materials, none of which offers a perfect solution:

- Type 304H, and to a lesser extent Type 316H, avoid the potential for stress relaxation cracking due to the absence of niobium. However, the materials are susceptible to intergranular stress corrosion cracking should the tank be exposed to liquid water. As such, if the tank is drained, and then cooled to ambient temperature, then the tank must be exposed to only dry air, without the possibility of moisture condensing from the air. One approach to accomplish this is to introduce only dry air into the interior of the tank. However, moisture could still condense on the outside of the tank. A potential solution is to introduce only dry, and heated, air into the tank such that the metal temperatures never fall below the dew point. To what extent this can be guaranteed by the maintenance company is perhaps an open topic
- Various industries have considered Type 316H as a potential alloy, but the nuclear industry in the past decades have reported hundreds of stress relaxation cracking failures in advanced gas

²⁸ M. Oum et al., "Therma 4910 for molten salt storage tanks: Improved stress relaxation cracking and high temperature properties", SolarPACES 2022, Albuquerque, NM, USA

²⁹ A. Bonk et al., "Thermal Energy Storage using Solar Salt at 620°C: How a reactive gas atmosphere mitigates corrosion of structural materials", SolarPACES 2022, Albuquerque, NM, USA

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cooled reactors³⁰. In general, Type 316H is argued to have high susceptibility to stress relaxation cracking based on various industry failures and lab scale testing

- Type 347H reduces the potential for intergranular stress corrosion cracking, but introduces the potential for stress relaxation cracking. The cracking mechanism is a function of the metal thickness, the number of welding passes, the stress distribution, and the time / temperature history of the tank. If the development rate of the cracking can be predicted with the necessary accuracy, then it might be possible to fabricate a tank in Year 0 of the project, and then fabricate a replacement tank to be placed into service in, for example, Year 10 of the project. However, if the development rate of the cracking cannot be predicted with the required certainty, then the project may be obligated to build two tanks in Year 0 in preparation for an expected failure at an undefined time.

5.16.2 Stress Relaxation Cracking in Type 347H

As noted above, adding niobium to stainless steel results in the beneficial formation of niobium carbide. If the parent metal was never welded, then the material properties would be compelling for use in a solar project. However, welding, and the subsequent chemical reactions, introduce a range of undesirable characteristics.

Source of Cracking

The 3 main contributing factors are 1) susceptible microstructure, 2) residual stresses, and 3) elevated temperature in creep dominated regimes³¹.

Susceptible Microstructure

The metallurgical mechanism contributing to cracking is believed to be the non-homogeneous precipitation of niobium carbonitrides (Nb (C,N)) in the grain interiors and at the grain boundaries. The material, in the as-received condition, has typically received a stabilizing heat treatment to homogenize the precipitates in the solid solution alloy. However, welding the material 1) dissolves many of the precipitates, 2) coarsens the grain structure, and 3) imposes a set of residual stresses. In the grain interiors, precipitation of the niobium carbonitrides occurs during elevated temperature service, or during post weld heat treatment. The precipitation mechanism, which is promoted by residual stresses, results in a fine grain structure with a high strength. In contrast, precipitation of the niobium

³⁰ Pickle, T. (Colorado School of Mines, Golden, Colorado, USA), et al, Evaluation of Alternative Base Materials for Mitigation of Stress Relaxation Cracking in Thermal Energy Storage Tanks, SolarPACES 2024, 30th International Conference on Concentrating Solar Power, Thermal, and Chemical Energy Systems

³¹ E. C. B. Dillingh, A; Aulbers, A.P., "Stress Relaxation Cracking-Augmented Recommended Practice," TNO2016

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carbonitrides at the grain boundaries results in a coarse grain structure due to a low coherency with the γ -austenite grain boundaries. The coarse structure at the grain boundaries has a lower strength than the fine grain structure within the grain boundaries. As a consequence, a zone develops between the grain interiors and the grain boundaries, and this zone is depleted of niobium carbonitrides. The depleted region is termed a precipitate free zone. A scanning electron microscope image of a precipitate free zone, which is on the order of a few microns wide, is shown in Figure 5-40³².

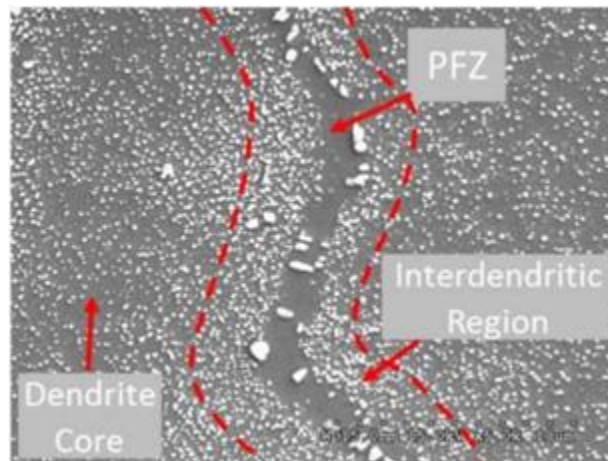


Figure 5-40 Image of a Precipitate Free Zone in the Weld Metal Near a Grain Boundary
It should be noted that the precipitation process occurs on the time scale of years.

Residual Stresses

During welding, the metal in the arc zone undergoes an increase in volume due to 1) the coefficient of thermal expansion, and 2) the change in phase from solid to liquid. After the weld is completed, the zone cools and shrinks. This places the top portion of the weld zone in tension, which is matched by compressive forces in the areas under and adjacent to the weld zone.

The magnitude of the residual welding stresses is a function of the following:

- Section thickness. For section thicknesses greater than about 3 mm, multiple welding passes are generally required. Adding a welding pass to an existing pass generally results in an increase in the residual stresses
- Weld length. Residual stresses are generally additive with weld length.

³² J. A. Siefert, J. P. Shingledecker, J. N. DuPont, and S. A. David, "Weldability and Weld Performance of Candidate Nickel based Superalloys for Advanced Ultra Supercritical Fossil Power Plants Part II: Weldability and Cross-weld Creep Performance," *Science and Technology of Welding and Joining*, vol. 21, pp. 397-428, 2016

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For thick metal sections (i.e., 50 mm), residual stresses can be as high as 500 to 600 MPa. For long weld lengths (> 1 m) in thin plates, residual stresses can reach values up to 600 to 700 MPa. The residual stress values noted here correspond to the material at ambient temperature.

Creep Characteristics

In commercial central receiver projects, the 300-series of austenitic stainless steels are selected for service on the hot side of the plant (565 °C) due, in part, to the creep resistance of the material.

During normal operation, material stresses are limited to ASME allowable values, which are generally in the range of 25 to 30 percent of the yield strength.

Due to the combination of a high resistance to creep and low operating stresses, the material exhibits limited creep produced by dislocations at the grain boundaries. As a result, the potential for the material to relieve residual stresses, welding or otherwise, is limited.

Stress Relaxation Cracking

The residual welding stresses, which, depending on the geometry, can fall in the range between the yield stress and tensile strength, do not relax due to creep. Concurrently, over a period of years, the material develops the precipitate free zones noted above. The zones have a low strength relative to the surrounding grain structure. The combination of high residual stresses and low strength causes the material to preferentially creep at the grain boundaries. Due to the high strains in these limited regions of low strength, the material starts to develop micro voids, which can then develop into cracks.

Since the highest residual stresses are typically below the surface of the weld, the cracks generally begin below the surface of the material and then progress out towards this surface. This leads to two undesirable characteristics:

- Subsurface cracks are difficult to locate
- Once the crack reaches the surface, and the crack is easy to locate; however, the crack has already likely progressed through the full thickness of the material.

Mitigation Measures

A range of mitigation measures are available to reduce the potential for stress relaxation cracking, including 1) conducting a post weld heat treatment, 2) selecting a weld filler material other than E347 (Type 347H), and 3) selecting a non-stabilized stainless steel for the tank.

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Post Weld Heat Treatment

Post weld heat treatment offers two benefits for components prior to elevated temperature service: 1) reduction in residual stress, and 2) stabilization of the microstructure^{33,34}. Some literature suggests a single heat treatment of 875 °C for a few hours is sufficient for reducing the susceptibility to stress relaxation cracking. However, a heat treatment near the solution temperature (> 1,050 °C) is needed for a complete stress relief and is recommended to retain the ductility in a highly restrained weld.

A multi-step process for the post weld heat treatment could be the best option for preventing stress relaxation cracks in thick weldments, but the procedure must be carefully engineered to prevent relaxation cracking during heat treatment. Parameters such as the heating rates, hold times, temperatures, and quenching rates are all important parameters that influence the final stress state and the final microstructure. A three-step process consists of 1) an initial stress relief, 2) a solution heat treatment to dissolve the microstructure and effect a full stress relief, and 3) a stabilization heat treatment to form the desired precipitates.

1. The initial stress relief is recommended to reduce some of the initial stresses prior to ramping up to the solution temperature
2. The solution temperature helps to dissolve intergranular carbonitrides and reduces δ -ferrite by promoting the formation of γ -austenite in a predominantly two phase field (γ -Nb(C,N)). The reduction of the intergranular carbonitrides helps to minimize the precipitate free zones and allows for coarsening of intragranular Nb(C,N).
3. The final step of stabilization, initially developed by Morishige³⁵, is a step to further stabilize the microstructure by reducing the remaining free carbon through the nucleation and growth of Nb (C,N) using any of the remaining Nb. This helps to prevent the formation of chromium carbides ($M_{23}C_6$) during cooling.

Figure 5-41 shows an overlay of the three-step post weld heat treatment times and temperatures relative to precipitation C-curves for the formation of coarse niobium carbide, fine niobium carbide, and chromium carbide³⁶. The three-step post weld heat treatment method has been used successfully in industry for decades.

³³ J. R.D. Thomas, "HAZ Cracking in Thick Section of Austenitic Stainless Steels-Part 2," *Welding Journal*, 1984

³⁴ R. D. Thomas, Jr. and R. W. Messler, "Welding Type 347 Stainless Steel- An Interpretive Report," WRC Bulletin 1997

³⁵ N. Morishige, M. Kuribashi, H. Okabayashi, and T. Naiki, "On the Prevention of Service Failure in Type 347 Stainless Steel," presented at the Third International Symposium of the Japan Welding Society, 1978

³⁶ V. O. B. Messer, T Phillips, "Optimized heat treatment of 347 for high temperature application," *Corrosion*, 2004.

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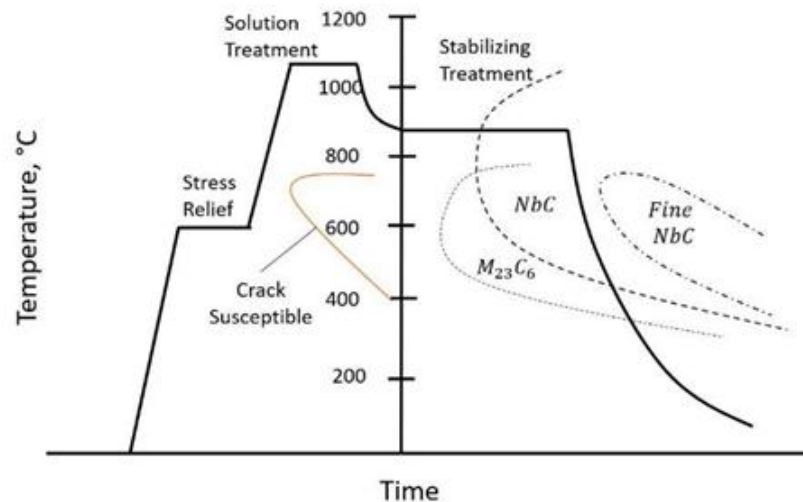


Figure 5-41 Three-Step Post Weld Heat Treatment Time and Temperature Schedule

Weld Filler Material

The principal candidate weld filler materials for a tank fabricated from Type 347H are the following:

- ER347H³⁷, which is essentially the tank material. It is an austenitic material, with ferrite numbers in the range of 2 to 9, with a typical value of 4.
- E16.8.2³⁸, which is an austenitic material with a ferrite number of 1 to 6.

For both of these consumables, bismuth-bearing constituents are held to very low levels (< 0.002 percent Bi) as required by API 582, Welding Guidelines for the Chemical, Oil, and Gas Industries.

E16.8.2 was developed as a more ductile alternative to 347H consumables to avoid failures in the heat affected zones of thick (> 12 mm) 347H base materials. The main benefit of E16.8.2 is reduced δ -ferrite, which is prone to transforming to a brittle σ phase.

³⁷ Metronode Welding Consumables for High Temperature Alloys, Data Sheet C-11, 347H Stainless Steels, Metronode Products LTD, Surrey, KT16 9LL, March 2016

³⁸ Metronode Welding Consumables for High Temperature Alloys, Data Sheet C-12, 16.8.2 for High Temperature 3XXH Stainless Steels, Metronode Products LTD, Surrey, KT16 9LL, May 2019

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The σ phase is a tetragonal crystal structure, with a precipitation temperature between 600 °C and 1,000 °C³⁹. The structure is composed of various intermetallic compounds, including FeCr, FeMo, and (FeNi)_x(CrMo)_y. The σ phase can increase the hardness, and decrease both the toughness and the elongation, of steels.

The σ phase issues from the phase transformation of $\delta \rightarrow \sigma$ (δ -ferrite to σ phase). When the Cr content is below 20 weight percent, the precipitation of the σ phase is not readily observable in austenitic stainless steels; however, the σ phase can be formed quickly when the Cr content is between 25 and 30 weight percent.

The addition of strong ferrite stabilizers other than Cr, such as Si and Mo, can also lead to the formation of the σ phase.

The δ -ferrite is a body centered cubic structure, in which Cr, Si, and Mo can efficiently diffuse. Further, the δ -ferrite regions, compared to the γ -austenitic regions, are rich in Cr. As such, the $\delta \rightarrow \sigma$ phase transformation preferentially occurs in the δ -ferrite regions. Precipitation can also occur directly in the δ -ferrite particles.

As such, reducing the δ -ferrite in the weld region offers the benefit of increased ductility, which improves the tolerance of strain localizations and mismatches in strengthening mechanisms.

Non-Stabilized Stainless Steels

The potential for stress corrosion cracking in a 300-series stainless steel is directly associated with the addition of niobium to the alloy. If niobium is not present, then the reprecipitation of niobium carbonitrides described above does not occur. This, in turn, results in a matrix which has 1) a more isotropic distribution of strength and ductility, and 2) a lower propensity for the development of internal cracks.

Nonetheless, the alternate candidate stainless steels, including Type 304H, Type 316H, and Type 304LNB, each of which have their own set of advantages and disadvantages. The topic is discussed in some detail in Section 5.16.1.

5.16.3 Stress Relaxation Cracking in Type 316LNB with Lean Austenitic Weld Filler

Type 316LNB is a low carbon version of Type 316, with additions of nitrogen and boron to provide resistance to creep similar to the properties of Type 316H. A recent study compared the resistance to

³⁹ Hsieh, Chih-Chun, and Wu, Weite, (Department of Materials Science and Engineering, National Chung Hsing University, Taiwan), 'Overview of Intermetallic Sigma (σ) Phase Precipitation in Stainless Steels', International Scholarly Research Network, Volume 2012, Article ID 732471, January 2012

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stress relaxation cracking of Types 316LNB with Type 347H⁴⁰. The weld filler material for Type 316LNB was a lean austenitic (ER16.8.2, 16 Cr - 8 Ni - 2 Mo), which has demonstrated improved toughness and thermomechanical properties compared to matching filler materials for Type 347H (347H-16).

Testing at 600, 700, and 800 °C show no cracking for the 22-hour test for Type 316 LNB with the ER16.8.2 filler material, while stress relaxation cracking is observed for alloy 347H with the matching filler material.

Future efforts will expand the test matrix, including longer times at 600 °C to simulate the conditions in a commercial tank. The planned work will also investigate the microstructural conditions needed for fracture, such as secondary phases in the partially melted weld zone.

In general, initial test results show that Type 316LNB is less prone to the development of stress relaxation cracking than Type 347H. However, additional testing is required to help predict the long term (30+ year) strength and ductility characteristics of the welded material in commercial service.

⁴⁰ Pickle, T. (Colorado School of Mines, Golden, Colorado, USA), et al, Evaluation of Alternative Base Materials for Mitigation of Stress Relaxation Cracking in Thermal Energy Storage Tanks, SolarPACES 2024, 30th International Conference on Concentrating Solar Power, Thermal, and Chemical Energy Systems

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6. Nitrate Salt Steam Generators

6.1 *Commercial Configuration*

The Rankine cycles in essentially all solar projects use a single reheat design. Correspondingly, the steam generators in essentially all solar projects use separate heat exchangers for superheating, reheating, evaporation, and preheating. Separate heat exchangers are required due to 1) differences in the characteristics of latent heat transfer (evaporator) and sensible heat transfer (superheater, reheater, and preheater), and 2) different operating pressures (superheater and reheater).

A common approach to the fabrication of the heat exchangers is a U-tube bundle in a straight shell, with 2 passes on the shell side. The fluid on the shell side is typically salt. This is done for two reasons:

1. The high pressure fluid is the water/steam, and accommodating the high pressure fluid in tubes is easier to accomplish than accommodating the high pressure fluid in the shell
2. The salt in the heat exchanger can reasonably be expected to freeze at least once in the life of the project. If the salt is on the tube side, and freezes in the tubes, then the solid salt can form essentially immovable plugs. Thawing the tubes anywhere along the length of the frozen plug runs the risk of yielding the tube due to the nominal 4 percent increase in salt volume on thawing. If the freeze/thaw cycle is repeated, then there is the possibility of rupturing the tube. (This was the failure mechanism for the tube bundle in the kettle evaporator at the Solar Two demonstration project.) In contrast, if the salt is on the shell side, the thawing process begins by ensuring that the salt in the lines to and from the heat exchanger is liquid. Activating the electric heat tracing on the shell and the channel first melts the salt on the inner surface of the shell and the channel. Since the heat input from the heat tracing is typically a modest value, the initial mass of salt which melts is also a modest number. The volume change on thawing still occurs, but the liquid salt has an exit path through the salt lines. As such, the potential for plastically yielding the shell is low.

Most commercial heat exchangers use a flat tubesheet. During fabrication, the tubes are inserted into the tubesheet, the ends of the tubes are strength welded to the face of the tubesheet, and the tubes are then rolled into the tubesheet. The rolling operation occurs after the welding operation. This prevents any air which might be trapped in the gap between the tube and the tubesheet from heating and expanding into the weld region, which could produce a porous weld.

In some commercial projects, the evaporators use forced circulation, while other projects use natural circulation. The selection is largely the responsibility of the equipment supplier.

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6.2 *Heat Exchanger Location*

Similar to the discussion in Section 3.2, Equipment Location for Oil-to-Salt Heat Exchangers in Parabolic Trough Projects, there is a range of options for the location of the steam generator heat exchangers.

In some commercial projects, the heat exchangers are located on an elevated platform between the storage tanks. The height of the platform is slightly above the mid-point height of the tanks. As such, the heat exchangers can drain to the tank with the lower liquid level.

In other projects, the steam generator is located at grade, and the heat exchangers drain to a tank located below grade. Drain pumps, located on the drain tank, pump salt from the drain tank back to the main storage tanks.

In general, locating the heat exchangers at grade is less expensive than locating the equipment on an elevated platform. However, locating a drain tank below grade is not an inexpensive option. Further, the drain tank must be heat traced on a continuous basis, and this represents a continuous auxiliary energy demand.

