SolarDynamics LLC

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

DOE Grant Number DE-EE0009810

Volume 2 - Specifications for Central Receiver 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 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

- 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

This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) Solar Energy Technologies Office (SETO) submitted under DE-FOA-0002378 Topic Area 4a: CSP: PERFORM, Technical Specifications, Control No: 2378-1549, with Award Number DE-EE0009810.

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1.5 Disclaimer

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.

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 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.

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

- 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, ranger 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

- 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.

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 NOx.

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/m^2-year. Many desert sites in the US and in North Africa have annual radiations in the range of 2,500 to 2,700 kWh/m^2-year, while the best sites in the world (Atacama desert in Chile) have annual values on the order of 3,500 kWh/m^2-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 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 ($/kWhe).

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.

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/ft^2), 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 prevents liquefaction of the soil during an earthquake, and prevents changes in the vertical heat flux from the tank foundation into the soil.

- 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/m^3
- 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 $NaNO_3$ and KNO_3 system. The nitrate salt used in commercial tower projects is a mixture of 64 mole percent $NaNO_3$ and 36 mole percent KNO_3, which is equivalent to 60 weight percent $NaNO_3$ and 40 weight percent KNO_3.

Note that the mixture is not at the eutectic composition of 50 mole percent $NaNO_3$ and 50 mole percent KNO_3. For solar applications, the fraction of $NaNO_3$ is increased to 60 weight percent to reduce the cost of the salt. Increasing the $NaNO_3$ fraction has the following effects:

- The liquidus temperature increases from 220 °C (at the eutectic composition) to a nominal 238 °C

- 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.

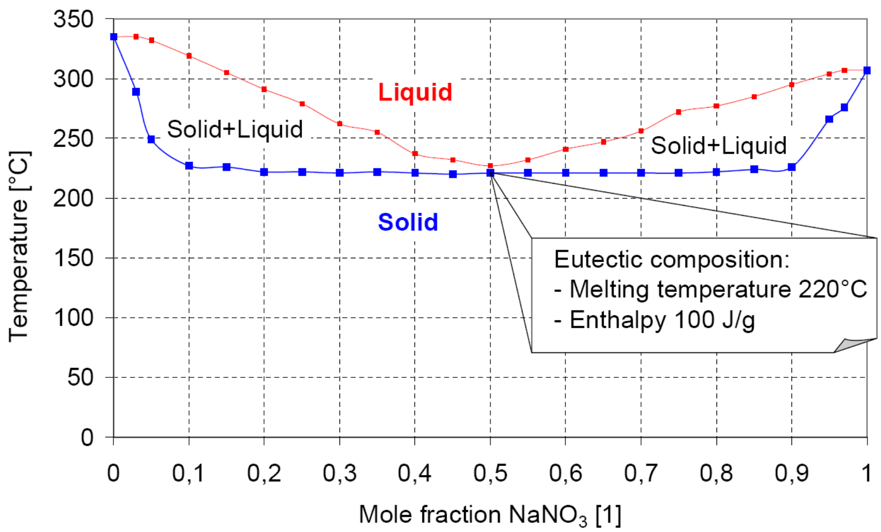

Figure 2-1 Phase Diagram of $NaNO_3$ - KNO_3 Mixtures

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.

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 ºC, 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 ºC under theoretical clear sky conditions, and the electric heat trace circuits are de-energized at the normal operation temperature set points.