6.3 *Salt-Side Pressure Relief Valves*

In the steam generators at most commercial projects, the salt is on the shell side of the heat exchangers and the water/steam is on the tube side. This is done for two reasons:

- Fluid pressures on the water/steam side are approximately 20 times the pressures on the salt side. The relatively small diameters of the tubes are structurally more efficient than the relatively large diameters of the shells at accommodating the high pressures of the water/steam
- At some point, the salt in the heat exchangers is likely to freeze. If the salt is on the shell-side, then the thawing process can consist of the following steps: activate the heat tracing on the lines to and from the heat exchanger; ensure that the salt in the lines is liquid; and then activate the heat tracing on the shell. The salt will begin to melt in a thin film on the inside of the shell, and the change in the volume during melting is offset by a small flow of salt from the heat exchanger. In this manner, plastic deformations of the shell can be avoided.

6.3.1 *Steam Generator Tube Ruptures*

At some point in the life of the project, a rupture in a steam generator tube is expected. This will introduce steam into the shell side of the superheater or the reheater, and introduce a mixture of water and steam into the shell side of the evaporator or the preheater. Since the design pressure on the

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water/steam side is higher than the design pressure on the salt side, a tube rupture will produce a transient rise in pressure on the salt side.

The rupture will also force a mixture of salt, steam, and possibly water through the preheater and into the distribution header at the bottom of the cold salt tank. The two- or three-part flow will leave the header, traverse the inventory in the cold tank, evaporate the remaining water, and introduce a mass of steam into the tank ullage space.

The rupture may also force a mixture of steam and salt backwards through the following hot salt equipment (depending on the use and the location of check valves):

- The hot salt pump. The steam and salt mixture would traverse the pump column downward, traverse the salt inventory upward, and introduce a mass of steam into the tank ullage space
- The minimum recirculation valve for the pump. The steam and salt mixture would enter the recirculation distribution header at the bottom of the hot tank, traverse the inventory upward, and introduce a mass of steam into the tank ullage space.

In addition, the rupture may also force a mixture of steam and salt backwards through the following cold salt equipment (depending on the use and the location of check valves):

- The attenuation pump. The steam and salt mixture would traverse the pump column downward, traverse the salt inventory upward, and introduce a mass of steam into the tank ullage space
- The minimum recirculation valve for the pump. The steam and salt mixture would enter the recirculation distribution header at the bottom of the hot tank, traverse the inventory upward, and introduce a mass of steam into the tank ullage space.

6.3.2 Relief Valve Options

Two options are available for the location of the relief valves:

Roofs of the salt tanks

Each of the salt storage tanks have a) atmospheric vents, and b) pressure/vacuum relief valves. The capacities of the vents and the relief valves would be large enough to accommodate the steam/air flow rates associated with a tube rupture in the steam generator and maintain the ullage pressure within the design limits for the tanks (i.e., + 50 / - 2.5 mbar).

The pressure/vacuum relief valves have only one or two moving parts, and their reliability is excellent. However, the valves require both heat tracing and insulation, and the reliability of the heat tracing and the insulation is not guaranteed. This is particularly so on top of the tanks, where access for

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maintenance is inconvenient, and therefore infrequent. At the Solar Two demonstration project, the heat trace on the pressure / vacuum relief valve for the hot tank did not fully reach the set point temperature, and a small amount of salt vapor froze in the valve. When the tank filled during one morning startup, the ullage pressure increased to the point where the tank bottom bowed down, and the tank lifted from the foundation. Fortunately, the relief valve opened before the tank ruptured. Apparently, this occurrence is not unique among commercial solar projects. At one plant, the wall-to-roof connection split due to a sticking pressure / vacuum relief valve, and at other plants, more than one instance of the tank lifting from the foundation has occurred.

In the interests of protecting the reliability and the availability of the tanks, the demands on the pressure / vacuum relief valves should be held to the minimum. This can be accomplished by providing separate pressure relief valves in the inter-heat exchanger piping to reduce the transient pressures imposed on the storage tanks.

Roofs of the salt tanks and inter-heat exchanger salt piping

In commercial projects, it is common to provide pressure relief valves on both the storage tanks and in the inter-heat exchanger piping within the steam generator. The relief valves in the steam generator discharge to a safe location, and this location is separate from the ullage space in the main storage tanks.

On a related point, the steam generator system will have some number of control and isolation valves. The valve arrangement will vary from project to project. If it is possible to isolate a heat exchanger by closing the valves, either by placing the valves in manual or during maintenance, then the ASME Code will not permit a relief design that relies on the valves located on the storage tanks. Instead, local relief valves will be required on the inter-heat exchanger piping, with the number of relief valves dependent on the geometry of the piping and the location of the salt isolation and control valves.

6.4 Cold Reheat Steam Electric Heater

Prior to turbine synchronization, live steam is sent to the condenser through the turbine bypass system. The system consists of a high pressure throttle valve, a steam attemperator, a low pressure throttle valve, and a second steam attemperator.

Early in the startup process, the temperature of the salt entering, and leaving, the superheater is approximately 290 °C. This results in a saturation pressure in both the evaporator and the superheater of about 74 bar.

With the bypass system in service, the steam from the superheater must be reduced in pressure to a value which can be accepted by the reheater. As such, the high pressure throttle valve reduces the pressure to the nominal design pressure of the reheater, which is about 32 bar. The corresponding cold reheat steam temperatures are in the range of 235 to 240 °C.

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To provide an adequate margin against salt freezing at the cold end of the reheater, an electric steam heater is located downstream of the high pressure throttle valve and upstream of the cold end of the reheater. The capacity of the electric heater depends on the details of the startup sequence (i.e., steam flow rate and pressure versus time), but is typically on the order of several MWe.

6.5 *Cyclic Operation*

In a solar project, the steam generator, almost by definition, undergoes a daily startup and shut down cycle.

During overnight hold, cold salt is circulated through the steam generator. The salt is supplied by the cold salt attemperation pump, and the salt leaving the steam generator is returned to the cold salt tank. The heat input from the cold salt compensates for the heat losses from the heat exchangers and associated piping. The heat input also heats the makeup water added to the steam drum to compensate for steam and condensate losses in the Rankine cycle.

To start the steam generator, the hot salt pump is started, and hot salt is blended with cold salt from the attemperation pump at a point upstream of the superheater / reheater. Early in the startup process, the steam flows produced by the superheater and the reheater are not at temperatures high enough to admit to the turbine. The live steam and the hot reheat steam flows are attemperated and throttled, and the flow is then directed to the condenser through a turbine bypass system.

When the live and the reheat steam temperatures reach values which can be accepted by the turbine, a portion of the steam flow in the bypass system is sent to the turbine. The turbine is synchronized, and loaded to a value that will prevent a trip on reverse power. The steam flow in the bypass system is eventually stopped, with all of the steam flow now directed to the turbine.

The relative proportions of hot salt and cold salt at the mixing point upstream of the superheater / reheater are adjusted until the flow of cold salt reaches zero. The output of the turbine is adjusted by changing the speed of the hot salt pump.

To shut down the steam generator, the above process is essentially reversed.

The blending process of adding hot salt to cold salt is a consequence of the vendor limits on allowable rates of temperature change and thermal shock. Typical limits include a maximum rate of temperature change of 10 °C/min, and a maximum thermal shock (Fluid temperature - Metal temperature) of 60 °C. The limits are developed by the vendor to ensure that the heat exchangers have a low cycle fatigue life corresponding to at least 30 years of daily startup and shutdown cycles.

The vendor also specifies a minimum allowable flow rate on both the shell- and tube-sides. A typical value is 16 percent, which corresponds to a turndown ratio of 6:1. At low flow rates, the pressure drops

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on both the tube- and the shell-sides approach very low values. Local effects due to friction and buoyancy can then influence the flow distribution, which can produce stratified flow, channel flow, or both. If the flow distribution is not uniform, then the temperature distribution within the heat exchanger is also not uniform. Non-uniform temperature distributions can produce internal stresses which are non-uniform, unpredictable, and oscillating. These stresses may add to, or subtract from, the normal process stresses, which can lead to meaningful reductions in the low cycle fatigue life of the equipment.

6.6 *Project Experience*

6.6.1 **Overnight Hold**

The steady state heat losses through the insulation on the heat exchangers, and through the insulation on the steam generator piping, is a fraction of 1 percent of the design duty of the steam generator. However, the water and the steam leakage rates in the Rankine cycle can reach problematic levels if 1) the maintenance staff is not conscientiously repairing the steam and condensate leaks as they develop, and 2) tubes leaks in any of the heat exchangers result in steam being carried into the cold salt tank.

A condition can arise in which the energy in the flow of cold salt from the attemperation pump is not sufficient to compensate for the thermal demand of the steam generator; i.e., the temperature of the salt returning to the cold tank falls below the minimum operating temperature of 275 °C. The first line of defense is to add thermal energy to the drum by operating the electric water heaters that preheat the heat exchangers prior to filling with salt. The second line of defense is to add auxiliary steam to the drum from the auxiliary steam generator. The third line of defense is to blend a very small quantity of hot salt with the cold salt upstream of the superheater / reheater mixing station.

The first two approaches are straightforward to implement and to control, but both increase the auxiliary energy demand in the plant. The third uses energy collected by the receiver, but any energy withdrawn from the hot tank for temperature maintenance reduces the daily electric energy production. Further, the quantity of hot salt required is a small fraction of the flow capacity of the hot salt pump. This can lead to oscillations in the flow of salt from the hot pump, which, in turn, can lead to oscillations in the mixed salt temperature upstream of the superheater / reheater. Oscillations in the mixed salt temperature have been shown to lead to low cycle fatigue damage of the heat exchangers, particularly if the rates of temperature change exceed the vendor limit of 10 °C/min.

The solution to this problem is to generously size the electric water heaters in the steam generator, studiously repair any steam and condensate leaks in the Rankine cycle, and provide a means for artificially increasing the pressure drop on the salt side of the steam generator if the hot salt pump must be operated at a low flow rate.

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6.6.2 Low Cycle Fatigue

A commercial project experienced multiple failures of the tube-to-tubesheet connections in the superheater in the first years of commercial service. A possible explanation for the common failure patterns is as follows:

1. During startup and shutdown, the rates of change in the temperature of the mixed salt entering the heat exchanger routinely exceeded the vendor limits. The rates of change were both positive and negative, producing, in effect, oscillations in the inlet temperature. The oscillations generated differential thermal expansions across the diameter of, and through the thickness of, the tubesheet.
2. The differential thermal expansion was exacerbated by the low steam flow rates, and the very low pressure drops, during startup and shutdown. Specifically, the steam flow tended to channel, favoring some tubes over others. This resulted in lower temperatures in some regions of the tubesheet, and higher temperatures in other regions.
3. The non-uniform thermal expansion caused flexing in the tubesheet, which over several thousand cycles, caused a relaxation in the friction forces in the rolled tube-to-tubesheet connections.
4. If the rolled connection is lost, then the tube-to-tubesheet welds must carry the loads associated with differential expansion between adjacent tubes, and the differential expansion between the tubes and the tubesheet. If the stresses are high enough, the welds can crack due to low cycle fatigue.

There is additional evidence that rapid changes in the temperature of high pressure heat exchangers can lead to failures of the tube-to-tubesheet connections. Specifically, the Solar One central receiver project used a superheated steam receiver, operating at nominal conditions of 100 bar and 540 °C. The thermal storage system consisted of a single tank thermocline, filled with a mixture of rock, sand, and mineral oil (Caloria HT-43). Energy was transferred from the receiver steam into the rock/sand mixture by circulating Caloria through an economizer, a steam condenser, and a steam desuperheater. Energy was transferred from the rock/sand mixture by circulating Caloria through an economizer, an evaporator, and a superheater.

To isolate the demineralized feedwater in the Rankine cycle from the Caloria in the storage system, the storage system charging and discharging heat exchangers used a double tubesheet design. Tube-to-tubesheet connections were by means of rolled joints; seal welds were not used. Leaks would be confirmed by the presence of either steam or Caloria in the intra-tubesheet space.

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Within 400 hours, the tube-to-tubesheet joints started to leak ⁴¹. Repairs were attempted by applying seal welds to the ends of the tubes and re-rolling the tubes into the tubesheets. However, the repairs were unsuccessful, and the storage system was essentially abandoned in place.

A post-failure finite element model of the heat exchanger, using a rate of temperature change of 9.6 °C/min, showed that the tube stresses at the rolled connections approached the yield value. If a material is operating at or near the yield stress under daily cycling conditions, the low cycle fatigue life is expected to be very short.

6.6.3 Water Chemistry

High levels of O₂ can lead to accelerated corrosion rate, especially if the purity of the condensate and the feedwater purity is degraded by contamination (salts) passing through the demineralization system or from condenser cooling water in leakage. A deaerator can be used to control O₂ levels to the required values. Generally, for all-ferrous systems (those without copper tubing in heat exchangers) the desired levels are in the range of 5 ppb up to a maximum of 100 ppb.

Additionally, the pH of the feedwater is typically adjusted to a value of 9.4 or higher to reduce the solubility of protective oxides formed, and with adequate O₂ present, to support the formation of a stable layer of hematite on the water-side surfaces of the system. This is usually done by adding alkaline agents to the feedwater such as ammonium hydroxide, or ammonium hydroxide in combination with a neutralizing amine, such as ethanolamine.

At the Solar Two project, an examination of the water side of the preheater, after about 1 year of service, showed significant quantities of a loosely adherent deposit. An analysis showed the deposits to be magnetite. This is an undesirable condition, for the following reasons:

- Magnetite is a metal oxide, and due to its low thermal conductivity, deposits can reduce the heat transfer rates in the preheater and in the evaporator. Further, if the deposits form preferentially in some tubes and not others, this can lead to non-uniform flow distributions, non-uniform temperature distribution, and non-uniform stress distributions in the heat exchanger.
- Magnetite deposits are an indication of corrosion upstream of the evaporator. Depending on the water treatment approach selected for the project, the corrosion process generally accelerates if the pH is too low (< 9) and / or the dissolved oxygen levels are too low (< 5 ppb). This combination of pH and dissolved oxygen can occur in the feedwater if 1) the dosing of the alkaline agents is insufficient, and / or 2) the deaerator venting is too effective. Under two-phase

⁴¹ S. E. Faas, et. al., (Sandia National Laboratories, Albuquerque, New Mexico), 10 MWe Solar Thermal Central Receiver Pilot Plant: Thermal Storage Subsystem Evaluation - Final Report, Sandia Report SAND86-8212, June 1986

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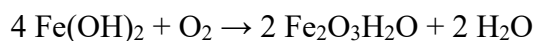
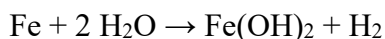
conditions in the evaporator, low oxygen will always occur, and low pH may occur depending on local two-phase conditions and the alkaline agents applied.

- In the evaporator, oxide deposits can come out of suspension at the locations in the tubes where evaporation begins. If the evaporator uses austenitic tubes, then a passivation layer is expected to form on the tubes. However, defects or local variations in the passivation layer can lead to sensitization and pitting corrosion. Specifically, the chemical conditions under the deposits (sodium, chloride, sulfate, ammonia, and the resulting pH) can be different than the surrounding area. If so, a situation can develop in which the protective oxide passivation dissolves and active corrosion can initiate. In the case of pitting corrosion, this can become autocatalytic. Specifically, the dissolution of iron within the pit generates positive iron ions, which draw in and concentrate mobile, acidic ions, such as chlorides, within the pit.
- If the evaporator is a forced recirculation design, then oxide particles in the feedwater can damage the bearing, erode the impellers, and damage the shaft seals in the steam drum recirculation pumps.

Possible Sources of the Deposits

Hematite

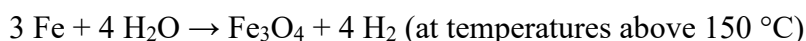
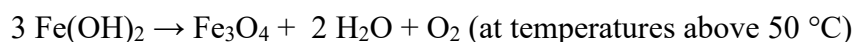
Hematite (Fe₂O₃), in a hydrated form, is formed on carbon steel in an oxidizing environment, as follows:



In general, hematite is a desirable form of iron oxide. It forms an adherent corrosion layer and isolates the parent metal from continued corrosion. It also exhibits a very low solubility in water compared to magnetite.

Magnetite

Magnetite (Fe₃O₄) is generally formed in a relatively reducing environment, through two different reactions:



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In general, magnetite is a less desirable form of iron oxide in flowing feedwater conditions ($>100\text{ }^{\circ}\text{C}$), as 1) it forms a more soluble passivation layer than hematite in water, and 2) it is subject to damage mechanisms such as flow accelerated corrosion, particularly in turbulent locations operating at temperatures between 100 and $250\text{ }^{\circ}\text{C}$. Removal of the magnetite layer via flow accelerated corrosion exposes new material to the corrosion process. It also results in corrosion products moving through the system, where deposition may occur and lead to further damage mechanisms (e.g. accumulation of deposits in evaporator).

Water Chemistry Control

Although perhaps stating the obvious, the water treatment staff needs to maintain strict control over the water chemistry. The problem is more acute in plants which cycle each day, particularly if the vacuum is broken in the condenser each evening and the condensate in the hot well is exposed to oxygen. Further, the inline instruments for measuring pH and dissolved oxygen need to be maintained in good working order and to receive periodic calibrations.

In general, control of pH in high purity systems is best accomplished through conductivity control, as opposed to strictly relying on direct pH measurement. In addition, older dissolved oxygen measurement technology relying on electrochemical potential measurements are less robust than optical methods employing luminescence technology. Application of ‘smart’ chemistry monitoring approaches, which incorporate instrumentation validation techniques, can further reduce the expertise required by the operating staff to maintain proper chemistry control.

If the experience from the Solar Two project, and from a range of commercial projects, are any indication, the water treatment system must be fully staffed, and water treatment specialists must be brought in on a periodic basis to review the effectiveness of the treatment system. Failure to satisfy each of these requirements will result in a significant negative effect on the plant availability.

6.6.4 Startup and Shutdown

A common steam generator arrangement in commercial projects consists of the following:

- Two 50-percent capacity steam generator trains
- Three 50-percent capacity hot salt pumps, discharging to a common header
- A common minimum flow recirculation valve for the three hot salt pumps
- Two 100-percent capacity cold salt attemperation pumps, discharging to a common header

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- A common minimum flow recirculation valve for the two attemperation pumps
- Salt control valves, at the cold ends of the superheater and the reheater, for distributing the relative flows between the two heat exchangers
- A startup feedwater heater, located downstream of the last high pressure feedwater heater and upstream of the steam generator preheater.

The salt pump and discharge valve arrangement is shown in Figure 6-1, and the intra-heat exchanger salt valve arrangement is shown in Figure 6-2.