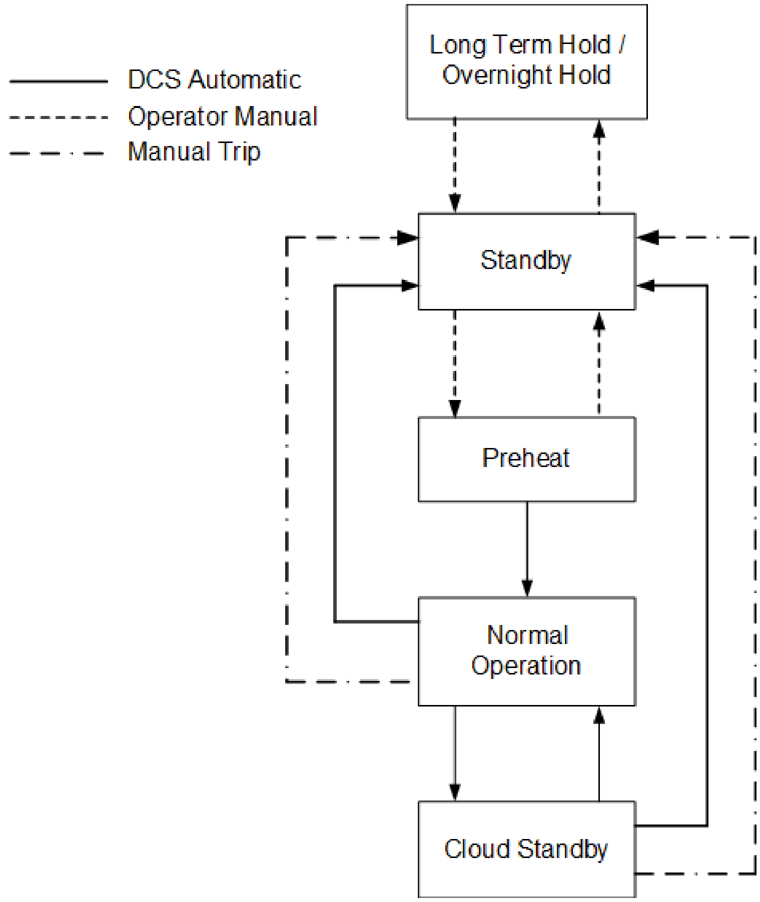

Figure 3-1 Energy Collection Section States and Transitions

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

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 ºC. 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 ºC. 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 ºC

- Normal Operation to Cloud Standby: Automatic temperature control is suspended, and the flow rate is controlled to achieve an outlet temperature of 540 ºC 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 ºC

- 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.

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) ºC 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

- 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) ºC and (project specific) bar, respectively, are established. Turbine output is set by adjusting the speed of the hot salt pump.

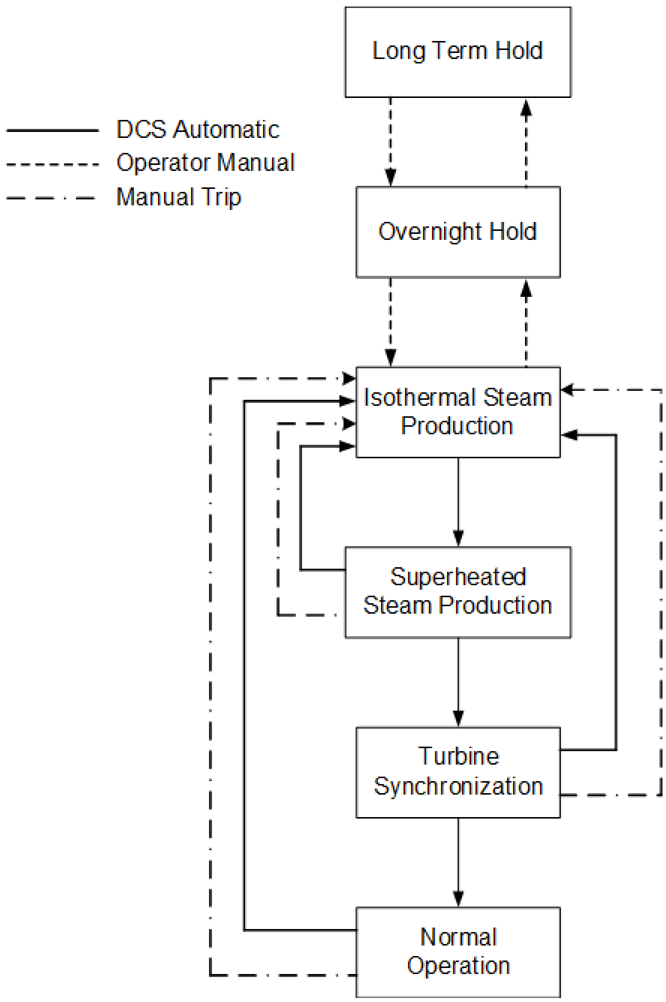

Figure 3-2 Energy Conversion Section States and Transitions

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

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 ºC 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

- 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.

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 NREL[1]. The SolarPACES CSP project database[2] 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.

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[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/

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

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

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.

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

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.

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.

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.

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

Sticking hot salt or attemperation pump minimum flow recirculation valves [1]	0.01	2	7	2
Sticking hot salt or attemperation 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.

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.

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.

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

<u>Preheating and Venting Process</u>

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.

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/m^2 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.

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.

- 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 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.

- 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.

<u>Heliostat Optical Errors</u>

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.

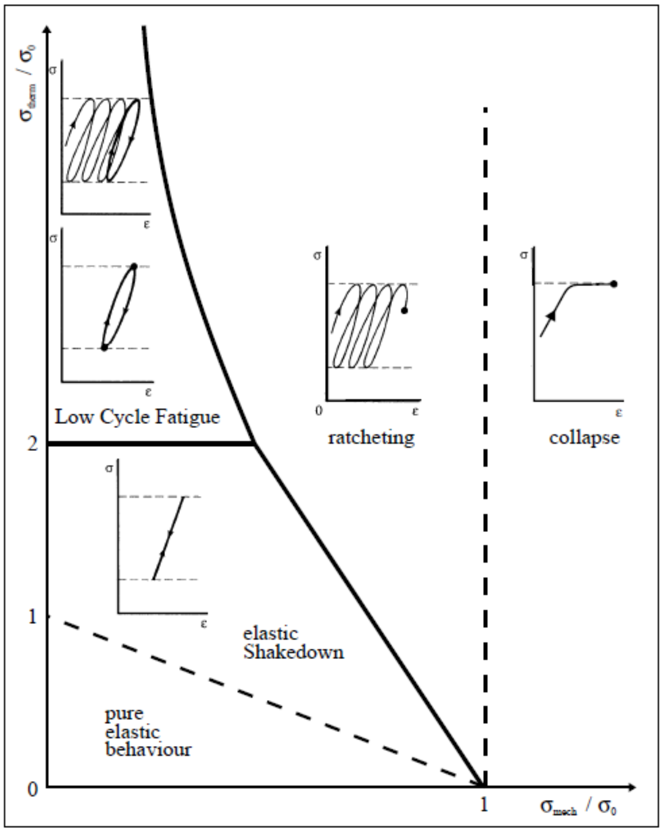

Figure 4-1 Bree Diagram of Primary Mechanical Stress and Secondary Thermal Stress

<u>Recommendations</u>

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.