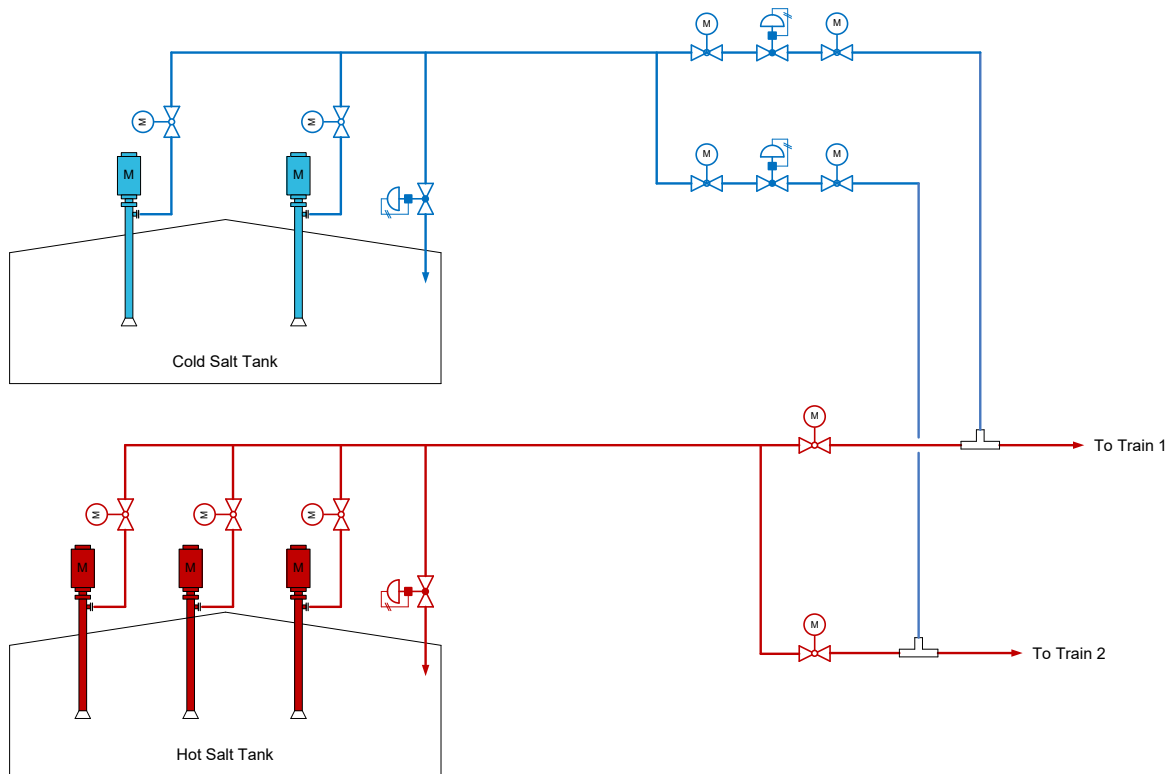


Figure 6-1 Salt Pump Arrangement in a Representative Commercial Project

This arrangement reduces the number of control valves to a minimum, and likely offers the lowest cost piping arrangement. However, the arrangement carries a range of liabilities, as discussed in the following subsections.

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Liability 1 - Sequential Startup

By placing the hot salt pumps on a common header, and placing the attemperation pumps on a common header, there is no method to start a second steam generator train if one train is already in service. To start the second train, the metal temperatures in the first train would need to be reduced to essentially the cold salt temperature to avoid thermally shocking the second train. In effect, starting the second train must be postponed until the first train goes through it's normal daily shutdown.

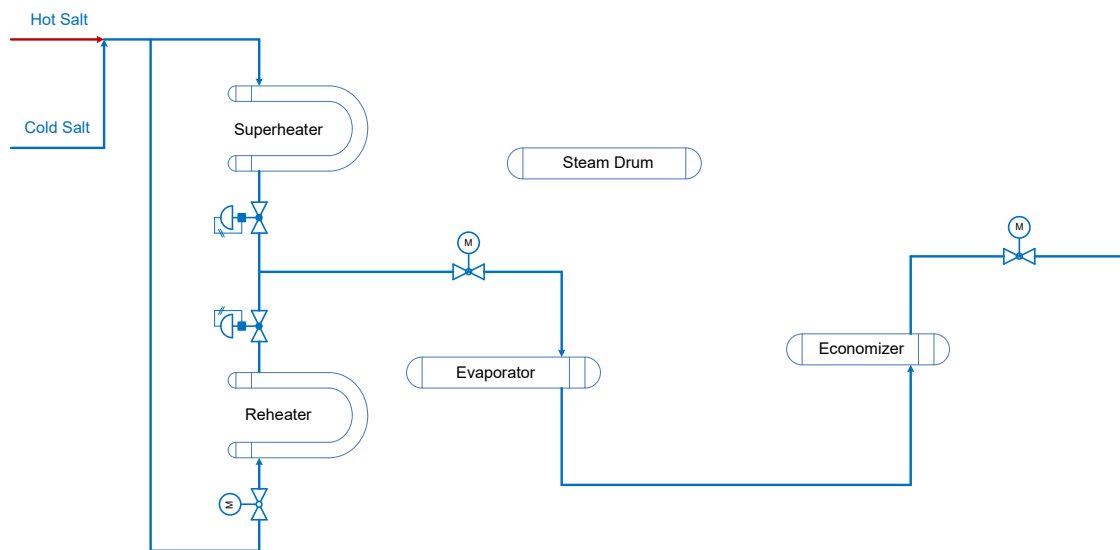


Figure 6-2 Intra-Heat Exchanger Salt Valve Arrangement in a Representative Commercial Project

Liability 2 - Hot Salt Flow Control

The salt control valves downstream of the superheater and the reheater must have Cv values which are high enough to accommodate the design point flow rates with a reasonable pressure drop (< 1 bar). During startup, hot salt is blended with cold salt at the mixing station upstream of the superheater / reheater. The rate at which hot salt is added to cold salt must satisfy the vendor limit regarding the allowable rate of temperature change, which is nominally $10\text{ }^{\circ}\text{C}/\text{min}$. Using representative flow - head curves for the attemperation pump, the acceleration rate at which hot salt can be added to the cold salt is $0.0023\text{ m}^3/\text{hr-hr}$ (6 gal/min-min). A flow rate of $0.0023\text{ m}^3/\text{hr}$ is approximately 0.15 percent of the design flow rate from the hot salt pump. Further, early in the startup process, the salt flow rate to the steam generator is perhaps 15 percent of the design flow rate. As such, the pressure drop through the steam generator is on the order of $0.15^2 = 2$ percent of the design pressure drop. To control the flow from the hot salt pump with this level of accuracy, in combination with a low discharge pressure, a backpressure must be imposed on the steam generator train. The backpressure is typically provided by

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the control valves downstream of the superheater and the reheater. However, these valves have reasonably large discharge coefficients, and to provide a meaningful backpressure on the system, the valves are forced to operate in the range of 2 to 4 percent open. Due to the low system pressures, the hot salt pump is already operating at the minimum speed. As such, all of the flow control adjustments during startup must be made by the control valves. As might be expected, the ability to adjust the flow in increments of 0.15 percent per minute is generally poor. The result is oscillations in the flow from the hot salt pump, which results in oscillations in the mixed salt temperature upstream of the superheater and the reheater. The oscillations can result in rates of temperature change which are a factor of 2 to 4 larger than the allowable rates.

Liability 3 - Salt Pump Recirculation Flow Control

The hot salt pumps and the attemperation pumps share a common pressure point at the mixing station upstream of the superheater / reheater. As such, the following parameters all influence one another:

- The discharge pressure of the hot salt pump
- Operating either one or two hot salt pumps
- The position of the minimum flow recirculation valve for the hot salt pump(s)
- The discharge pressure of the attemperation pump
- The position of the minimum flow recirculation valve for the attemperation pump

During the startup of the steam generator, the following characteristics apply:

- The pressure drop in the system is low
- The discharge pressures of the hot salt pump and the attemperation pump are both low
- The flow rate from the hot salt pump to the mixing station is low, which requires the minimum flow recirculation valve to control the (Mixing station flow) + (Recirculation flow)
- Depending on the speed range of the attemperation pump, the minimum flow recirculation valve for the attemperation pump can also be open to control the (Mixing station flow) + (Recirculation flow)

Under these conditions, the system flow stability can be marginal to poor. For example, to increase the flow of hot salt to the mixing station, the minimum flow recirculation valve for the hot pump can be

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closed slightly. Since the pressure drop in the steam generator flow path is different than the pressure drop in the hot salt pump recirculation line, a change in the recirculation valve position results in a change in the discharge pressure of the hot salt pump. This, in turn changes the effective discharge pressure of the attemperation pump, which changes the flow rate from the pump. In response, the minimum flow recirculation valve for the attemperation pump can change position to provide the required (Flow to the mixing station) + (Recirculation flow). The changes in the discharge conditions for the attemperation pump are then carried over to the hot salt pump, which normally results in a change in the recirculation valve position for the hot salt pump to provide the desired net hot salt flow to the mixing station. The system can then oscillate between two quasi-stable operating points, which results in oscillations in the mixed salt temperate at the mixing station.

Liability 4 - Minimum Steam Side Flow Rates

A typical vendor limit on the minimum allowable flow rate is 16 percent of the design flow rate.

If the design flow rate of the cold salt attemperation pump is 15 percent of the design steam generator flow rate, then operating the attemperation pump at 100 percent speed during overnight hold, and early in the startup process, satisfies the minimum flow requirement on the salt side of each of the heat exchangers.

Some projects use a recirculation pump between the steam drum and the cold end of the evaporator, and a separate recirculation pump between the steam drum and the cold end of the preheater. The flow capacities of each pump are greater than 15 percent of the design water flow rates, and operating the pumps early in the startup process ensures that the minimum water flow rates are always maintained in the evaporator and the preheater.

In contrast, steam flow is initiated in the superheater and in the reheater only when the blending of hot salt with cold salt begins. The steam flow rates begin at 0 percent, and increase with an increase in the flow rate of hot salt. As such, during each startup, the steam flow rates in both heat exchangers are necessarily below the minimum allowable flow rates. Due to the very low pressure drops at these flow rates, the flow distribution among the approximately 900 tubes is expected to be poor. Non-uniform flow distributions lead to non-uniform temperature distribution, which, in turn, lead to unpredictable, and potentially damaging, stress distributions.

Liability 5 - Startup Feedwater Heater

Prior to turbine synchronization, the turbine extraction feedwater heaters are not in service. As such, the temperature of the feedwater leaving the last high pressure feedwater heater is nominally equal to the saturation temperature in the deaerator. This temperature is well below the freezing point of the salt. As such, the temperature of the feedwater supplied to the preheater must be increased to a value which is high enough to prevent salt solidification at the cold end of the preheater.

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The same conditions apply once the turbine is synchronized, but with the turbine operating under part-load conditions.

A common approach to preheating the feedwater is to use a startup feedwater heater. The energy source for the startup heater is live steam, which has been throttled and attemperated.

This approach is very effective at preheating the feedwater, but it does incur a range of penalties, as follows:

- Using live steam for feedwater preheating is thermodynamically inefficient. Under part load turbine operating, the penalty in the Rankine cycle efficiency can range from 0.5 to 1.0 percentage points.
- The startup heater can undergo rapid changes in inlet steam flow. Although the steam flow is attemperated, the rates of change in the heater metal temperature can exceed typical commercial values; i.e., 10 °C/min. To what extent this influences the low cycle fatigue life of the heat exchanger is not yet known.
- The startup heater is used daily, and its operation is mandatory to start the steam generator. Further, one 100-percent heater is a typical design approach. As such, if the startup heater is not available, the plant incurs a forced outage.
- Starting the steam generator, synchronizing the turbine, and loading the turbine is an involved process. If the turbine trips due to a spurious control signal, the turbine can be quickly restarted once the trip is cleared. For this to occur, the steam generator must remain in operation; i.e., steam is dumped to the condenser through the bypass system. However, when the turbine trips, the extractions to the feedwater heaters are stopped, and the temperature of the feedwater leaving the last high pressure feedwater heater quickly decays. In response, the startup heater must be brought quickly into service. This produces a thermal shock on the tube side, in combination with a thermal shock on the shell side when live steam is introduced into the heater. To some degree, the low cycle fatigue life of the startup heater under thermal shock conditions is something of a commercial unknown.

6.7 Changes to the Equipment and the System Design

To avoid the problems experienced at a range of commercial projects, different approaches can be adopted regarding process design, fabrication techniques, and material selection.

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6.7.1 Process Design

The goals of a revised process design are as follows:

- Allow the equipment to operate continuously at the 1-percent load case without affecting the low cycle fatigue life of the equipment. The 1-percent load case is an arbitrary requirement; i.e., it is no more applicable than the 2- or the 5-percent case. But if the process design, at 1 percent load, can protect the equipment against stress distributions that could reduce the low cycle fatigue life, then the process design can comprehensively protect the equipment over the full range of part- and full-load conditions
- Establishing uniform flow distributions, on both the shell- and the tube-sides of the heat exchangers throughout the full range of startup and shutdown conditions. Uniform flow distributions avoid the potential for non-uniform temperature distributions and the associated potential for damaging stress distributions.

System Availability Requirements

Most commercial projects use two 50-percent capacity steam generator trains due, in large part, to the relatively low availability of the equipment. However, many of the availability problems can be traced to leaks in the heat exchangers. Potential sources of the leaks include the following:

- Relaxation of the tube-to-tubesheet connections in the superheater, likely due to exceeding the vendor limits on rates of temperature change and allowable fatigue cycles ⁴²
- Pitting corrosion of the stainless steel tubes in the evaporator. The corrosion is associated with solid deposits, which are generated from flow accelerated corrosion of the carbon steel equipment in the condensate system. The accelerated corrosion can be traced to inadequate control over the feedwater chemistry.

An example of a simplified, order-of-magnitude reliability and availability analysis for a steam generator system is discussed in the following sections.

Reliability Block Diagram

In at least two representative commercial projects, the steam generation system consists of the following subsystems:

⁴² Aalborg CSP, TEMA SGS/HX U-Tube Type Inspected and Repaired / Replaced by Aalborg Service Department, 2019

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- Three 50-percent capacity hot salt pumps. The 3 pumps share a common minimum flow recirculation valve
- Two 100-percent capacity cold salt attemperation pumps. The 2 pumps share a common minimum flow recirculation valve
- Two 50-percent steam generator trains. Each train consists of a superheater, a reheater, an evaporator, and a preheater. The salt flow at the hot end of each train splits into parallel flows between the superheater and the reheater. The salt flow from the cold end of the superheater combines with the salt flow from the hot end of the reheater, and the combined flow passes, in series, through the evaporator and the preheater.

Hot Salt Pump Subsystem The reliability of the pump subsystem is based on the following:

- Each pump group includes the pump, the discharge isolation valve, the discharge control valve, either the limit switches or the positioners on the valves, and the heat tracing on the valves. All of the items in the group operate in series. The reliability of the group is given by:

$$R_{Group} = \prod_{i=1}^i R_i$$

where R_i is the reliability of component i . The mean time between failure of the pump group is given by:

$$MTBF_{Group} = \frac{1}{\sum_{i=1}^i \frac{1}{MTBF_i}}$$

where $MTBF_i$ is the mean time between failure of component i .

- 2 of the 3 pump groups must be in service to provide the design flow rate. The reliability for a 2-out-of-3 arrangement is:

$$R_{2\ of\ 3} = R_{Group}^3 + 3 * R_{Group}^2 * (1 - R_{Group})$$

- The 3 pump groups operate in series with the minimum flow recirculation valve.

Attenuation Pump Subsystem The reliability of the pump subsystem is based on the following:

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- Each pump group includes the pump, the discharge isolation valve, the discharge control valve, either the limit switches or the positioners on the valves, and the heat tracing on the valves. All of the items in the group operate in series.
- The two pumps operate in parallel, and 1 of the 2 pumps must be in service to provide the design flow rate. The reliability of two components operating in parallel is given by:

$$R_{Subsystem} = R_1 * (1 - R_2) + R_2 * (1 - R_1) + R_1 * R_2$$

where R_1 is the reliability of pump group 1, and R_2 is the reliability of pump group 2. The mean time between failures of the subsystem is given by:

$$MTBF_{Subsystem} = \frac{1}{\frac{1}{MTBF_1} * \frac{MTTR_1}{MTBF_2} + \frac{1}{MTBF_2} * \frac{MTTR_2}{MTBF_1}}$$

- The in-service pump operates in series with the minimum flow recirculation valve.

Heat Exchanger Subsystem The reliability of the heat exchanger subsystem is based on the following:

- Train 1 operates in parallel with Train 2. However, both trains must be in service to provide the design steam flow rate
- In each train, the superheater operates hydraulically in parallel with the reheater. However, both the superheater and the reheater must be in operation for the train to be in operation.

Steam Generator System For the steam generator to operate, each of the 3 subsystems above must be in service. Further, the systems operate in series.

Reliability and Mean Time Between Failure Data

Reliability and failure rate data for the components in the hot salt pump and the attemperation pump subsystems are outlined in Table 6-1. For the purposes of the calculations, a reference period, T, of 4,000 hours has been assumed. This represents 1 calendar year of operation in a plant with an annual capacity factor of 45 percent.

It can be noted that there are little or no public data available on the reliability and the availability of salt steam generators in commercial service. The mean times between failures for each of the components in the table are largely based on a non-scientific survey of the experience from a range of parabolic trough and central receiver projects, using a mix of qualitative and quantitative information. In addition, the mean time between failure estimates listed for the superheaters and the evaporators represent a plant in

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which the heat exchanger reliabilities did not reach expected values. Problems with these heat exchangers have resulted in forced outage periods on the order of 10 to 14 days each. Clearly, the qualitative nature of some of the reliability information, in combination with the small sample size, means that the uncertainty in the calculated availability values is likely to be high. Nonetheless, the general trends shown in the calculations have been observed in commercial practice.

Table 6-1 Inputs to the Reliability Calculations for the Steam Generator - Current Experience

System Component	Mean time between failures,		Failure rate, 1/hr	Reliability, []	Failures per yr., each	Failure repair time, hr	Failures x Repair time, hr
	Years	Hours					
Salt pump	10	40,000	$2.50 * 10^{-5}$	0.905	0.1	48	5
Discharge isolation valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Discharge control valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Discharge flow meter	0.2	800	$1.25 * 10^{-3}$	0.007	5	1	5
Minimum flow recirculation valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Superheater / reheater ¹	2	8,000	$1.25 * 10^{-4}$	0.493	0.5	336	168
Evaporator ²	0.5	2,000	$5.00 * 10^{-4}$	0.243	2	240	480
Preheater	30	120,000	$8.33 * 10^{-6}$	0.905	0.033	336	11
Salt piping	50	200,000	$5.00 * 10^{-6}$	0.980	0.02	48	1
Heat tracing ³	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12

Notes:

1. The superheater and the reheater are supplied with salt in a parallel arrangement; however, both heat exchangers must be in service for the steam generator to be in service. The mean time between failures for the superheater / reheater combination is defined by the mean time between failures for superheater, which is the more problematic of the two due to thicker metal sections associated with higher operating pressures.
2. Evaporator outages are due to tube leaks associated with inadequate control over the water chemistry.
3. There are perhaps 100 heat trace circuits in the steam generation system. The reliability values reflect not a failure of an individual circuit, but a system level failure that would prevent the steam generator from operating.

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For the purposes of the calculations, a Weibull probability distribution has been assumed. The distribution of the time to failure, f (Time; Scale parameter, Shape parameter), is given by:

$$(\text{Shape} / \text{Scale}) (\text{Time} / \text{Scale})^{(\text{Shape} - 1)} \exp (-(\text{Time} / \text{Scale})^{\text{Shape}})$$

The cumulative probability of failure, F (Time; Scale parameter, Shape parameter) is given by:

$$\exp (-(\text{Time} / \text{Scale})^{\text{Shape}})$$

and the Reliability is $1 - (\text{Cumulative probability of failure})$.