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.

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 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

- Provide ad hoc training to the operators in topics such as thermal stress, low cycle fatigue, corrosion, and flow distribution.

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.

<u>Receiver Outlet Temperature</u>

The receiver outlet temperature should, in general, be as high as possible to provide a Rankine cycle efficiency that is a 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:

$$NO_3^{-} \leftrightarrow NO_2^{-} + \frac{1}{2} O_2$$

The nitrite ion, in turn, is in quasi-equilibrium with the oxide ion:

$$NO_2^{-} \leftrightarrow NO + O^{-}$$

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 a only 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 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 ($O^{=}$)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 ($/kWht) 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 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 H_2O from O_2 and NO, at a nominal temperature of 275 °C, will need to be identified.

<u>Receiver Inlet Temperature</u>

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)

- 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.

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.

<u>Inlet Vessel</u>

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.

<u>Outlet Vessel</u>

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:

- 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.

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.

<u>Receiver Panels</u>

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.

<u>Outlet Vessel</u>

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 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

<u>Tubes</u>

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.

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.

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/m^2. 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/m^2
- 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 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/m^2. 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/m^2

- 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

<u>Distribution Ring Header</u>

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:

- 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.

<u>Ring Header Supports</u>

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. The 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

- 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/ft^2). 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/m^3), 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.

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

- 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 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.

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 ∫ (Specific volume) * d(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 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.

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.

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.

Designation	Title
API	Standard 650, Welded Steel Tanks for Oil Storge
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

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 Manufactures 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.

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 weight fraction $NaNO_3$)(0.85 percent) + (0.4 weight fraction KNO_3)(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:

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

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:

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

Similarly, potassium perchlorate ($KClO_4$) 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 ($NaClO_4$) 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.

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:

$$Mg(NO_3)_2 \Rightarrow MgO + 2\ NO_2 + \tfrac{1}{2}\ O_2$$

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, N_2NO_3. 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.

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.

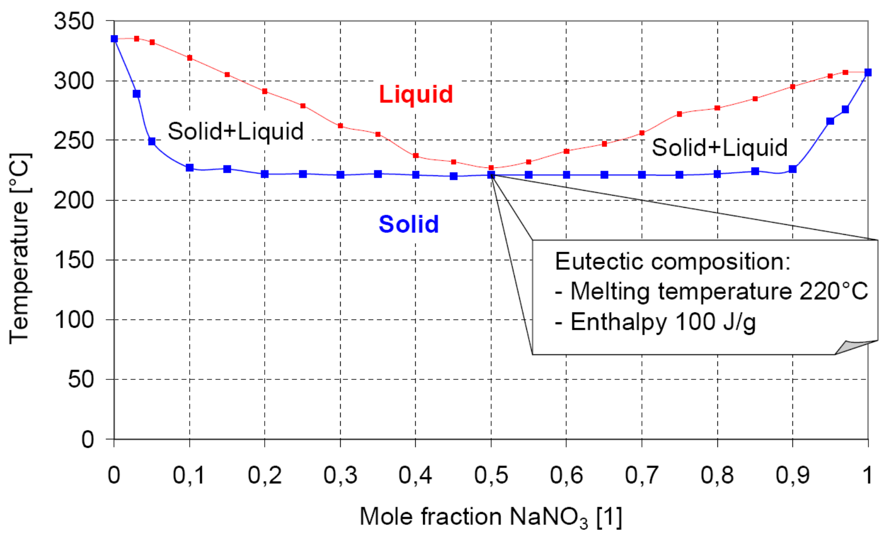

Figure 6-1 Melting Point of Binary Nitrate Salt Mixtures

Other properties, specifically the density, can be estimated by adjusting the 60/40 properties based on molar volumes, as described by Bradshaw [3].

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.

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[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

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:

$$Mg(NO_3)_2 \rightarrow MgO + 2\ NO_2 + \tfrac{1}{2}\ O_2$$

The magnesium oxide precipitates as a solid, and the NO_2 and the O_2 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 NO_2. For a plant with a storage mass of 35,000 metric tons, the total NO_2 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, NO_2 produced from magnesium decomposition comes into contact with H_2O released from the salt. The two combine to form nitric acid, H_2NO_3. 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.

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.

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 ± 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.

<u>Post Weld Heat Treatment of Type 347H Pipe and Fittings</u>

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 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 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.

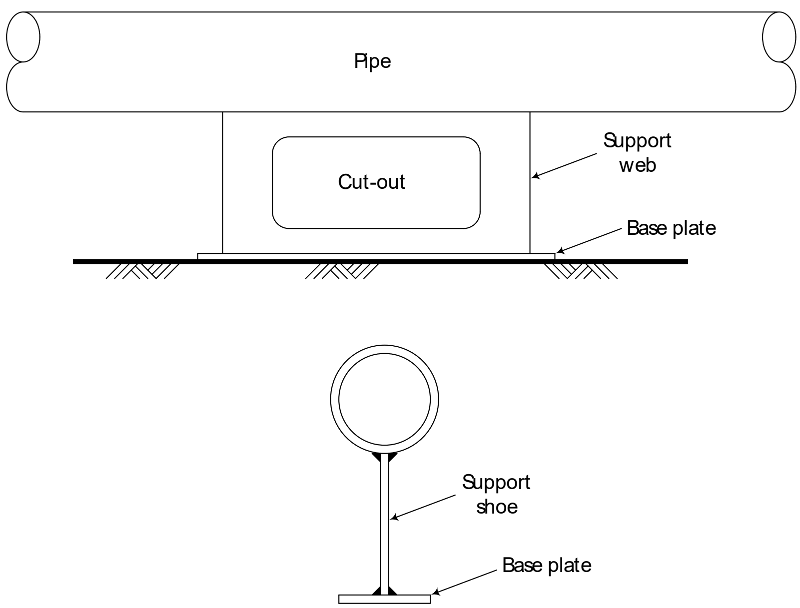

Figure 6-2 Pipe Support, Anchor, and Guide

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.

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.

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 °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 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 °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 the stem packing relies on graphite, and will require periodic replacement, regardless of the valve temperature.

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.

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 plates. A potential design of a flow venturi in combination with diaphragm pressure transmitters is shown in Figure 6-3.

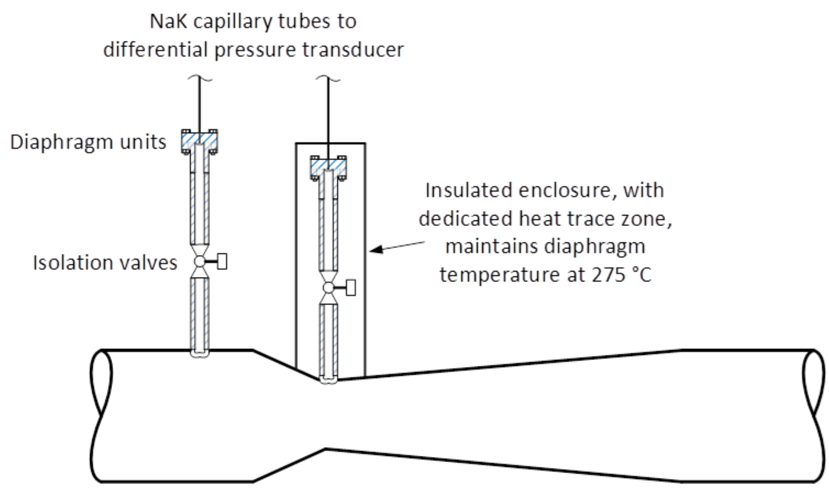

Figure 6-3 Flow Venturi with Diaphragm Pressure Transmitters

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 °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.

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.

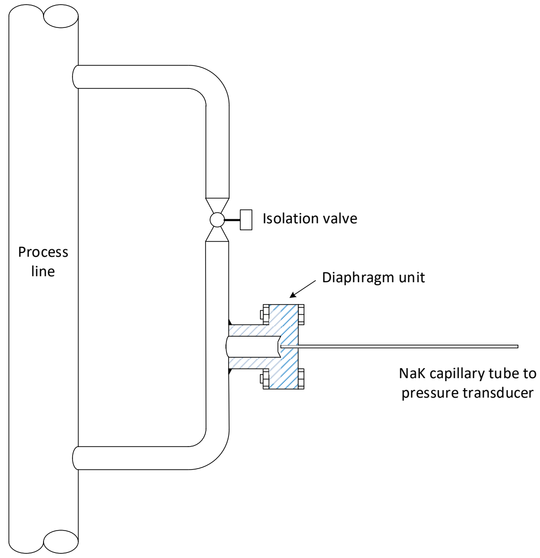

Figure 6-4 Pressure Transmitter in a Vertical Process Line

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.