In the reliability estimate for each of the components listed in Table 6-1, the scale parameter and the shape parameter are both set to values of 1. This is done as there is limited availability of information to support a value other than 1. However, the scale parameter for the superheater and the evaporator are both set to a (somewhat arbitrary) value of 0.5. This is to reflect the damage experienced by these heat exchangers in the early years of commercial service. Damage to the superheater is associated with high rates of temperature change, during startup and shutdown, for the period required to develop the tuning parameters for the control system. Damage to the evaporator is due to corrosion associated with inadequate control over the water chemistry.

System Availabilities Based on Current Heat Exchanger Experience

An estimate of the subsystem and the system availabilities for plants with two 50-percent heat exchanger trains is listed in Table 6-2.

Table 6-2 Steam Generator Availability - Current Experience

Subsystem	Mean Time Between Failures, hr.	Availability
Hot salt pumps	111	0.959
Attemperation pumps	593	0.992
Heat exchangers	741	0.737
Overall system	50	0.701

It can be noted that the availability of commercial solar projects in financial models are often quoted to be in the range of 90 to 92 percent. The steam generator necessarily operates in series with the heliostat field, the receiver system, the storage system, the Rankine cycle, and the balance of plant systems (i.e., water treatment and compressed air). A representative set of availabilities (in some cases, availability goals) for these systems is outlined in Table 6-3.

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Table 6-3 System and Plant Availabilities - Current Experience

	Availability
Heliostat field	0.985
Receiver system	0.975
Thermal storage system	0.999
Steam generation system	0.701 ¹
Rankine cycle	0.990
Balance of plant	0.995
Total plant	0.662

Note:

1. Current experience, with two 50-percent trains

From the table, the following can be seen:

- For a project to meet its projected financial requirements, the reliability of the heat exchangers must be improved. This will require changes in items such as process control, fabrication methods, and water chemistry monitoring
- From the reliability information listed in Table 6-1, the availability of one 50-percent capacity steam generator train is on the order of 0.848. However, both trains must be in service to provide the design steam flow rate. For trains in parallel service, but in an arrangement in which both trains must operate, the combined availability of the two trains is $0.848 * 0.848 = 0.737$. If the availability of one 50-percent train is below commercial requirements, then adding a second 50-percent train, of the same design and the same availability, does not improve the availability of the system.

System Availabilities Based on Projected Heat Exchanger Experience

If the problems with process temperature control and feedwater chemistry can be solved, then the reliability of a single steam generator train should improve to the values shown in Table 6-4. In the component reliability calculations, the Weibull distribution scale parameter and shape parameter are set to values of 1 to reflect the long-term maturity of the design.

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Table 6-4 Reliability Analysis for a Single Steam Generator Train - Projected Performance

System Component	Mean time between failures,		Failure rate, 1/hr.	Reliability, []	Failures per yr., each	Failure repair time, hr.	Failures x Repair time, hr.
	Years	Hours					
Hot salt pump	10	40,000	$2.50 * 10^{-5}$	0.905	0.1	48	5
Discharge isolation valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Discharge control valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Discharge flow meter	0.2	800	$1.25 * 10^{-3}$	0.007	5	1	5
Minimum flow recirculation valve	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12
Superheater / reheater ¹	10	40,000	$2.50 * 10^{-5}$	0.905	0.1	240	24
Evaporator ¹	10	40,000	$2.50 * 10^{-5}$	0.905	0.1	240	24
Preheater ¹	10	40,000	$2.50 * 10^{-5}$	0.905	0.1	240	24
Salt piping	50	200,000	$5.00 * 10^{-6}$	0.980	0.02	48	1
Heat tracing ²	1	4,000	$2.50 * 10^{-4}$	0.368	1	12	12

Notes:

1. Forced outages for tube plugging, which are not coordinated with the scheduled plant-wide maintenance outages.
2. There are perhaps 100 heat trace circuits in the steam generation system. The reliability values reflect not a failure of an individual circuit, but a system level failure that would prevent the steam generator from operating.

With these values, the calculated system availability in a mature design is estimated to be 0.89, as noted in Table 6-5.

To reach a system availability of 0.89, the following must occur:

- The heat exchangers, which have the largest influence on the system availability, must operate with a mean time between failures of perhaps 10 years

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Table 6-5 Steam Generator Availability - Projected Performance

Subsystem	Mean Time Between Failures, hr.	Availability
Hot salt pumps	111	0.959
Attemperation pumps	593	0.992
Heat exchangers	5,000	0.939
Overall system	64	0.894

- The allowable types of heat exchanger failures must be restricted to leaks in a small number of tubes; i.e., there are no wholesale failures of the tube-to-tubesheet connections or systemic tube corrosion
- The cumulative number of tubes requiring plugging is no more than the allowance provided by the vendor.

To satisfy these requirements, the process temperature control and the feedwater chemistry must always be correct throughout the life of the project.

Potential for a Single Train Design

If the availability of the steam generation system is increased to a value of 0.89, then the availability of the overall plant is increased to a nominal value of 0.85, as follows:

Table 6-6 System and Plant Availabilities - Projected Performance

	Availability
Heliostat field	0.985
Receiver system	0.975
Thermal storage system	0.999
Steam generation system	0.894 ¹
Rankine cycle	0.990
Balance of plant	0.995
Total plant	0.845

Note:

1. Projected performance, with two 50-percent trains

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However, a plant availability of 0.85 is still below commercial requirements. To improve the plant availability, two potential approaches for improving the availability of the steam generation system include the following:

- As shown in Table 6-5, the heat exchangers have the largest influence on the system availability. One approach to improving the availability of the heat exchangers is to switch from two 50-percent trains to one 100-percent train. By doing so, this avoids the following effect on reliability of two trains, operating in parallel, required to meet a design condition:

$$\text{System Reliability} = (\text{Reliability}_{\text{Train 1, 50 percent capacity}}) * (\text{Reliability}_{\text{Train 2, 50 percent capacity}})$$

- Replacing the three 50-percent hot salt pumps with two 100-percent hot salt pumps.

With these changes, the availability of the steam generation system improves to a value of 0.95, and the availability of the overall plant increases to a value of 0.90. A value of 0.90 is consistent with the projections often proposed in financial models.

Informal discussions with steam generator vendors indicate that the heat exchangers for a representative 125 MWe plant can be built in the sizes required for one train. As such, the single train design should be preferred over the two 50-percent train approach.

It can be noted the typical commercial approach to the design of the hot salt pumps (i.e., three 50-percent pumps) allows the use of one hot salt pump to supply the two steam generator trains during startup and shutdown. Compared to one 100-percent pump, the use of one 50-percent pump provides a better match between the H-Q characteristics of the pump with the flow and pressure drop characteristics of the heat exchangers during part load conditions. Nonetheless, using one 100-percent for startup and shutdown can result in acceptable pump performance as long as the following occurs:

- A hydraulic analysis of the steam generation system is conducted to confirm that pump is always operating at acceptable combinations of pump speed, flow, and total developed head
- Simultaneously, the Cv characteristics of the split-range salt control valve at the cold end of the preheater are selected such that the operating points of the hot salt pump, operating in parallel with the attemperation pump, provide the required hydraulic stability.

It should be noted that the reliability calculations in Table 6-1 and Table 6-4 include only the major equipment items. A more thorough analysis would include items such pressure instruments, temperature instruments, heat trace circuits, number of tube-to-tubesheet connections, permissives and interlocks in the logic for the distributed control system, and communication errors. As such, the estimates of availability should be viewed as indicative, rather than quantitative. Nonetheless, the general trends regarding the minimum acceptable values for equipment reliabilities should largely be applicable.

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Also, reducing the number of trains by a factor of 2 (i.e., one 100-percent train rather than two 50-percent trains) means that the number of equipment items are also reduced by a factor of 2. Reducing the number of equipment items operating in series generally improves the system availability.

Independent Trains

Even if two 50-percent steam generator trains are selected for a commercial project, each train should be independent of the other. Each train would consist of one 100-percent capacity hot salt pump, one 100-percent capacity attenuation pump, the steam generator heat exchangers, and a steam bypass system to the condenser. The salt pump arrangement is illustrated in Figure 6-3, and the intra-heat exchanger salt valve arrangement is shown in Figure 6-4. (Clearly, the analysis described below for two 50-percent independent trains can readily be extended to one 100-percent independent train.)

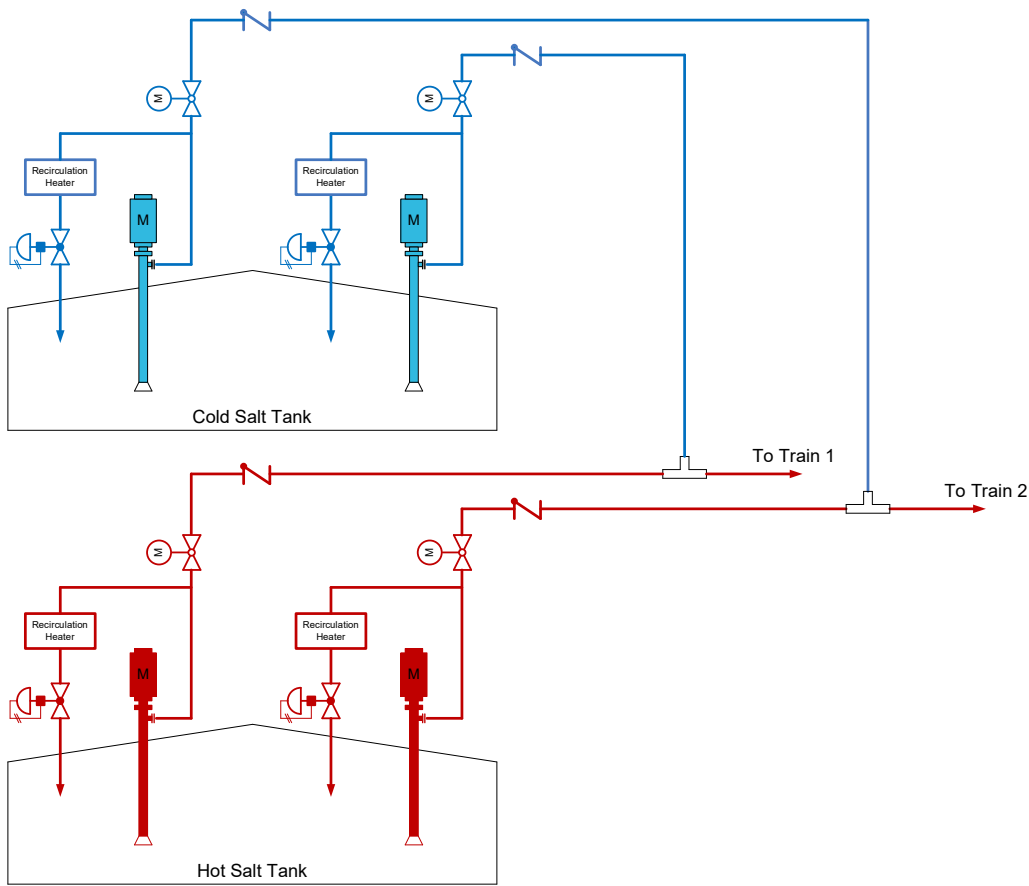


Figure 6-3 Salt Pump Arrangement with Independent Steam Generator Trains

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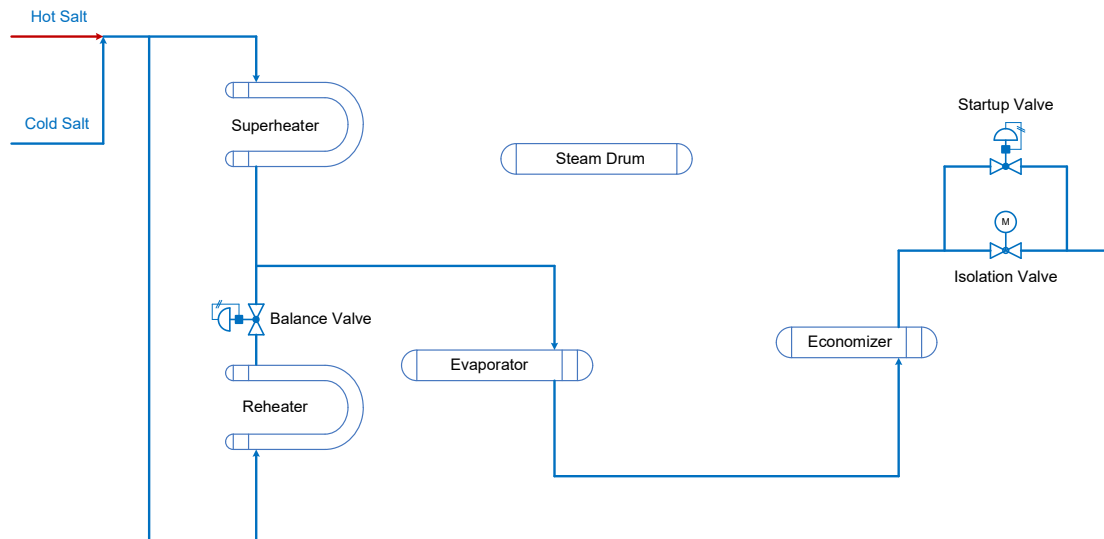


Figure 6-4 Intra-Heat Exchanger Salt Valve Arrangement

Independent trains allow either train to start at any time, regardless of the operating state of the other train. In addition, independent trains avoid a situation in which:

- Multiple pumps share a common discharge header
- Multiple pumps share a common minimum flow recirculation valve
- Multiple hot salt pumps, in parallel with multiple attemperation pumps, share a common mixing station upstream of the superheater / reheater.

The sharing of headers and mixing stations means that the discharge pressure of one pump influences the flow from any one, and perhaps more than one, of the other pumps operating in parallel.

Further, it is recommended that a split-range startup salt valve be added in parallel to the normal isolation valve at the cold end of the preheater. With the isolation valve closed, the startup valve performs two important functions: 1) the valve adds an artificial pressure drop to the system, and 2) the hot salt pump and the cold salt pump now see a common discharge pressure. These function provide the following benefits:

- If the pressure drop in the system is increased, then the pumps are no longer obligated to operate at their minimum speeds. As such, varying the pump speed now becomes a potential means of system control. The value in this approach is that pump speeds can be both accurately measured and accurately controlled.

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- If the head - flow characteristics of the pumps are well defined, and if the flow - pressure drop characteristics of the piping, the valves, and the heat exchangers are well defined, then it is possible to both calculate and to accurately control the relative flow rates of hot salt and cold salt. This, in turn, allows the salt temperature at the mixing station upstream of the superheater / reheater to be controlled by changing the relative speeds of the pumps.
- The control approach avoids the need to match the discharge pressures from the two pumps using signals from pressure transmitters, or to set the flow rates from the two pumps using signals from flow meters. The accuracy of salt pressure and flow instruments is generally too low to control an acceleration in the hot salt flow rate on the order of 0.0023 m³/hr-hr.
- It is possible to select a discharge coefficient for the split range startup valve that allows the minimum flow recirculation valves for both the hot salt pump and the attemperation pump to be set to fixed positions for steam generator flow rates beginning with startup, through turbine synchronization, and then to an intermediate turbine load. As such, all of the system valves remain in fixed positions, and both of the salt pumps see a common discharge pressure, which reduces the potential for oscillations in the mixed salt temperature upstream of the superheater / reheater.

The above approach to steam generator system control during startup was successfully demonstrated at the Solar Two demonstration project ⁴³.

Pump Redundancy

A spare hot salt pump, and a spare attemperation pump, would be stored in the site warehouse. The reliability of salt pumps has been excellent, and providing an installed redundant pump can be counterproductive to the plant availability. Specifically, each pump requires an isolation valve, a startup control valve, a check valve (see below), and a vent valve.

Electric Salt Heaters in Minimum Flow Recirculation Lines

One example is the piping arrangement for the external electric salt heaters for the salt storage tanks. At one commercial project, two 100-percent salt heaters could receive salt from any of the 3 hot salt pumps in the hot tank, or salt from either of the 2 cold salt attemperation pumps in the cold tank. However, a total of 9 salt valves were required to either direct the flow as needed or to drain the equipment. In contrast, an alternate design would locate a separate electric heater in the minimum flow recirculation line for each of the salt pumps. Each heater would be located downstream of the minimum flow recirculation valve, and each heater would be located at an elevation above the salt tanks. The minimum

⁴³ Kelly, B (Bechtel National Inc., San Francisco, California), 'Lessons Learned, Project History, and Operating Experience of the Solar Two Project', Sandia Report SAND2000-2598, November 2000

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flow valve would keep the electric heater flooded during operation, and the piping would be arranged such that the piping and the heater drain automatically when the pump is not in operation. Granted, a typical project might require 2 (or 3) hot salt pumps and 2 attemperation pumps, and as such, would require 4 (or 5) electric salt heaters. Further, the 4 (or 5) heaters could be more expensive than two 100-percent capacity heaters. However, the improvement in plant availability due to the elimination of 9 salt valves will more than offset the marginal cost of the smaller electric heaters. Plus, the use of salt heaters on each of the pumps provides essentially the same redundancy as the two 100-percent heaters in the example commercial project.

Pump Discharge Check Valves

Conventional swing check valves should be provided in the discharge lines on each of the salt pumps. During the early stages of the startup process, the required flow rate of hot salt is very small; something in the range of 2 to 20 m³/hr. If the flow of hot salt were to be initiated by selecting a pump speed, and then opening an isolation valve in the pump discharge line, the likelihood that a flow as small as 2 m³/hr could be established is very low. The more likely scenario is that one of the following conditions will occur:

- The discharge pressure of the hot salt pump is slightly too high, in which case the initial flow of hot salt is too high (20 m³/hr).
- The discharge pressure of the hot salt pump is slightly too low, and cold salt starts to flow backwards towards the hot salt pump.

In either condition, it will not be possible to provide the required control of the mixed salt temperature at the entrance to the steam generator.

The best method for accomplishing this task is to provide a conventional swing check valve at the pump discharge. To initiate a flow, a hot salt pump speed is selected which provides a discharge pressure that is slightly below the cold salt pump discharge pressure. The differential pressure causes the hot salt pump check valve to close. (The pump is protected from overheating by the flow through the minimum flow recirculation line.) The speed of the hot salt pump is then increased in increments of 0.1 percent. The discharge pressure of the hot salt pump eventually rises to a value that is slightly greater than the cold salt pump discharge pressure, the check valve starts to open, and a very small, and stable, flow of hot salt is established.

The shutdown procedure for the steam generator is essentially the mirror image of the startup procedure. As such, the requirement for a check valve on the hot salt pump during startup is mirrored by a requirement for a check valve on the cold salt attemperation pump during shutdown.

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6.7.2 Startup and Shutdown Procedures

A proposed heat exchanger and salt valve arrangement is shown in Figure 6-5. The startup valve adds an artificial pressure drop during startup and shutdown. Valve 1 and Valve 2 direct the flow of hot salt early in the startup, and late in the shutdown, process.