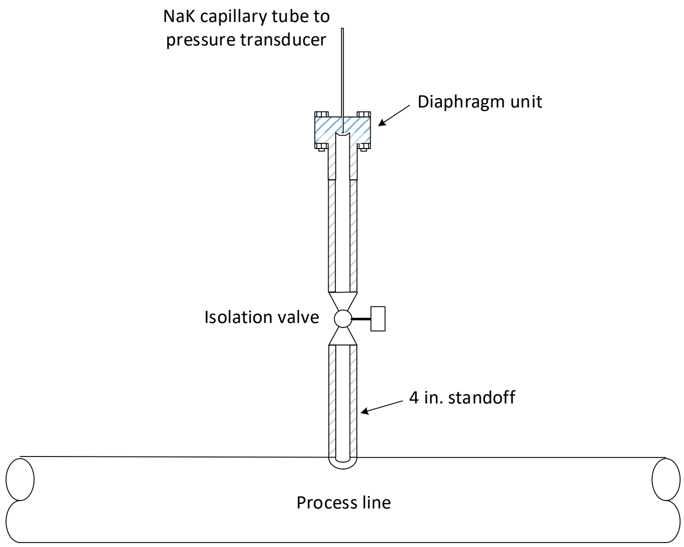

Figure 6-5 Pressure Transmitter in a Horizontal Process Line

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

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.

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.

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.

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.

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: $^{5}/_{16}$-inch (nominal 8 mm) for 600 V service for long piping zones; and $^{3}/_{16}$-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

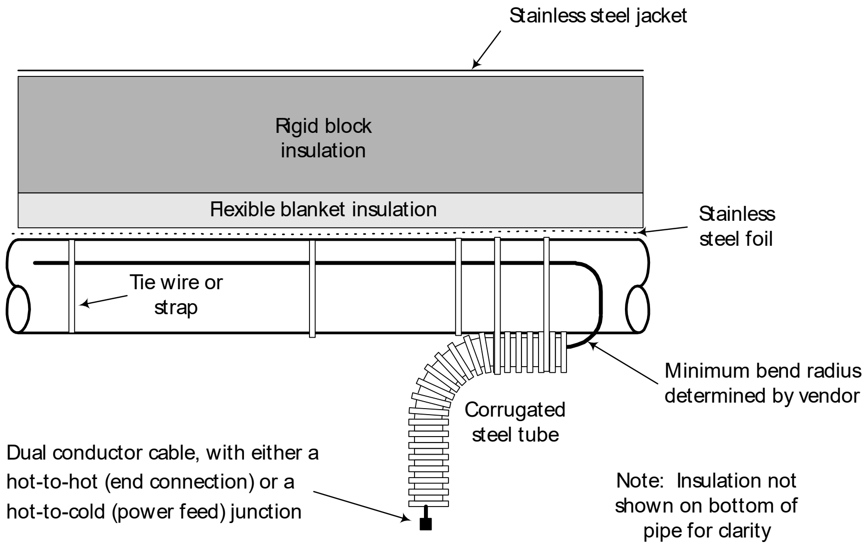

Figure 6-6 Heat Trace Arrangement on Pipe at End of Zone

- 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 °C to 300 °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.

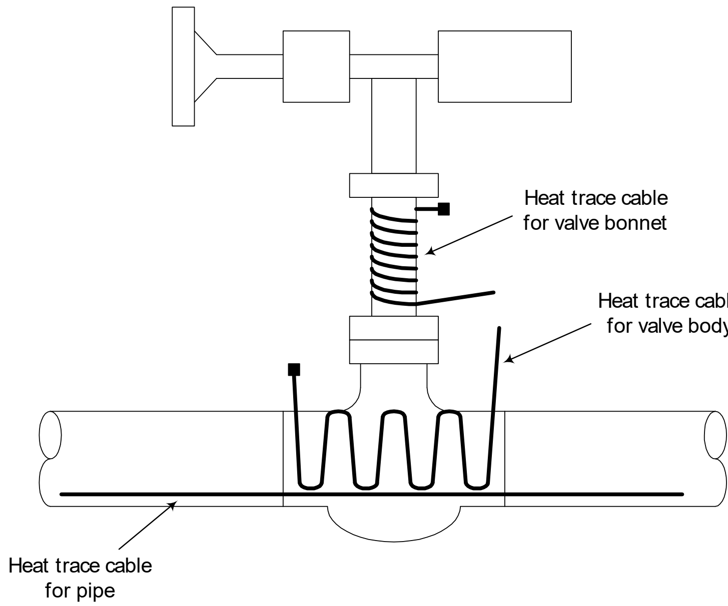

Figure 6-7 Typical Heat Trace Cable Arrangement on Valves

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

- 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.

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.

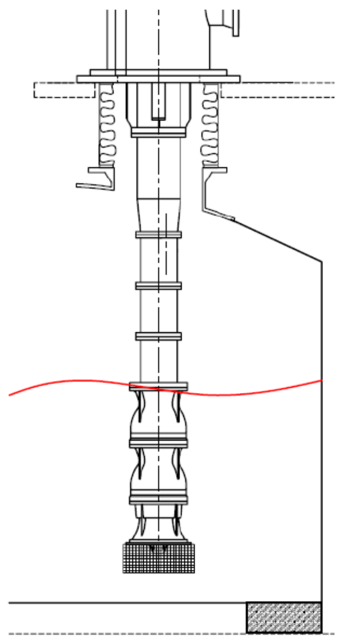

Figure 6-8 Elevation View of Salt Pump

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.

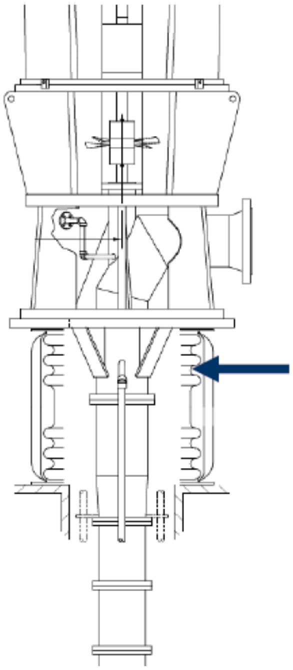

Figure 6-9 Detail of Salt Pump Discharge Head and Bellows

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.

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 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.

<u>Option 1</u>

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.

<u>Option 2</u>

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

- 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.

<u>Option 3</u>

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-pecent 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 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.

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 size 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:

- 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 $(FeNi)_x(CrMo)_y$. 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.

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 attemperation 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

- 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

- 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 a 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

- 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.

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 a 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.

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.

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.

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 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.

Table 6-2 Temperature Cycles for the Cold Salt Tank

Type of cycle	Number in 30 years	Duration, hours	Temperature, °C Beginning	Temperature, °C Ending	Rate of temperature change [1], °C / hr
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.

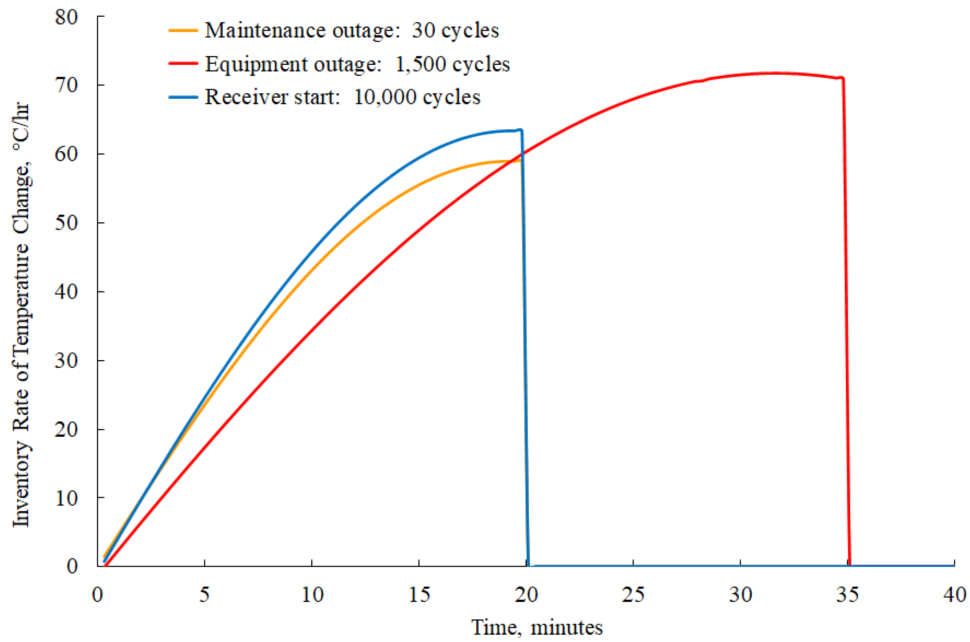

Figure 6-10 Rates of Change in the Cold Tank Inventory Temperature

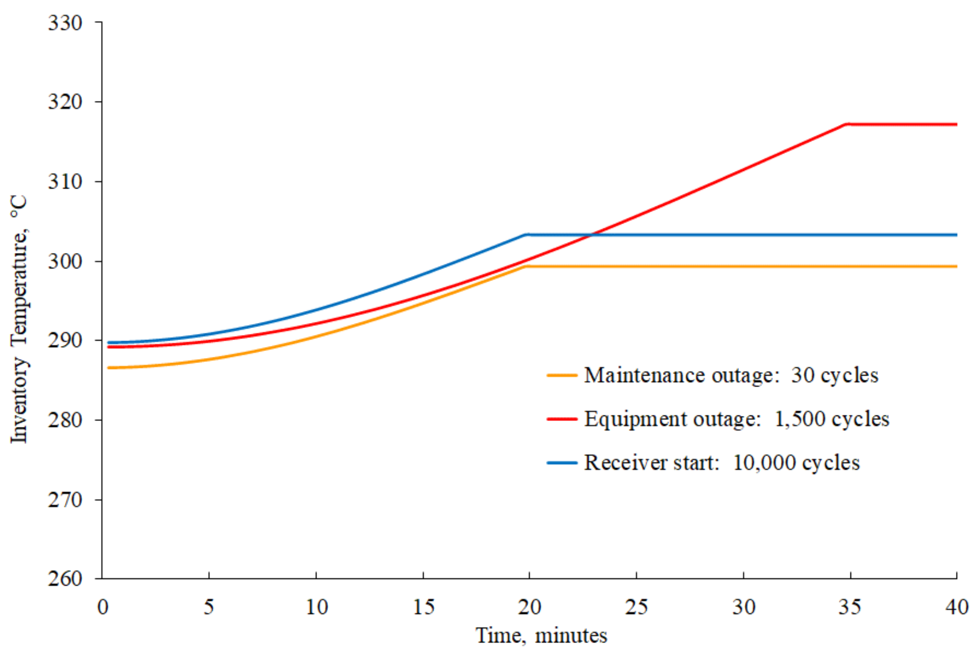

Figure 6-11 Cold Tank Inventory Temperatures versus Time

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 (~ 520 °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 (~ 550 °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

Type of cycle	Number in 30 years	Duration, hours	Temperature, °C Beginning	Temperature, °C Ending	Rate of temperature change [1], °C/hr
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]

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.