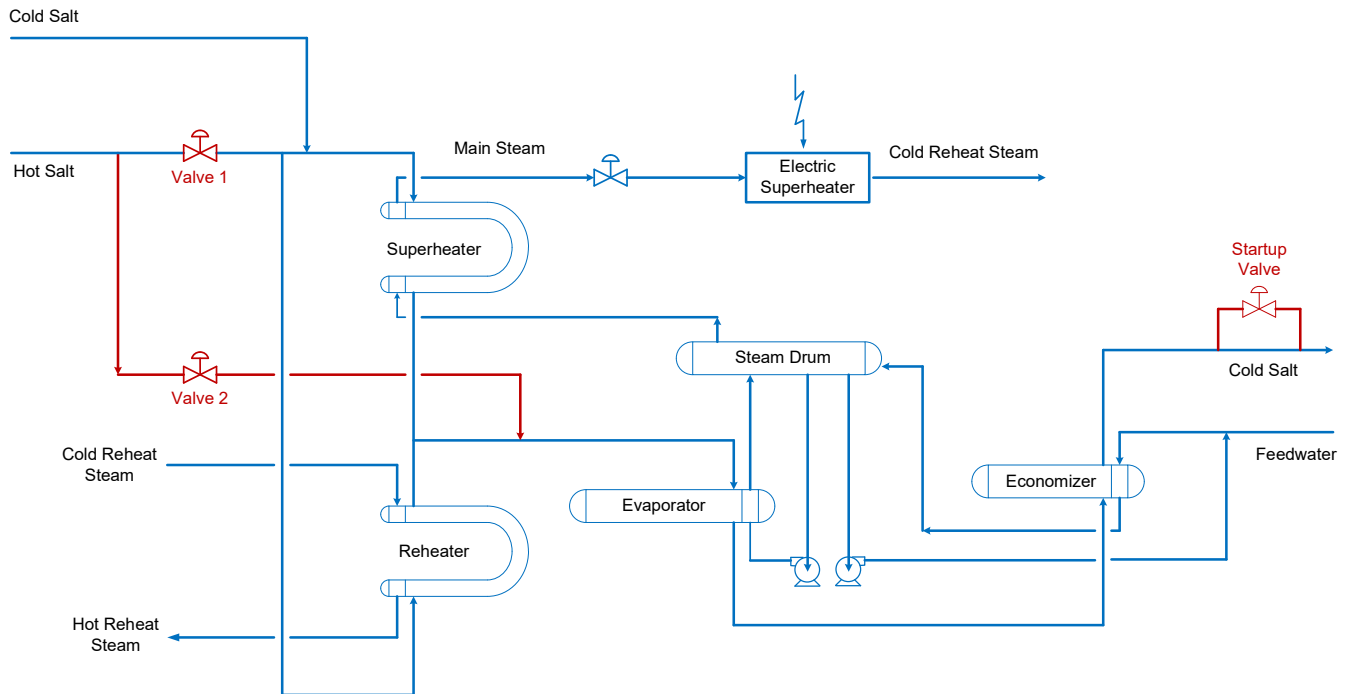


Figure 6-5 Steam Generator Heat Exchanger and Valve Arrangement

Early in the startup process, the temperature of the mixed salt upstream of the superheater / reheater is close to the cold salt temperature. As such, the temperature of the live steam at the hot end of the superheater is also close to the cold salt temperature. To ensure that the temperature of the salt at the cold end of the preheater does not challenge the salt freezing temperature, the saturation pressure in the steam drum is maintained at a nominal value of 100 bar. When live steam from the superheater at 100 bar is throttled to a pressure which can be accepted by the reheater (~ 30 bar), the temperature of the cold reheat steam can drop close to, or slightly below, the freezing temperature of the salt. To prevent salt from freezing at the cold end of the reheater, the temperature of the cold reheat steam is raised by means of an electric steam superheater.

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Isothermal Steam Production

As discussed in Section 6.5, the steam flows in the superheater and in the reheater during startup necessarily span the range of 0 percent to the minimum flow rate allowed by the vendor (~ 16 percent). Low flow rates often produce poor flow distributions, which raise the possibility of undesirable temperature and stress distributions. However, if the heat exchanger is operating under isothermal conditions between flow rates of 0 and 16 percent on both the shell-side and the tube-side, then the potential for damaging stress distributions is largely eliminated.

One method for operating the superheater and the reheater under isothermal conditions is to establish a shell-side flow rate of cold salt at a temperature of 295 °C, and simultaneously establish a tube-side flow rate of steam at a temperature of 295 °C. The shell-side conditions are established by operating the cold salt attemperation pump at the design speed. The tube-side conditions are established by 1) selecting a set point pressure of 80 bar for the throttle valves in the steam bypass system to the condenser, and 2) supplying hot salt to a mixing station upstream of the evaporator, and 3) isolating the hot salt flow to the mixing station upstream of the superheater / reheater.

Step 0 - Overnight Hold

During overnight hold, cold salt is supplied to the steam generator, as shown in Figure 6-6. No steam production occurs, and the temperature of the salt passing through the heat exchangers decays only slightly due to the moderate heat losses from the heat exchangers and the system piping.

The recirculation pump from the steam drum to the cold end of the preheater is in service. Steam drum makeup water from the Rankine cycle combines with the recirculation flow at a point just upstream of the preheater. The direct contact heat exchange ensures that the relatively cold makeup water does not cause salt to solidify at the cold end of the preheater.

The startup salt valve at the cold end of the preheater is set to a position which provides the desired additional pressure drop for the salt pumps later in the startup process. A set of hydraulic calculations for a representative commercial project show that the startup valve can be set to a fixed position for Steps 0 through 4 discussed below.

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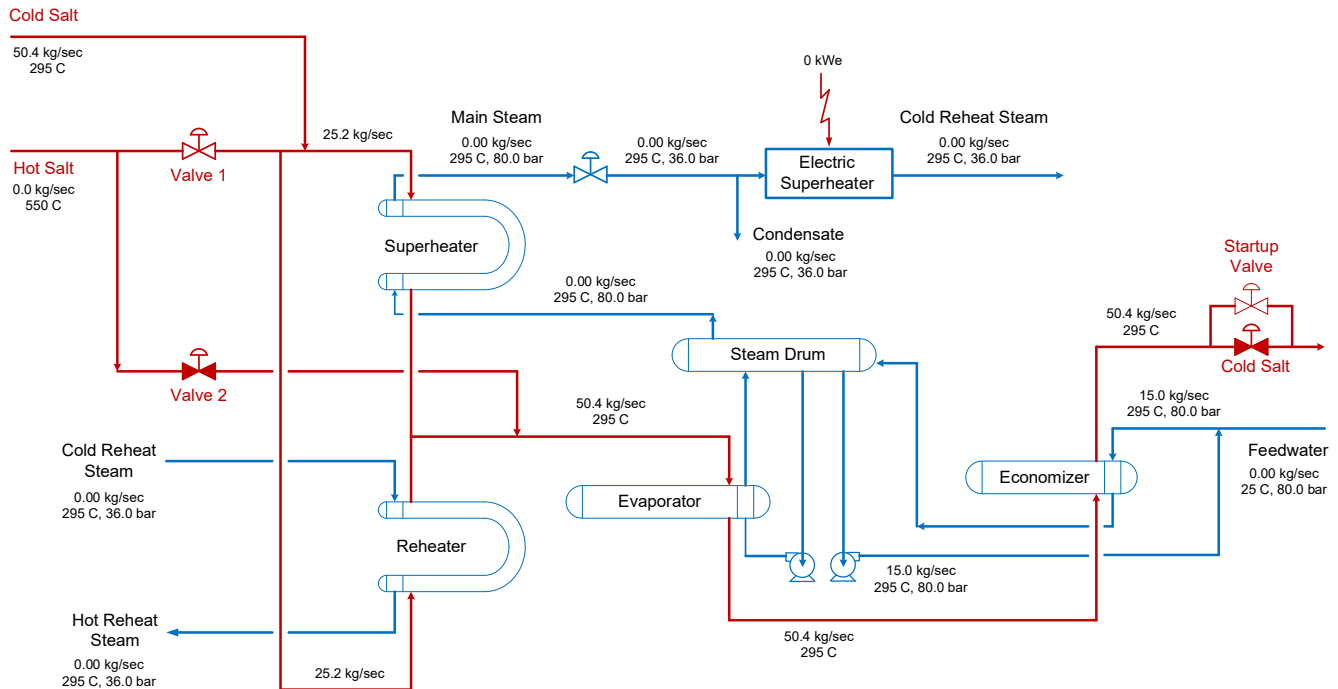


Figure 6-6 Step 0 - Overnight Hold

Step 1 - Isothermal Steam Production

The basic flow arrangement for the first step in this process is shown in Figure 6-7. The salt and steam flow rates are based on a nominal 125 MWe Rankine cycle with two 50-percent steam generator trains.

Cold salt, at a temperature of 295 °C, is supplied to the hot end of the superheater and to the hot end of the reheater. Simultaneously, saturated steam, at a temperature of 295 °C, is supplied to the superheater, and cold reheat steam, at a temperature of 295 °C, is supplied to the reheater. As such, no heat transfer occurs in either heat exchanger, and the temperature of the salt leaving both heat exchangers is 295 °C.

At the salt mixing station upstream of the evaporator, hot salt is added to the flows leaving the superheater and the reheater. This raises the temperature of the salt entering the evaporator to 428 °C, which provides the thermal energy required to 1) generate saturated steam, at a temperature of 295 °C, for delivery to the superheater and the reheater, and 2) establishes steam flow rates equal to 20 percent of the design flow rates in the both superheater and the reheater.

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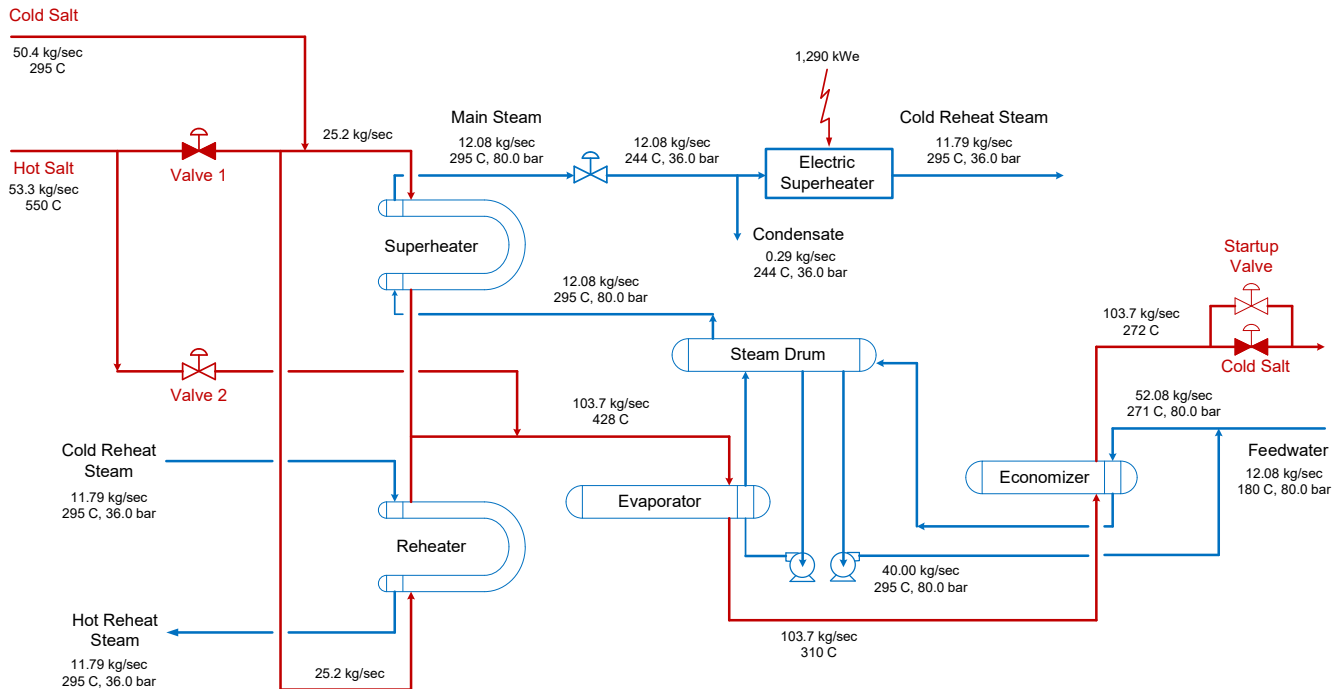


Figure 6-7 Step 1 - Isothermal Steam Production

The saturated steam pressure of 80 bar is established in the drum by the high pressure throttle valve in the steam bypass system to the condenser. A steam flow rate equal to 20 percent of the design flow rate is larger than the minimum required by the vendor (16 percent). However, the steam flow rate decays to 16 percent in the Step 2 below.

Throttling live steam, at a pressure of 80 bar, to the maximum pressure which can be accepted by the reheater (36 bar), results in a decrease of the steam temperature to the saturation temperature corresponding to 36 bar (244 °C). The quality of the steam leaving the high pressure throttle valve is on the order of 97 percent, and 0.3 kg/sec of condensate is removed in a trap prior to the steam entering the cold end of the reheater. To ensure that no heat transfer occurs in the reheater, the temperature of the cold reheat steam is increased to the cold salt temperature (295 °C) by means of a 1.3 MWe electric steam superheater.

The flow rate and temperature of the makeup water from the Rankine cycle are 12.1 kg/sec and 180 °C, respectively. The feedwater temperature assumes that auxiliary steam, at a nominal pressure of 10 bar, is supplied to the deaerator for preheating. The recirculation of saturated water from the drum to the cold end of the preheater raises the mixed feedwater temperature to a value of about 271 °C, which is high enough to prevent salt from freezing at the cold end of the preheater.

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As a technical note, all commercial solar projects use an auxiliary steam generator for equipment preheating and to supply steam to the turbine shaft seals to establish a vacuum in the condenser. An auxiliary steam generator is required, as steam generator vendor generally does not permit the use of the main evaporator for small steam flow rates over extended periods of time.

Superheated Steam Production

Step 2 - Switching the Salt Mixing Station

Once flows of 20 percent have been established on the tube-sides of the superheater and the reheater, the hot salt / cold salt mixing station is switched from a point upstream of the evaporator to the normal point upstream of the superheater / reheater, as shown in Figure 6-8.

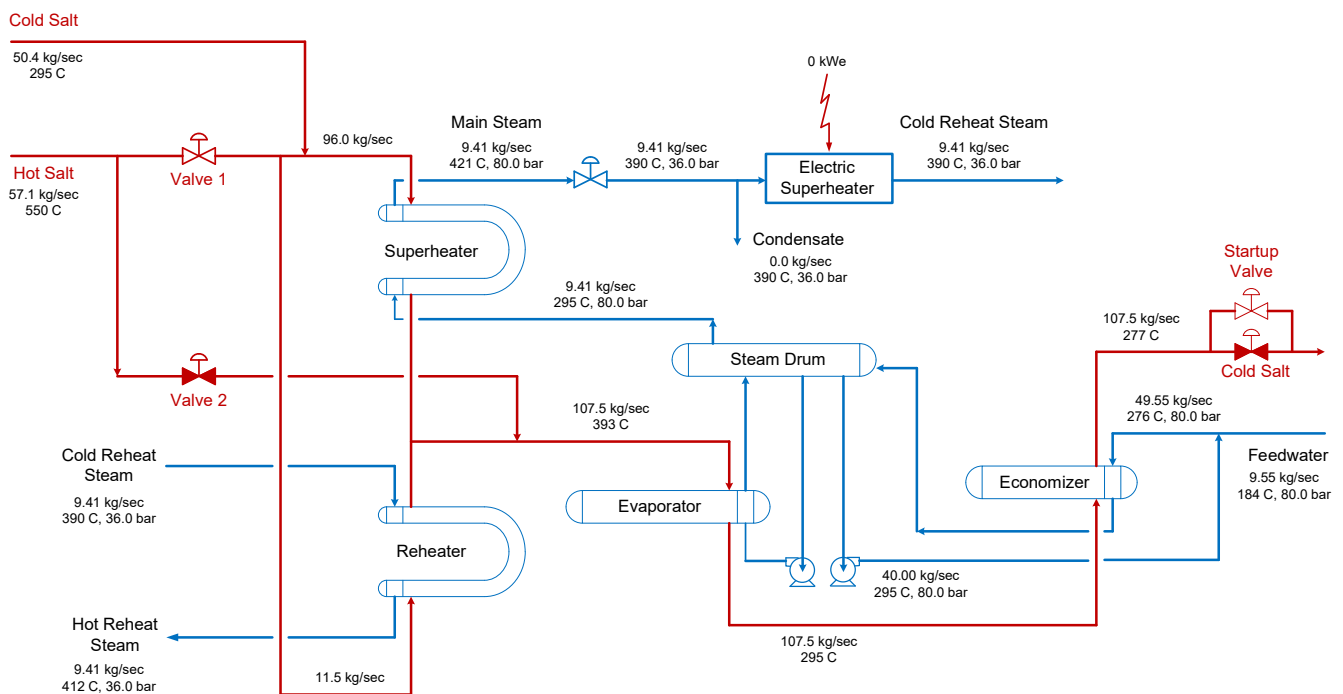


Figure 6-8 Step 2 - Live Steam and Hot Reheat Steam Production

The transition basically involves opening Valve 1 and closing Valve 2 at rates which keep the following sum a constant value: $C_v \text{ Valve 1} + C_v \text{ Valve 2}$. Under these conditions, no changes occur in either the hot salt or the cold salt flow rates, and the transition should be hydraulically stable. The rates at which Valve 1 and Valve 2 move are:

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- Amenable to calculation, based on curves of Cv versus valve position
- Programmed in the DCS such that the mixed salt temperature at the hot end of the evaporator decreases at no more than the vendor limit of 10 °C/min
- Programmed in the DCS such that the mixed salt temperature at the hot end of the superheater / reheater increases at no more than 10 °C/min.

Moving the mixing station to a point upstream of the superheater / reheater raises the temperature of the salt entering both the superheater and the reheater. As such, heat transfer now begins in the superheater and in the reheater, which would normally increase the thermal demand on the steam generator. However, one of the goals of the transition is to maintain the cold salt and the hot salt flow rates fixed in both Step 1 and Step 2. With the flow rates fixed, the feedwater flow rate decreases to a new value of 9.55 kg/sec to offset the increased thermal demands of the superheater and the reheater. A feedwater flow rate of 9.55 kg/sec corresponds to the 16 percent minimum value specified by the vendor.

Step 3 - Turbine Synchronization

At the end of Step 2, the steam generator has nominal live steam and hot reheat steam flow rates of 16 percent, and live steam and hot reheat steam temperatures slightly above 410 °C. This combination of steam flow rate and temperature should be adequate to admit steam to the turbine under the conditions of a warm start, accelerate the turbine to synchronous speed, synchronize the turbine, and load the turbine to a value which is high enough to prevent a trip on reverse power. (If the turbine is undergoing a cold start or a hot start, then steam can be admitted to the turbine starting when the steam temperatures are lower than, or higher than, 410 °C, respectively.)

The steam supplied to the turbine is diverted from the turbine bypass system. Whatever steam is not required by the turbine continues to be bypassed to the condenser.

Step 4 – Low Load Operation

Once the turbine is synchronized, the extraction feedwater heaters are placed in service. However, saturated water from the steam drum continues to be recirculated to the cold end of the preheater to maintain the temperature of the salt at the cold end of the preheater at acceptable values.

To increase the thermal output of the steam generator, and to increase the live steam and the hot reheat steam temperature, the speed of the hot salt pump is increased. The increase in the discharge pressure of the hot pump will result in a decay in the flow rate from the attemperation pump. The rate at which the speed of the hot salt pump is increased is such that the mixed salt temperature at the hot end of the superheater / reheater satisfies the vendor limit of 10 °C/min.

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At some point in this transition, the speed of the attemperation pump will need to be reduced. The rate at which the speed of the attemperation pump is reduced is such that the mixed salt temperature at the hot end of the superheater / reheater satisfies the vendor limit of 10 °C/min. Eventually, the flow rate from the attemperation pump stalls. A check valve on the discharge of the attemperation pump prevents a reverse flow through the pump. At this point, the attemperation pump can be stopped, and only salt from the hot pump is supplied to the steam generator.