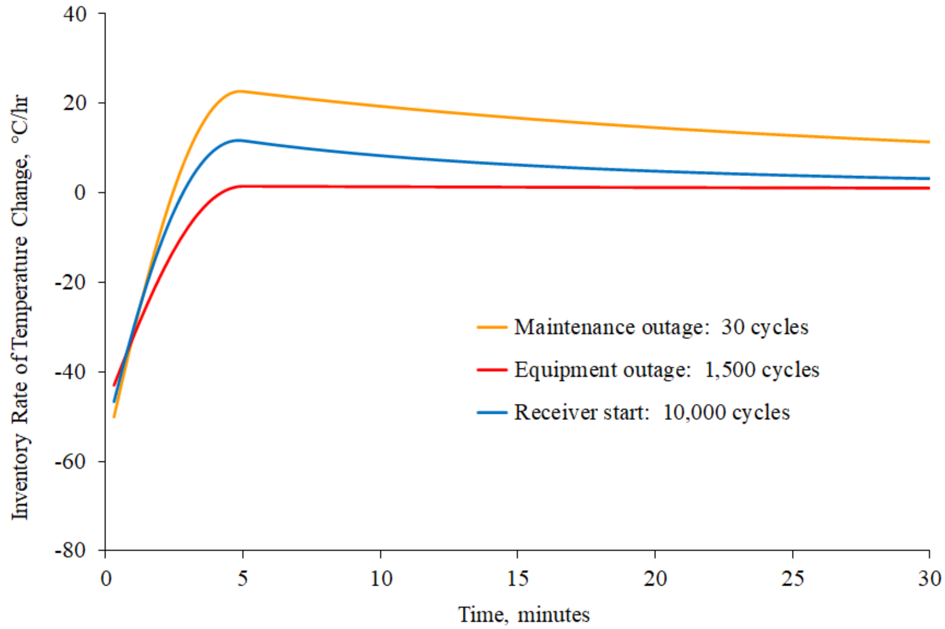

Figure 6-12 Rates of Change in the Hot Tank Inventory Temperature

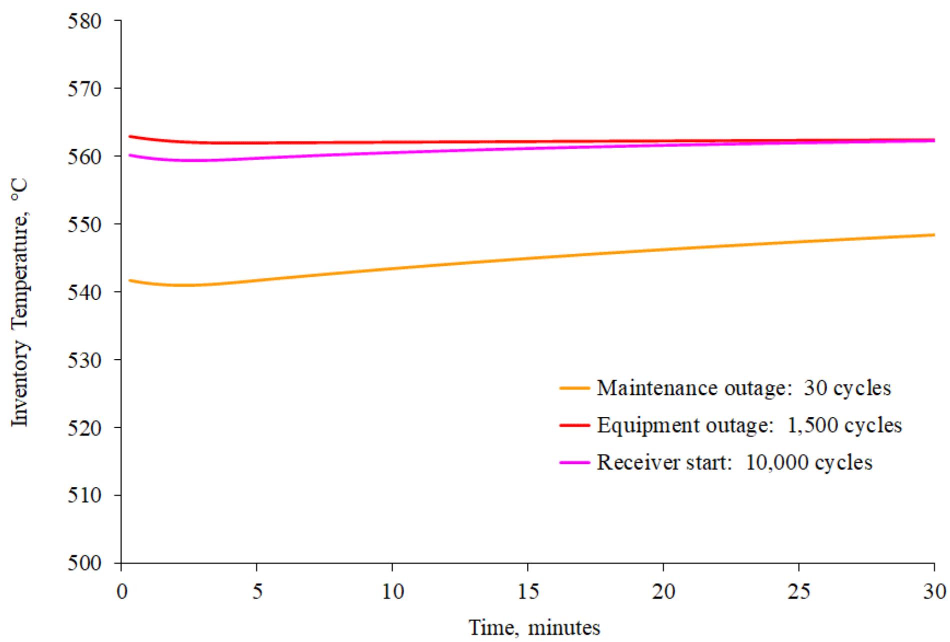

Figure 6-13 Hot Tank Inventory Temperatures versus Time

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:

<u>Receiver Trip</u> 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.