Once the steam generator trains reaches its design temperature profile (565 °C at the hot end of the superheater / reheater, and 295 °C at the cold end of the preheater), the salt isolation valve at the cold end of the preheater can be opened and the startup valve closed. Opening the isolation valve will reduce the pressure drop in the system, and the speed of the hot salt pump will need to be reduced to maintain a nominally constant flow rate of hot salt. It's unlikely that the valve transition can be accomplished without a change in the hot salt flow rate. However, even if the flow rate does change, the normal temperature profile along the steam generator has already been established, and the potential for thermally shocking the heat exchangers is very low.

Throughout Steps 1 through 4 above, the goals of the process design are as follows:

1. Avoid the use of pressure and flow measurements to set the relative flow rates of hot salt and cold salt. The accuracy of the pressure and flow instruments are inconsistent with the accuracy with which the flow rate of hot salt must be accelerated to satisfy the vendor limits on rates of temperature change.
2. During startup and shutdown, the salt-side pressure drops are very low. To reduce the potential for oscillations in the flow rates from pumps operating in parallel, the 1) the system pressure drop is artificially increased by means of a startup valve, and 2) the following valves are set to fixed positions: the startup valve at the cold end of the preheater; the minimum flow recirculation valve for the hot salt pump; and the minimum flow recirculation valve for the attemperation pump.
3. Process control inputs should be limited to those items which are accurate and reliable. In general, this limits the inputs to temperature measurements.
4. Process control outputs should be limited to those items which can be accurately measured and controlled. To a first order, this includes the speeds of the hot salt and the attemperation pump.
5. Items 2, 3, and 4 are satisfied if the mixed salt temperature to, and the thermal duty of, the steam generator are set by adjusting the relative speeds of the hot salt and the attemperation pumps.
6. During various stages of startup and shutdown, the flows from the hot salt and the attemperation pump will approach a value of zero. To prevent reverse flows through the pumps, conventional

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check valves should be provided in the discharge lines. A total of 4 swing check valves were used at the Solar Two demonstration project, and the valves proved to be reliable.

To shut down the steam generator, the turbine is brought to the minimum load, and then tripped to preserve the metal temperatures in preparation for the next startup. The steam generator is shut down in essentially a reverse of the 4 steps above.

Part-Load Steam Generator Models

In Steps 1 through 4, the heat exchangers are operating under part-load conditions. The independent parameters under which the steam generator is operating include the following:

- The hot salt and the cold salt temperatures
- The feedwater flow rate, temperature, and pressure
- The water recirculation flow rates to the cold end of the preheater and to the cold end of the evaporator
- The saturation pressure in the drum, as set by the high pressure throttle valve in the bypass system to the condenser, and the cold reheat steam pressure, as set by the low pressure throttle valve in the bypass system to the condenser
- The evaporator blowdown flow rate
- The saturated steam flow rate to the superheater and the cold reheat steam flow rate to the reheater.

This leaves a number of dependent operating parameters, which must be calculated:

- Hot salt and cold salt flow rates
- Feedwater temperature at the hot end of the preheater
- Live steam and hot reheat steam temperatures
- Salt temperatures at the cold end of the superheater, the cold end of the reheater, the hot end of the evaporator, the cold end of the evaporator, the hot end of the preheater, and the cold end of the preheater.

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The calculation involves the following steps:

1. A trial salt temperature profile along the steam generator train is assumed
2. A trial salt flow rate profile is developed, based on matching the duties on the salt side with the duties on the water/steam side
3. From the shell- and tube-side flow rates, heat transfer coefficients are calculated on the shell- and tube-sides. From these, in combination with shell- and tube-side fouling factors, an overall heat transfer coefficient is calculated for each heat exchanger
4. The required log mean temperature differences in each heat exchanger are calculated from the following expression:

$$LMTD = \frac{Q}{U_{Overall} * A}$$

where LMTD is the log mean temperature difference, °C, calculated in the usual manner

Q is the heat exchanger duty, W

$U_{Overall}$ is the overall heat transfer coefficient, W/m²-°C, calculated in 3 above

A is the heat transfer area, m², calculated at the design point conditions

5. The actual log mean temperature differences in each heat exchanger are calculated from the following temperatures: salt in; salt out; water/steam in; and water/steam out.
6. If the required and the actual log mean temperature differences differ, then a perturbation is made to the salt inlet temperature on a particular heat exchanger. A new salt flow rate is calculated to match the duties on the salt and the water/steam sides, from which new values are calculated for the overall heat transfer coefficient, the required log mean temperature difference, and the actual log mean temperature difference.
7. Full convergences are first developed for the superheater and the reheater. The preheater is then converged, with new calculations developed for the superheater and the reheater at each step in the convergence of the preheater. The evaporator is then converged, with new convergences for the superheater, the reheater, and the preheater at each step in the convergence of the evaporator.

The convergence routines are required to ensure that 1) the minimum allowable shell- and tube-side flow rates are always satisfied, 2) the temperature of the salt at the hot end of the evaporator in Step 1 does not exceed the allowable design value, and 3) the temperature of the salt at the cold end of the preheater is always above the minimum design value.

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6.7.3 Static Salt Mixers

During steam generator startup and shutdown, a representative allowable rate of temperature change for the superheater and the reheater is 10 °C/min. At one 125 MWe commercial project, the temperature change rate translates to an allowable rate of change (i.e., acceleration) in the hot salt flow rate of 25 liters/min per minute. As a point of reference, the design flow rate for the hot salt pump is 14,200 liters/min. As such, the allowable rate of change in the hot salt flow rate is 0.17 percent of the design value rate per minute. A comprehensive process design is required to satisfy this limit. The process design must define, as a function of time, the following parameters:

- Hot salt pump: speed, position of the discharge control valve, and position of the minimum flow recirculation valve
- Attenuation pump: speed, position of the discharge control valve, and position of the minimum flow recirculation valve
- Position of the startup control valve at the cold end of the preheater.

Any variation or oscillation in the set point of 25 liters/min per minute will result in a variation or an oscillation in the mixed salt temperature at the hot ends of the superheater and the reheater. One potential method for modulating or damping the variation would be to provide a static mixer at the hot salt / cold salt mixing station upstream of the superheater and the reheater.

There are potentially two conditions under which the static mixer could be operating.

Condition 1: The process design can satisfy the 25 liter/min per minute requirement.

If the relative proportions of cold salt / hot salt are correct, then the purpose of the static mixer would be to limit the degree of radial stratification in the salt temperature in the pipe.

If the flow, when it reaches the superheater / reheater, has a radial stratification, then this could generate a non-predictable temperature, and a non-predictable stress, distribution in the shell near the inlet nozzle. The distribution belt inside the heat exchanger will dissipate much, or all, of this stratification. Nonetheless, there is little design information on what the allowable radial stratifications might be at the inlet to the heat exchanger. A possible design approach might be to: 1) develop a CFD model to estimate the degree to which the radial stratification dissipates as a function of distance between the mixing station and the inlet to the superheater / reheater, and 2) develop a CFD / FEA model of the entrance to the heat exchanger to determine if the residual radial stratification imposes unacceptable local thermal stresses.

Three outcomes are possible:

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1. The largest radial stratifications are likely to occur early in the startup process, or late in the shutdown process, when the salt flow rates are the lowest. The CFD analyses show that radial stratification levels during low flow conditions do not present a problem with heat exchanger stresses, and the mixer can safely be dropped from consideration.
2. The inverse to the above is that a mixer is required to limit the radial stratification. This is, nonetheless, a useful result, as it quantifies the need for, and the benefits of, a mixer. A note: The mixer would likely be designed for the low flow rates associated with startup and shutdown. As a result, the pressure drop of the mixer during full load conditions could have a measurable effect on the annual auxiliary energy demand.
3. The CFD / FEA analyses might be considered to be overly expensive, too lengthy in terms of the project schedule, or the results might be viewed as inconclusive. The project would then, in effect, be forced into making a guess on the need for a mixer. If so, one possible approach would be to install a mixer as, in essence, a brute force form of insurance. Granted, the penalty would be an increase in the auxiliary energy demand of the salt pumps. However, the benefit would be a reduction in the uncertainties regarding the transient thermal stresses in the superheater and in the reheater.

Condition 2: The process design does not satisfy the 25 liter/min per minute requirement.

The consequences of Condition 2 have been observed in one commercial project, and may well be present in other projects. The flow characteristic is similar to a plug of cold salt, followed by a plug of hot salt, followed by a plug of cold salt, and so on. The flow develops, in effect, a longitudinal temperature stratification.

Under these conditions, there is a basic deficiency in the process control. A static mixer would have no effect on the longitudinal stratification, and there would be no benefit to installing a mixer.

Summary

The process design must, unambiguously, have the control authority to satisfy the 25 liter/min per minute requirement. The process design can be developed using piping isometric diagrams, standard pressure drop formulas for pipes and fittings, H-Q curves for the two salt pumps, and equations listing valve discharge coefficients as functions of the valve positions.

A static mixer would be added only if the CFD / FEA analyses showed that radial thermal stratifications reached values which produced unacceptable thermal stresses at the hot ends of the superheater and the reheater.

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6.7.4 Fabrication Techniques

A common joining technique for the tube-to-tubesheet connection is to strength weld the ends of the tubes to the face of the tubesheet, and then plastically deform the tubes into the tubesheets. However, there are alternate joining techniques that use welded connections.

Header / Coil

In conventional heat exchangers, the tubesheet must withstand the pressure differential between the shell- and the tube-side fluids. If the differential pressure is high, then the tubesheet must be thick (150 to 200 mm), as a flat plate is structurally inefficient when subjected to bending loads. However, thick metal sections are generally undesirable in equipment which must thermally cycle each day. The problems with transient stresses are compounded if thick metal sections are joined to thin metal sections.

A more efficient structure for withstanding a large differential pressure is a pipe section, particularly if the high pressure fluid is on the inside of the pipe. A section of pipe can act as a tubesheet, using the following arrangement:

- A series of holes are drilled along the length, and around the circumference, of the pipe
- Short, tapered nozzles are welded to the pipe at each hole
- The ends of the heat exchanger tubes are welded to the ends of the short nozzles.

The fabrication process is illustrated in Figure 6-9.

This fabrication approach offers three important advantages for equipment in cyclic service:

1. For a given differential pressure, the thickness of a pipe wall will be much less than the thickness of a flat tubesheet. Reducing the thickness of metal sections is generally beneficial in terms of controlling transient stresses.
2. All of the tube-to-tubesheet connections are welded. Leaks are much less likely to develop in a welded connection than in a friction connection.
3. The nozzles between the pipe and the tubes are tapered, which provides a transition in metal thickness. A tapered transition, rather than an abrupt transition as occurs between a flat tubesheet and a tube, reduces the transient thermal stresses.

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Figure 6-9 Example of Header / Coil Heat Exchanger Fabrication ⁴⁴

Internal Bore Welding

In an internal bore welded design, tapered nozzles are machined from the face of a flat tubesheet. The ends of the tubes are then welded to the ends of the nozzle. Due to space limitations, the nozzle-to-tube butt welds are performed by a robotic welder inside the tube. A cross section example of the tubesheet and the welded connection is shown in Figure 6-10.

This fabrication approach offers many of the benefits associated with the header / coil design.

Allowable Rates of Temperature Change

Heat exchanger designs based on strength welding the tubes to the tubesheet, and then expanding the tubes into the tubesheet, have an allowable rate of temperature change of 8 to 10 °C/min, depending on the vendor.

Informal discussions with Aalborg CSP, a supplier of header / coil designs, indicate that allowable rates of temperature change are on the order of 12 to 13 °C/min.

⁴⁴ <https://energetica-india.net/news/aalborg-csp-header-and-coil-technology-celebrates-its-10-years-leakage-free-anniversary>

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Figure 6-10 Example of an Internal Bore Welded Tube-to-Tubesheet Connection

The difference between the two allowable rates is not a large value. However, experience with commercial projects has shown that process designs have difficulties complying with a vendor limit of 10 °C/min. Relaxing the limit to 12 °C/min will not solve the problems with current process designs. However, revisions to the process design, as discussed above, in combination with higher allowable limits set by the vendor, should lead to needed improvements in steam generation system availability.

6.7.5 Material Selection in the Evaporator

In a number of commercial projects, the tube material in the evaporator is Type 347H stainless steel. This material certainly provides the strength necessary for operation at both the normal salt inlet temperature of 450 °C, and the tube-side design temperature of 510 °C. Further, the tube material matches the materials in the tubesheet and in the shell, which simplifies the welding processes. Nonetheless, the choice of materials does not have unanimous concurrence⁴⁵. For example, although the steam generator is not designed to ASME Section I requirements, Section I does not permit the use of stainless materials in evaporators due to the potential for stress corrosion cracking. Similarly, European norm EN12952-3, Water Side Boilers, does not permit the use of austenitic materials in boilers or in steam drums.

The stress corrosion mechanism is promoted by chlorides, caustic, or reduced sulfur which may be present in the feedwater. Further, stainless steel is susceptible to pitting corrosion if deposits accumulate

⁴⁵ Aalborg CSP, 'NE1 700 MW CSP Project, Material Selection for Evaporator Tubes, CT-SGS', Document 40088C-D156-00006, Revision A, December 2018

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on local portions of the surface but do not accumulate on portions adjacent to the deposits. The deposits can result from corrosion of the carbon steel components in the condensate system. The corrosion can result from poor control over the pH and the dissolved oxygen in the feedwater.

An alternate to Type 347H as the tube material is a ferritic alloy with a high chromium content, such as T91⁴⁶. This material offers acceptable corrosion resistance to salt at the normal operating conditions, and was successfully used in the first of two evaporators at the Solar Two demonstration project. Further, the material does not use nickel as an alloying elements; as such, it is largely immune to stress corrosion cracking mechanisms.

⁴⁶ Bradshaw, R. W. and Goods, S. H. (Sandia National Laboratories, Livermore, California), 'Corrosion of Alloys and Metals by Molten Nitrates', Sandia Report SAND2000-8727, August 2021

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7. Valves in Salt Service

7.1 *Valve Types*

Isolation valves in salt service, including gate, ball, angle plug, swing check, and triple offset butterfly, will provide similar isolation characteristics to service in water. Similarly, control valves in salt service, including globe and angle plug, will provide similar control characteristics to service in water. However, not all types of valves are suitable for service in salt if the operating periods are measured in months to years.

Salt is an oxidizing material, and will develop oxide layers on all types of iron and nickel alloys. If the valve uses sliding surfaces, such as a gate valve or a ball valve, then oxide layers will develop on both the sliding and the stationary faces. Further, the oxide layers can mechanically bond to one another if the valve is not moved frequently; i.e., every few days. The corrosion process results in valve binding, even to the extent that the valve operators cannot develop the forces required to overcome the friction and the bonding forces. At the Solar Two demonstration project, the pneumatic actuator on an 8 in. ball valve developed forces high enough to plastically deform the valve stem, yet the valve remained in the open position.

In general, the only types of valves suitable for salt service are the ones where the plug moves perpendicular to the seat. The oxide layers still develop on the sealing surfaces. However, the oxide binding forces are lower if the surfaces move perpendicular to, rather than parallel to, one another. As such, the only suitable valve types are globe, angle plug, and triple offset butterfly. Nonetheless, it can be noted that some of the triple offset butterfly isolation valves at a commercial project developed internal leaks over a period of about 3 years. The source of the leakage was not fully identified, but it may be deformations of the valve bodies that are higher than expected due to the transmission of piping loads that are higher than expected. The solution for the commercial project was to replace the triple offset butterfly valves with valves which have a higher body rigidity; i.e., globe valves.

7.2 *Stem Sealing*

A common stem sealing material is graphite. The material is soft, which allows it to readily conform to the shape of the stem. Also, in the absence of an oxidizing environment, it is thermally stable. However, in salt service, the salt oxidizes the graphite to CO₂, and the packing eventually disappears.

At the Solar Two project, the recommended stem packing consisted of alternating layers of 1) Teflon washers, and 2) Inconel wire braids impregnated with graphite. Over time (weeks to months), the graphite in the braid oxidized, and the stem packings required periodic replacement of the Inconel braids. In general, this combination of packing materials was judged as marginally adequate. In subsequent commercial projects, other stem packing materials have been used, including various

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fluoropolymers, mica, and vermiculite. In some cases, the packing performed as intended, and in other cases, the packing developed leaks almost immediately. In general, a robust packing material, with a longevity measured in years, has yet to be identified.

To this end, the only stem seal which has been shown to be effective in preventing a leak is a bellows seal. This, in turn, implies that the preferred valve types are limited to those which can use a bellows seal; i.e., valves with translating stems, such as globe and angle plug valves.

As might be expected, this approach has met with resistance in commercial projects:

- A bellows stem seal is much more expensive than a packing stem seal. Budgetary estimates from a study conducted for DOE indicate that adding a bellows stem seal can triple the cost of a valve
- For isolation service, large globe valves are more expensive than triple offset butterfly valves
- The bellows may, under specific sets of circumstances, rupture. These circumstances include moving the valve with frozen salt in the bellows, or exposing the bellows to high transient pressures generated by a leak in a heat exchanger.

Using bellows seals on valves will increase the capital cost of the project. However, the cost penalty can be reduced by developing process designs that minimize the use of valves. One example is the piping arrangement for storage tank recirculation heaters, discussed in Section 6.7.1. The cost penalty is also likely to be offset by an increase in plant availability. Leakage past the valve stem seals exposes the heat trace cables on the valve body and on the adjacent piping to salt. The cables operate at temperatures which are high enough ($> 650\text{ }^{\circ}\text{C}$) to decompose the salt. Several of the decomposition products are various oxides, which aggressively corrode the outer metal covering on the heat trace cables. Corrosion lifetimes are often measured in weeks. Once the covering corrodes, the internal heating cables are exposed to moisture, and can fail in a matter of days. Failed cables lead to immovable valves and frozen salt piping, which can necessitate forced outages, lasting hours to days. A project economic analysis will often show that the marginal improvement in plant availability and revenue, due to a properly functioning heat trace system, justifies the marginal expense for bellows stem seals.

In a thermal storage system in a commercial parabolic trough project, the salt control valves used bellows stem seals. The valve vendor supplied an engineered insulation enclosure for the valve body and the bellows region. The enclosure included the requisite heat trace cables and control thermocouples. The valve, bellows, and insulated enclosure are operating at expected, with no stem sealing failures due to frozen salt in the bellows.

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7.3 *Triple Offset Butterfly Valves*

In general, and particularly for large valves (> 18 in.), triple offset butterfly valves are less expensive than globe valves. Since the valve provides good sealing characteristics, there is a motivation to use the valves wherever possible. Nonetheless, triple offset valves carry potential liabilities, as follows:

- Conventional stem packings, rather than bellows seals, must be used
- In large sizes, the valve bodies are not as rigid as those for globe valves. There have been instances in which large forces needed to be applied to piping to align the piping with the valve bodies. Once the welding was complete, the alignment forces were removed, which can result in a deformation of the triple offset valve body and a reduction in sealing effectiveness.

7.4 *Isolation Valves*

In the process design, careful consideration should be given to selection of globe, angle plug, and triple offset valves for isolation service. The detrimental effects on plant availability associated with leakage past valve stems should not be discounted. The selection of a valve with a bellows seal will likely increase the capital cost of the project, yet decrease the levelized energy cost.

7.5 *Control Valves*

For valve sizes of 20 in. and below, the preferred valve for control service is a globe valve with a bellows seal. For pipe sizes of 24 in. and above, there is a temptation to use a triple offset valve for control service due to cost considerations. However, the Cv versus percent open curves for triple offset valves show strong non-linear behavior, which can lead to difficulties in process control. If a control valve larger than 20 in. is required, the preferred option would be two 50-percent capacity globe valves with bellows seals.

7.6 *Alternates to Globe and Triple Offset Butterfly Valves*

The discussion above concentrates on the relative benefits and liabilities of globe valves and triple offset butterfly valves. However, alternate valve designs may also be available from commercial suppliers. Early in the project, discussions can be held between the project and potential valve suppliers to determine if alternate valve concepts could offer a suitable combination of structural rigidity, pressure drop, control linearity, hermetic stem sealing, and cost.

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7.7 *Pressure Relief Valve in Series with a Rupture Disc*

Some form of salt pressure relief system is required on the heat exchangers in the steam generator, and on the receiver inlet vessel in a central receiver project. Candidate relief devices include relief valves and rupture discs.

7.7.1 Relief Valves

The relief valves are conventional commercial designs, applicable to a wide range of process conditions.

The valve body, and the piping downstream of the valve, will require heat tracing. Due to the different geometries, separate heat trace zones will be required for the valve body and the discharge piping. The heat tracing must be active at all times.

The principal advantage is the ability of the valve to automatically reseal should the fluid pressure temporarily rise above the set point pressure. The principal liabilities include the following:

- A Code requirement to periodically remove the valve for recalibration
- The potential for seepage through the valve due to 1) corrosion between the valve and the seat, or 2) thermal relaxation of the valve spring.

7.7.2 Rupture Discs

The rupture discs are also conventional commercial designs, applicable to a wide range of process conditions.

The rupture disc, and the piping downstream of the disc, will require heat tracing. Separate heat trace zones may be required for the disc and the discharge piping, depending on the size of the disc and its holder. The heat tracing must be active at all times.

The principal advantages are 1) an immunity from seepage, and 2) no need for periodic removal for recalibration. The principal liabilities include the following:

- The disc, by definition, operates at stresses very close to the yield value. As such, the disc can accumulate a combination of creep and fatigue damage due to the daily plant pressure and temperature cycles. At some point, the disc can develop a crack, either a small one or a large one
- When the disc develops a crack, either due to a pressure above the set point or due to creep plus fatigue damage, the disc will rupture. This will require a forced plant outage to replace the disc.

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7.7.3 Passive Pressure Relief Versus Active Pressure Relief

A nuclear project developer is proposing a design in which a nuclear reactor, cooled by sodium, supplies energy to a salt system. The salt system consist of heat transport piping, a thermal storage system, and a steam generation system. Energy is transferred from the reactor loop to the salt system through a sodium-to-salt heat exchanger. One of the goals of the design is to separate nuclear components, and the associated licensing requirements, from the portions of the plant associated with heat transport, thermal storage, and electric power production.

A group of cold salt pumps supply cold salt to the sodium-to-salt heat exchanger. Depending on the final equipment selection, the shutoff head of the pumps may exceed the maximum allowable working pressure of the heat exchanger. To protect the heat exchanger, a pressure relief device, such as a rupture disc or a relief valve, can be used. Ideally the pressure relief device would be passive (i.e., no moving parts) to help satisfy the nuclear regulatory requirements.

In theory, a rupture disc could be viewed as a passive device. However, the piping to and from the rupture disc must be maintained at temperatures above the salt freezing point at all times. The temperatures are maintained by the heat trace system. The heat trace system is an active system, as it relies on 1) thermocouples and logic controllers to monitor the zone temperatures, 2) electric contactors to energize and de-energize the heat trace cables, and 3) transformers and switchgear to supply electric power to the contactors. As such, the regulatory agencies are unlikely to view the rupture disc as a passive device.

7.7.4 Relief Valve in Series with a Rupture Disc

As an alternate to a relief valve or a rupture disc, a relief valve can be located in series with a rupture disc, as shown in Figure 7-1.

A monitoring device registers the pressure in the space downstream of the rupture disc, and indicates whether the disc is intact or not.

Depending on the geometry of the series installation, separate heat trace zones may be required for the disc, the monitoring device, and the relief valve. The heat tracing must be active at all times.

Locating the valve downstream of the disc resolves some of the liabilities noted above, as follows:

- The seat and the plug in the relief valve are isolated from the corrosive effects of salt
- If the rupture disc breaks, then the relief valve continues to operate in the normal manner, and the plant does not incur a forced outage.

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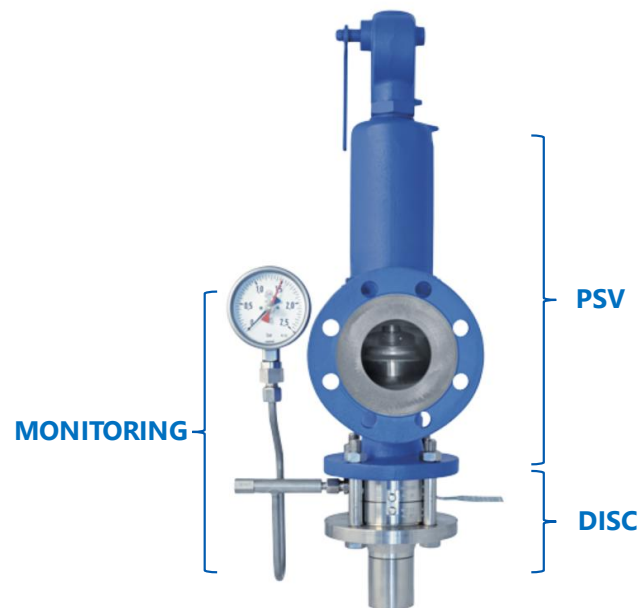


Figure 7-1 Pressure Relief Valve in Series with a Rupture Disc

Nonetheless, the theoretical benefits of the series installation may not be fully realized.

- Given the odd geometry of the detection instrument, this location may require a separate heat trace zone; i.e., heat trace cable, control thermocouples, and thermal insulation. However, the detection instrument, and the associated heat tracing, is not critical to the operation of the plant. As such, this is one location in the plant that 1) might easily be ignored by the maintenance staff, and 2) should it stop working, no one knows that it stopped working.
- If the heat tracing on the detection instrument stops working, or if the insulation on the detection instrument becomes damaged, then the detection instrument can act like a thermal fin. This has to potential to freeze the salt in the stagnant zone between the rupture disc and the relief valve, which would prevent the proper functioning of the relief valve. Salt equipment, with stagnant conditions and small physical dimensions, is almost a guarantee of salt solidification.

Providing a rupture disc in series with a pressure relief valve will eliminate problems with salt weeping through a relief valve. However, the benefits are likely to be temporary. At some point, problems will develop with the rupture disc, the detection instrument, or both, and the theoretical benefits of investing in the rupture disc will be lost.

The design with the highest reliability is likely to be a pressure relief valve. When the valve starts to develop an internal leak, the maintenance staff will have an additional task regarding salt collection from the discharge piping downstream of the valve. However, the additional task will be temporary. The

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leakage will persist only until the plug and the seat can be resurfaced when the valve is removed for periodic recalibration.

7.7.5 Energy Involved in the Flow Downstream of a Relief Device

The piping downstream of a pressure relief device must be directed to a safe location. Depending on the size of the relief device, the safe location could be a dedicated concrete pit or one of the main storage tanks. In either case, the relief device is discharging to ambient pressure, and all of the pressure head upstream of the device is converted to a velocity head downstream of the device.

At a proposed commercial project, a pressure relief device, located near the discharge of a salt pump, is required to protect a downstream heat exchanger. The relief pressure in this example is 16 bar. With a downstream pressure of 1 bar, the pressure differential of 1,500,000 Pa is converted to kinetic energy with a nominal velocity of 40 m/sec.

Since the relief device is located near the salt pump, the relief device is likely to discharge back to the salt tank. The mass flow rate through the relief device will depend on the H-Q curve of the pump. However, the kinetic energy in the flow returning to the storage tank will be a large value. The kinetic energy is eventually converted to heat in the storage tank, but the amount of energy to be dissipated is such that the conversion device must be robust and not a design afterthought.

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8. Piping in Salt Service

In parabolic trough projects using organic heat transfer fluids, both the cold salt and the hot salt piping will be carbon steel, with a typical specification of ASTM A106, Grade B or C.

In parabolic trough plants using inorganic heat transfer fluids, and in central receiver projects, the hot salt piping will be a 300-series stainless steel. The principal candidate materials include Type 304L, 316L, 304H, 316H, and 347H. The 'L' signifies a low carbon alloy, with a maximum carbon content of 0.02 percent. The 'H' signifies a high carbon alloy, with a minimum carbon content of 0.04 percent. Adding carbon to the alloy disrupts the lattice structure, which improves the resistance to creep.

The design temperature in both types of plants is above 538 °C. As such, if the allowable stress values for the piping are to be obtained from Subpart 1 in Section II D of the Code, then the 'H' grades of stainless steel will be required. For example, in Subpart 1, Type 304L is not permitted for equipment designed to Section III or to Section VIII Division 1 at temperatures above 427 °C due to the potential for dimensional changes due to permanent deformations. In contrast, ASME B31.3 lists allowable stresses for the 'L' grades of stainless steel piping for temperatures up to 815 °C.

8.1 *Type 304H and Type 316H*

At temperatures in the range of 500 to 600 °C, the carbon in Type 304H and Type 316H reacts with chromium to form chromium carbide. The carbide is preferentially formed at the grain boundaries, as there is space in the matrix to do so. This results in a reduction in the corrosion resistance of the alloy at the grain boundaries. If the alloy is exposed to a combination of chlorides (present in the salt), tensile stresses, and liquid water, then cracks can start to develop between the grain boundaries. Once the cracks start to develop, the process cannot be stopped, even if the liquid water is removed. This type of corrosion is known as intergranular stress corrosion cracking.

With the plant in normal service, the metal temperatures never fall far enough for the pipe to be exposed to liquid water. This, in and of itself, is enough to prevent intergranular stress corrosion cracking. However, the pipes can be exposed to liquid water if, during an outage, the equipment cools to ambient conditions, and then 1) it rains, or 2) nighttime temperatures fall below the dew point. Under these conditions, intergranular stress corrosion cracking can develop.

8.2 *Type 304L and Type 316L*

One method for reducing the potential for intergranular stress corrosion cracking is to use an 'L' grade of stainless steel. With a low carbon content, the potential to form chromium carbide is reduced. This, in turn, limits the depletion of chromium at the grain boundaries, which preserves the resistance to

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corrosion at the grain boundaries. However, the low carbon content reduces the resistance to creep, which can lead to dimensional changes due to permanent strains.

The allowable stresses for Types 304 L and 316L, at a temperature of 565 °C, are shown in Table 8-1. Also shown for reference are allowable stresses for Types 304H, 316H, and 347H.

Table 8-1 Allowable Stresses (MPa) for 300-Series Stainless Steel Piping at 565 °C

Type	Section II D, Subpart 1	B31.3
304L	Not permitted	43.4
316L	Not permitted	74.5
304H	85.5 ¹ , 69.6 ²	84.1
316H	104.1 ¹ , 77.2 ²	100.0
347H	111.7 ¹ , 92.4 ²	117.9

Notes:

1. Use of these stresses may result in dimensional changes due to permanent strain.
2. Lower allowable stress to reduce the potential for dimensional changes.

The principal penalty to the use of an L-grade material for piping is a lower allowable stress, which translates to thicker wall sections. Since piping is typically priced on \$/kg commodity basis, and since the unit prices for Types 304, 316, and 347 all fall within a similar range, selecting an L-grade material increases the cost of the piping by factors in the range of 20 percent to 150 percent.

It can be noted that there is some experience with the use of 304L for hot salt piping. At the Solar Two central receiver project, the design of the downcomer was developed using the properties of an H-grade stainless steel. However, the downcomer was fabricated using 304L, and the mismatch was discovered only after the downcomer had been installed. Rather than replace the downcomer with an H-grade material, the pipe stress analysis was repeated, but with iterations on the supports and the anchors until the stresses reached allowable values. During the (limited) 3-year life of the demonstration project, the downcomer operated as intended.

8.3 *Type 347H*

To avoid potential problems with intergranular stress corrosion cracking, most commercial projects have adopted Type 347H as the hot salt piping material. The alloy is essentially Type 304H with the addition of niobium. The niobium bonds with the carbon in the matrix to form niobium carbide. This reduces

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the quantity of free carbon available, which, in turn, reduces the potential for the formation of chromium carbide. If the chromium carbide concentration is reduced, then the potential for intergranular stress corrosion cracking is also reduced.

8.3.1 Knife Line Attack

Nonetheless, the alloy is not without its own set of problems. During welding, the metal temperatures are high enough to dissolve the niobium carbide. This makes available free carbon in the matrix. As the weld cools, the metal temperatures initially favor the reformation of niobium carbide. However, the weld typically cools at a rate that does not allow the equilibrium concentration of niobium carbide to form. Free carbon remains available in the matrix, which then allows chromium carbide to form as the weld continues to cool.

The welding process often results in 5 metal chemistry zones. Zones 1 and 5 are the original parent metal, to the left and to the right of the heat affected zone. Zone 3 is the center of the weld, where the concentration of niobium carbide is relatively high. Zones 2 and 4, which are in the heat affected zone, tend to have low concentrations of niobium carbide, but relatively high concentrations of chromium carbide. Zones 2 and 4 can then be susceptible to intergranular stress corrosion cracking in a phenomenon known as knife line attack. Knife line attack has been observed in Type 347H pipe fittings at the 10 MWe Solar Two demonstration project, and in Type 321H receiver tubes at a parabolic trough test facility in Spain.

8.3.2 Post Weld Heat Treatment of Type 347H

To reduce the potential for knife line attack, the weld can undergo a post weld heat treatment. The heat treatment involves the following steps:

- The residual weld stresses are partially relieved by raising the metal temperature to a nominal value of 650 °C. The goal is to reduce the potential for hot reheat cracking. Hot reheat cracking is a phenomenon in which the interior of the grain becomes strengthened by the precipitation of carbides during welding. Relaxation of residual stresses then occurs in the form of creep deformation at the grain boundaries, which can lead to cracking along the boundaries
- After a hold period for stress relaxation, the welds are solution annealed. The metal temperatures are raised high enough (~1,150 °C) to dissolve both the chromium carbide and the niobium carbide
- The metal temperature are held for a period which is long enough (~1 hour) for the equilibrium concentration of niobium carbide to form. This process reduces, but does not eliminate, all of the free carbon; i.e., there is an equilibrium concentration of niobium carbide, chromium carbide, and free carbon.

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- To reduce the potential for chromium carbide formation with the remaining free carbon, the weld zone is cooled rapidly through the temperature range which preferentially forms chromium carbide.

As might be expected, the heat treatment process is time consuming and not without expense. Further, the process is conducted outdoors, and if the time-temperature requirements cannot be accurately met, then it is possible to develop cracks at the weld or to increase, rather than decrease, the concentrations of chromium carbides.

8.4 Project Decisions

The candidate piping materials, in order of probable commercial preference, are as follows:

- Type 347H: The material offers 1) the highest allowable stress among the 300-series stainless steels, which results in the lowest piping mass and installed costs, and 2) the addition of niobium to the alloy reduces the potential for intergranular stress corrosion cracking that can occur in the other H-grade stainless steels. The principal liability is a (relatively low) risk of knife line attack should the piping be exposed to liquid water.
- Type 304L / 316L: Due to the low carbon content, the material is largely immune to intergranular stress corrosion cracking, and due to the absence of niobium, the material is largely immune to knife line attack. The principal liabilities are relatively low allowable stresses, which result in an increase in the pipe mass and the installed cost.
- Type 304H / 316H: The allowable stresses for 304H / 316H are higher than the values for the L-grade materials, but somewhat less than the values for Type 347H. As such, the mass of the piping and the installed cost should fall between these cases. The principal liability is a susceptibility to intergranular stress corrosion cracking should the pipe be exposed to liquid water during a maintenance outage.

Type 347H may be the preferred piping material, for the reasons noted above. Nonetheless, the project must make a decision as to whether, or not, to perform a post weld heat treatment of the piping.

There is no mandatory requirement to perform a heat treatment, as the Code does not require the heat treatment of stainless materials. Further, anecdotal evidence from commercial projects indicates that the reliability and the availability of welds that receive a heat treatment are essentially the same as the reliability and the availability of welds that do not receive a heat treatment. Nonetheless, there may be a benefit to performing a heat treatment, as follows:

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- Hot reheat cracking can occur in the heat affected zones of weld when reheated in the temperature range from 500 to 700 °C⁴⁷. At this temperature, residual stresses due to welding relax by creep. The heat affected zone may not be able to accommodate this strain without cracking if the microstructure had been sufficiently altered during welding. Pipe temperatures above 565 °C are not expected during normal plant operation. However, metal temperatures in the range of 600 to 700+ °C can be expected if a heat trace circuit has a lengthy duty cycle of 100 percent due, for example, to a contactor sticking in the closed position. A post weld heat treatment might allow the welds in this zone to avoid hot reheat cracking associated with an infrequent, but not unexpected, failure in the heat trace system.
- During the life of the project, the probability that the hot salt piping will be cooled to ambient temperature, and then opened to the ambient, during a maintenance outage is perhaps one-half to two-thirds. The exposure could result from repairs to a valve, inspection of a heat exchanger, or replacement of a pressure transmitter. During this time, liquid water could come into contact with the inside of the pipe due to rain, water leakage from a steam generator heat exchanger, or from ambient temperatures below the dew point. In addition, salt leakage from valve stem seals or from gaskets on pressure transmitters can expose both the insulation and the exterior of the pipe to salt. If the insulation is removed for replacement, liquid water can come into contact with the outside of the pipe due to rain or ambient temperatures below the dew point. In each case, a post weld treatment of the welds will reduce the potential for knife line attack at the edges of the welds.

In the final analysis, a post weld heat treatment can reduce the risk of hot reheat cracking and intergranular stress corrosion cracking. Nonetheless, in the absence of a post weld heat treatment, the potential for hot reheat cracking and intergranular stress corrosion cracking is generally regarded as low to moderate. The project will need to conduct a cost / benefit analysis to determine if the costs and the risks of post weld heat treatments justify the benefits of the heat treatments.

8.5 *Pipe Maintenance*

Once the plant enters commercial service, the salt piping, valves, and instruments will have been exposed to salt. When this equipment requires maintenance, the equipment will be cooled to ambient temperature. As such, the equipment can be exposed to liquid water if the metal temperature falls below the dew point, if it rains, or if the equipment is cleaned with water.

Carbon steel, in contact with nitrate compounds and liquid water, can develop stress corrosion cracking, as discussed above in Section 4.2. Stainless steel equipment, in contact with the chlorides in the salt, can develop intergranular stress corrosion cracking, as discussed in Section 5.16.1.

⁴⁷ Auzoux, Q., et al, (CEA Saclay DEN/DMN/SRMA Yvette, France), "Reheat cracking in austenitic stainless steel", 2001, <https://www.osti.gov/etdweb/servlets/purl/20401493>

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To reduce the potential for atmospheric condensation, the temperature of the equipment can be maintained above the dew point by means of portable electric or propane-fired air heaters. Temporary tarps or tents can limit the exposure of the equipment to rain. Further, water in any form, including distilled water, must not be used to remove residual salt from the equipment. If the equipment must be cleaned prior to further work, such as welding, then only mechanical methods should be used.

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9. Electric Heat Tracing

As discussed in Volumes 1 and 2, the electric heat tracing uses mineral insulated cables, electronic contactors, and pulse width control.

9.1 *Design Criteria*

Heat tracing is an expensive item, and there is an economic incentive to install as little heating capacity as possible.

The absolute minimum capacity would be one which maintains the equipment at the salt melting temperature (240 °C) at the design minimum ambient temperature and at the design plant operating wind speed. However, any reduction in the output from the heat tracing, or any degradation in the insulation, will result in equipment temperatures below the freezing point of the salt.

Some commercial projects have adopted a piping design criteria based, in part, on maintaining a metal temperature of 275 °C at the minimum design ambient temperature. This criterion is, in effect, an infinite preheat period. As an example, a DN300 (12 in.) line with 125 mm (5 in.) of calcium silicate of insulation requires a heat input of 185 W/m to maintain a temperature of 275 °C at an ambient temperature of 10 °C. However, in a commercial project, the pipe insulation is subject to damage due, for example, to maintenance personnel walking on the pipe. If the insulation is compressed by 25 mm (1 in.), then the local thermal resistance is reduced, and the steady state temperature which can be maintained by the heat trace drops to 232 °C. Similarly, the insulation can develop gaps due to daily thermal expansion and contraction cycles. If the local effective thermal conductivity of the insulation increases by 20 percent, then the steady state temperature which can be maintained by the heat tracing falls to 230 °C. In either situation, if stagnant salt is present in the line, then local freezing can be expected. Further, the heat tracing can no longer raise the pipe metal temperature above the freezing point, and the line will remain plugged until repairs are made to the insulation. When the plant is new, the pipe insulation will provide the required thermal resistance, and the heat trace system will be judged to be adequate. However, as the plant ages, the pipe insulation will degrade, and availability problems with frozen salt lines will start to appear.

A more robust design criterion is a preheat period of, for example, 12 hours. Continuing with this example, preheating the pipe to 275 °C in 12 hours requires a heat input of 354 W/m, or nominally double the heat input if the preheat period is infinite. With a heat input of 354 W/m, the steady state temperature of the pipe increases from 275 °C, as above, to a new value of 516 °C. Further, if the insulation is compressed by 25 mm, then the steady state pipe temperatures can, in principle, be raised as high as 435 °C. Similarly, if the effective thermal conductivity of the insulation increases by 20 percent due to gaps in the insulation, then the pipe temperatures can be raised to a theoretical value of 430 °C. It's unlikely that the duty cycles of the heat trace system would be set so high as to produce pipe temperatures as high as 430 °C, but the available heating capacity would be sufficient to both prevent

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freezing and to thaw a line should freezing occur. As such, the heat trace system will be adequate both when the plant is new and after the plant has accumulated decades of commercial use.

For a specific project, the preheat time would be selected based on a range of parameters, including the definitions of the plant operating states, the heat trace zone boundaries, the defined transition times between operating states, the importance of a particular zone in terms of plant availability, the installed cost of the heat trace system, and the difficulties in accessing the heat trace zones for maintenance of both the cables and the pipe insulation.

In general, the optimum heat trace preheat period has yet to be identified. A preheat period of 8 hours provides a robust system, but likely results in a system which is more expensive than can be justified. However, the preheat period should not be more than 24 hours, as the margins against insulation degradation are likely insufficient if a commercial project is to reach its availability targets over a time span of decades.

9.2 Control Software Location

Essentially all heat trace suppliers provide a standalone, and often proprietary, programmable logic controller to control the duty cycles of the heat trace cables. The alternate control approach is to use the plant distributed control system for this function. In most commercial projects, the former approach is selected over the latter in terms of convenience, purchase price, and programming price. Nonetheless, the latter is generally preferred over the former, for the following reasons:

- The selection of which heat trace circuits, and the selection of the duty cycles for each circuit, can be readily adjusted based on the plant operating state. This, in turn, reduces the annual demand for auxiliary electric energy, and makes for more stable operating temperatures in each zone
- The control system data historian can be used for the analysis of the heat trace system and the diagnosis of temperature control problems
- Alarms are transmitted directly to the control system operator screens. This avoids the need for an operator to move from an operator station to the heat trace vendor's console to analyze an alarm. This, in turn, makes it more likely that prompt responses will be made to each alarm.

At the Solar Two demonstration project, the original heat trace system installation used a proprietary programmable logic controller supplied by the system vendor. However, modifications to the heat trace components were made over the course of the project, and the control responsibilities were shifted from the vendor equipment to the distributed control system. This arrangement incurred expenses for adding input / output points to the control system. However, the benefits of improved convenience for the

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operators, improved accuracy in maintaining the zone temperature set points, faster diagnoses of unusual system characteristics (such as valve leakage), and a reduction in auxiliary energy demand more than offset the marginal expense to the control system.

In a commercial project, incorporating the heat trace control into the distributed control system will involve a large number of input and output signals. This, in turn, will increase the cost of the distributed control system. To reduce the change in price, a hybrid control approach may be possible, as follows:

The standard controller provided by the vendor receives inputs from the circuit thermocouples, compares the zone temperatures with the set point temperatures, and sends output signals to the contactors supplying electric power to the heat trace cables

The distributed control system contains the logic for changes to the set point temperatures, and the activation / deactivation of zones based on the plant operating mode

As mentioned in volume 2, all that can be achieved still maintaining vendor PLCs for control of circuits and temperature measurements, and having then a fiber optics communication link to DCS which can be used to send all signals in the PLC. The logics on set-point changes and zones activation/deactivation can reside in the DCS, and also the full historian with everything available in the PLC. The PLC would only have the logics for start/stop when temperature setpoint is reached, hysteresis, and errors management.

9.3 Cable Redundancy

Inquires with heat trace cable suppliers on the topic of cable failure rates indicate that the cables should last the life of the project. However, commercial experience has shown that cable lifetimes are often measured in years, rather than decades. The relatively short lifetimes can be due to the following effects:

- Flexing of the cables produced by the daily thermal expansions and contractions of the piping
- High operating temperatures if the circuit controls mistakenly operate the cables at a duty cycle of 100 percent for extended periods
- High operating temperatures if the zone control thermocouple is located in an incorrect position, or if the thermal insulation around the thermocouple is damaged.

Further, the cable lifetimes can as short as days or weeks if the cables are exposed to salt from a leaking valve stem or from a leaking instrument connection. The cables, when active, can produce surface temperatures in excess of 650 °C. At these temperatures, salt in contact with the cable will quickly

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decompose, producing NO_x and various oxides. The oxides are particularly corrosive, and will corrode the common sheath materials of stainless steel and Inconel. After this happens, the heating elements are exposed to water vapor in the air, and the elements soon fail.

In many commercial projects, the design criteria for the heat trace system often include a provision for only 1 installed redundant cable, regardless of the pipe size and the number of active cables needed to satisfy the preheat requirements. However, if the lifetimes of the cables are measured in, under ideal circumstances, years, then most of the heat trace zones will be forced to switch to the redundant cable in the first years of plant operation. Further, the zones will be unable to meet the heat trace design requirements starting in perhaps year 10 of commercial service. If a salt leak occurs, then the cable lifetimes can be two to three orders of magnitude shorter.

If a zone has only 1 redundant cable, then the loss of 2 cables on a small line (< 10 in.) means that the heat trace can no longer prevent the salt from freezing in a stagnant line. The loss of 2 cables on a large line means that the zone can no longer satisfy the required preheat time from ambient. In both cases, the only solution is to remove the insulation from the lines, replace the failed cables, and reinstall the insulation. Depending on the location of the zone, and the length of the zone, the repair process could require a forced outage lasting between 1 day to 2 weeks. For a nominal 125 MWe project, this could incur a loss in revenue of \$150,000 to \$2,000,000.

For a specific project, an economic analysis should be conducted to determine the optimum number of installed redundant cables. The analysis should include the following considerations:

- Experience with cable failure rates on commercial projects, rather than projected cable lifetimes offered by the heat trace supplier
- Susceptibility to a salt leak, including the number of valves in the zone, the type of stem seal on the valves (bellows or packing), and the number of instruments in the zone using flanged connections
- Zone location, zone dimension, and access for repair. For example, the riser and downcomer piping in the tower of a central receiver project generally have poor accessibility, and the number of redundant cables is likely to be a higher number than for a short zone consisting of horizontal piping at grade.
- Effect on the plant availability of a failed cable, including longer preheat periods for an empty line, or the thawing time for a frozen pipe
- Installed cost of redundant cables.

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In general, the optimum number of installed redundant cables has yet to be identified, and the number is likely to be a function of the characteristics of the zone. Nonetheless, the following values for redundancy may, in the long run, prove be representative:

- 100 percent for zones 1) with valves using conventional stem packings, and 2) with instruments with flanged connections
- 50 percent for zones 1) with valves using bellows stem seals, and 2) without instruments.

9.4 Control Thermocouples

In many commercial projects, 2 thermocouples are installed in each zone for temperature monitoring and control. In some cases, one of the thermocouples is an installed spare. As such, the temperature measured at the thermocouple is taken to be the temperature along the entire length, and around the entire circumference, of the zone.

Clearly, this is a simplified version of the actual conditions. Depending on the pattern of the cables installed in the zone, and depending on the location of the control thermocouple, local zone temperatures can differ from the measured zone temperature by as little as 1 °C to more than 150 °C. These deviations can be above, or below, the zone set point temperature.

Further, if the control thermocouple of Zone 1 is close to Zone 2, then the control of the heat trace circuits in Zone 1 can be determined by the temperature in Zone 2. At one commercial project, this led to a situation in which the heat trace cables, on a section of carbon steel piping, operated with a duty cycle of 100 percent. Within a few weeks, the pipe section failed. A postmortem examination estimated that metal temperatures exceeded 500 °C, which produced rapid oxidation of the steel.

In general, using 1, or in some cases 2, thermocouples for zone temperature control may not be adequate. The number of thermocouples should be based on the following:

- For long zones downstream of isolation valves, vent valves, drain valves, and pressure relief valves, multiple thermocouples are useful for identifying leakage patterns and the potential for frozen plugs
- For long zones with multiple pipe supports and anchors, multiple thermocouples are useful for identifying local areas with low pipe temperatures due to the higher heat losses at the supports and the anchors
- For short zones connected to other short zones, multiple thermocouples are useful for controlling each zone independently of the adjacent zones

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- For carbon steel lines, multiple thermocouples are useful for identifying local areas with metal temperatures above the set point value. This, in turn, reduces the potential for overheating and damage due to corrosion.

Adding thermocouples, including the wiring back to the DCS, is not without expense. However, the ability to analyze the temperature profiles throughout a zone has proven to be useful in preventing local freezing, preventing local overheating, and in diagnosing the events leading to an outage.

Thermocouples should be added to a zone until the marginal cost of the thermocouple exceeds the marginal revenue associated with the expected increase in plant availability.

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10. Designs for Demonstration Projects

System designs for demonstration projects may have different goals than system designs for commercial projects. For example, the goal for a pumped heat storage concept may be the demonstration of the concept, while the goal of a commercial project may be, in part, an availability of 92 percent. As such, a demonstration project may sacrifice features such as redundancy in what is likely to be a program with tight budget constraints.

Malta is proposing a pump heat storage concept that uses nitrate salt as the high temperature storage media. One potential method to reduce costs is to use the cold salt pump as an emergency backup for the hot salt pump, and to use the hot salt pump as an emergency backup for the cold salt pump. In principle, this would allow 2 pumps to perform the same duty as 4 pumps (i.e., two 100-percent cold salt pumps, and two 100-percent hot salt pumps).

One variation for the design is illustrated in Figure 10-1. Salt from the cold tank flows by gravity to a cold pump sump located below grade. The level in the sump is maintained by a control valve in the line from the tank to the sump. A similar arrangement is used for the hot salt. The two sumps are connected by a jumper line, which is located above the normal liquid levels in the two sumps.

During normal operation, the project is in either charge mode, with the cold salt pump running, or is in discharge mode, with the hot salt pump running. The liquid levels in the two sumps remain below the level of the jumper line.

10.1 *Transfer of Pump Operation*

If there is a failure of the cold salt pump during a charge period, then the level in the cold sump will rise, and salt will flow from the cold sump to the hot sump through the jumper line. The hot salt pump is started, and acts as a surrogate for the cold salt pump by changing valve positions in the pump discharge lines. Naturally, sending cold salt into the hot sump will cause a decay in the sump temperature, and impose some level of thermal shock on the hot pump. The mass of salt in the hot sump, in combination of the methods selected for mixing within the hot sump, will determine the rate of temperature change for the hot pump. The rate of change in the sump temperature must be consistent with the allowable rate of change specified by the pump vendor. If the rate of change in the sump temperature is greater than the allowable rate, then the pump bowls will thermally contract relative to the shaft, which can lead to rubbing or seizure.

Similarly, if there is a failure of the hot salt pump during a discharge period, the cold salt pump can act as a surrogate for the hot salt pump. The limitations on allowable rate of temperature change for the cold pump also apply.

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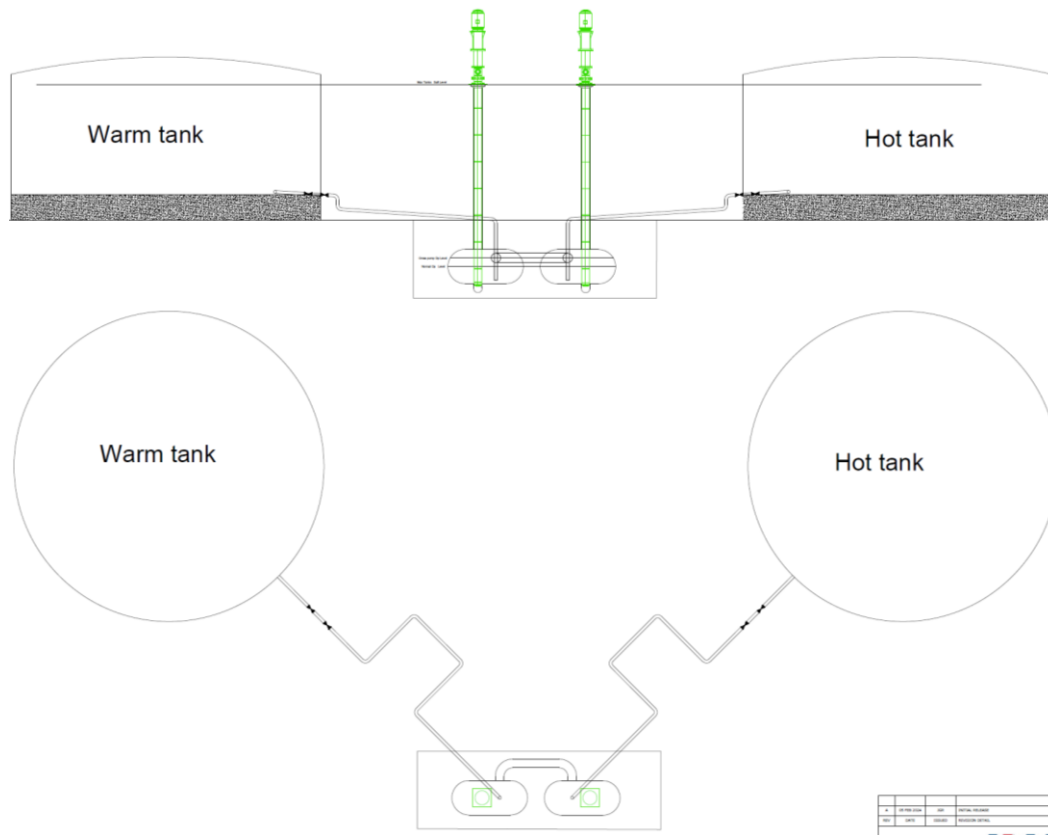


Figure 10-1 Below Grade Pump Sumps with Extended Pumps

10.2 Extended Pumps

The concept of a pump sump, located below grade and fed by gravity from the main storage tanks, was successfully demonstrated at the Solar Two demonstration project. Nonetheless, there was an instance where the level control for the hot sump did not perform as intended and the sump flooded. Salt leaving the sump saturated the mineral wool insulation on the sump, and thermally shocked the concrete in the pit, leading to spalling. In all, the damage was relatively minor. However, had the overflow not been detected and stopped, then the entire concrete pit would have filled with salt. Since there was no method to remove the salt from the pit as a liquid, the only option available would be to allow the salt to freeze, and then remove the solid salt using jackhammers. The time required for the salt to freeze was probably on the order of weeks, and the duration of the forced outage to remove the salt, and make repairs to the pump sump insulation and insulation, may have been on the order of 2 to 3 months.

In a commercial project, the dimensions of the pump sump and pit would be larger, and the time to recover from a sump overflow would certainly be measured in months. Although the probability of this situation would be regarded as low, the financial consequences of a multi-month outage would be

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significant. To avoid this financial risk, commercial projects use salt pumps supported on structures above the main storage tanks. The pumps have extended shafts and draw suction from locations just above the floor of the tank.

In the conceptual layout shown in Figure 10-1, pumps with extended shafts are shown. The throttle bushing near the top of the pump column is located above the maximum liquid level in the storage tanks. In addition, the design pressure for the pump sump is equal to the maximum static head in the storage tanks. If there is a failure in the level control system for the pump sump, the associated pump will trip, and the static pressure in the sump will rise to the static pressure in the storage tank. However, the idled pump will not overflow due to the elevated location of the throttle bushing. Nonetheless, there are some complexities with this arrangement: a structure will need to be provided to support the pumps; and a comprehensive heat tracing and insulation concept will be required to maintain the required temperatures on the pump columns (particularly the hot pump).

10.3 Conventional Pumps

If extended pumps are replaced with pumps of a more conventional length, some of the complexities associated with structural support and temperature maintenance can be reduced. A potential arrangement is illustrated in Figure 10-2.

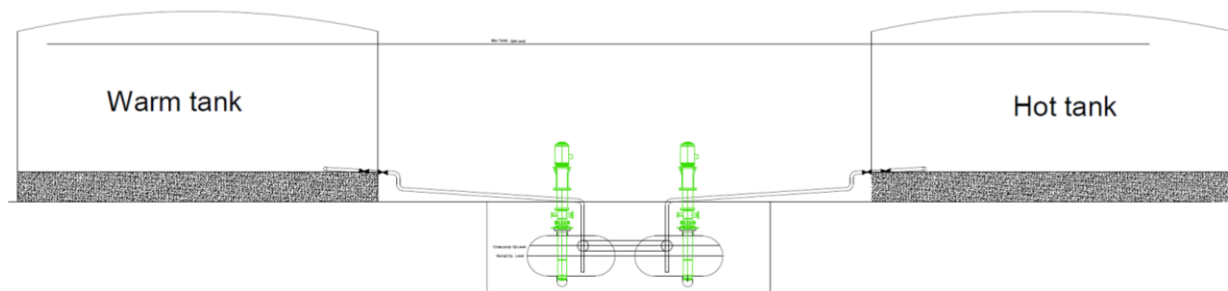


Figure 10-2 Below Grade Pump Sumps with Conventional Pumps

The shorter pumps may also offer a better ability to accommodate transient thermal stresses and a lower capital cost. However, the throttle bushings at the tops of the pumps are located below the maximum liquid level in the tanks. As such, if there is a failure in the sump level control system, it is possible to flood the sumps and flow salt into the sump pit.