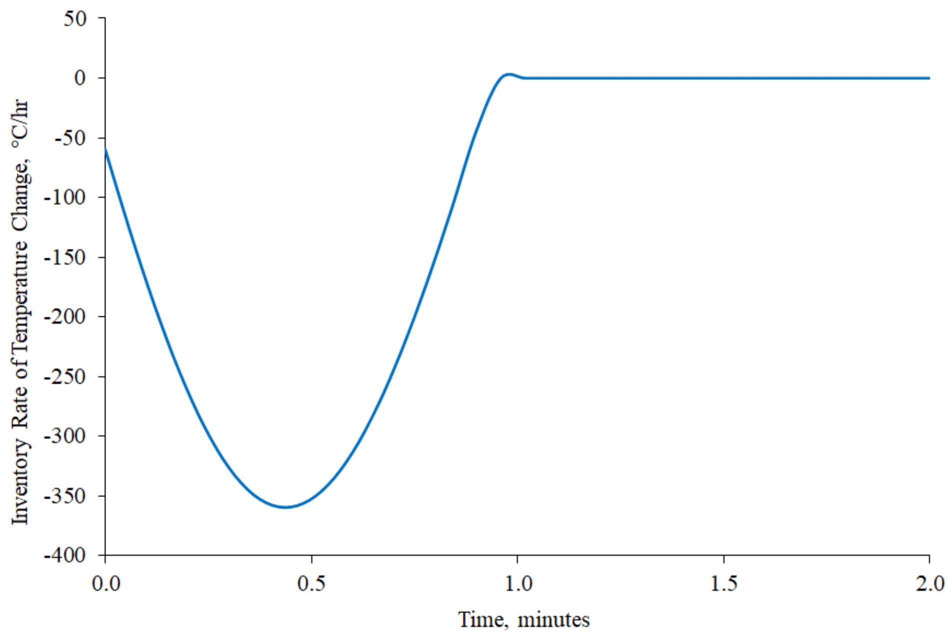

Figure 6-14 Rate of Change in the Hot Tank Inventory Temperature Following a Receiver Trip

<u>Selected Crossover Temperature</u> 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.

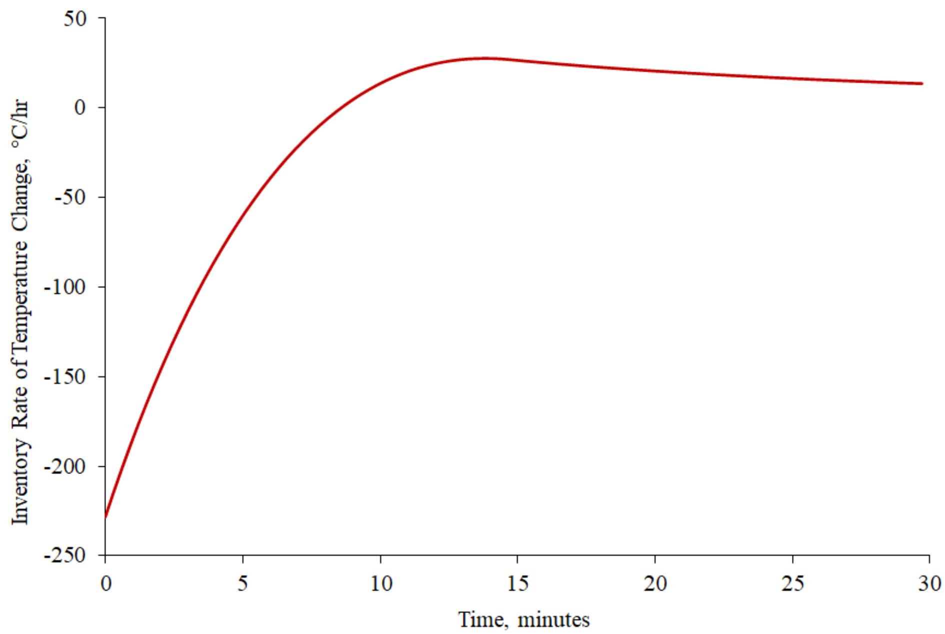

Figure 6-15 Effect of a Crossover Temperature 90 °C Lower than the Inventory Temperature

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 with the nitrite ion (NO_2^-) and oxygen in the ullage gas in the storage tanks based on the following reaction:

$$NO_3^- \leftrightarrow NO_2^- + \frac{1}{2} O_2$$

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:

$$NO_2^- \rightarrow O^- + NO$$

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 NOx 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 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 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

- 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 NOx 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.

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/m^2. 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

- 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

- 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

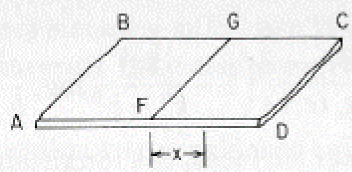

Figure 15.5

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.

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.

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 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 m^2-°C/W (1/2000 Btu/hr-ft^2-°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 m^2-°C/W (1/1000, 1/2000, and 1/3000 Btu/hr-ft^2-°F), and 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 (~ 240 °C) to satisfy this requirement. However, during turbine startup and low load operation, final feedwater temperatures are in the range of 180 to 230 °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 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 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.

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.

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.

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.

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.

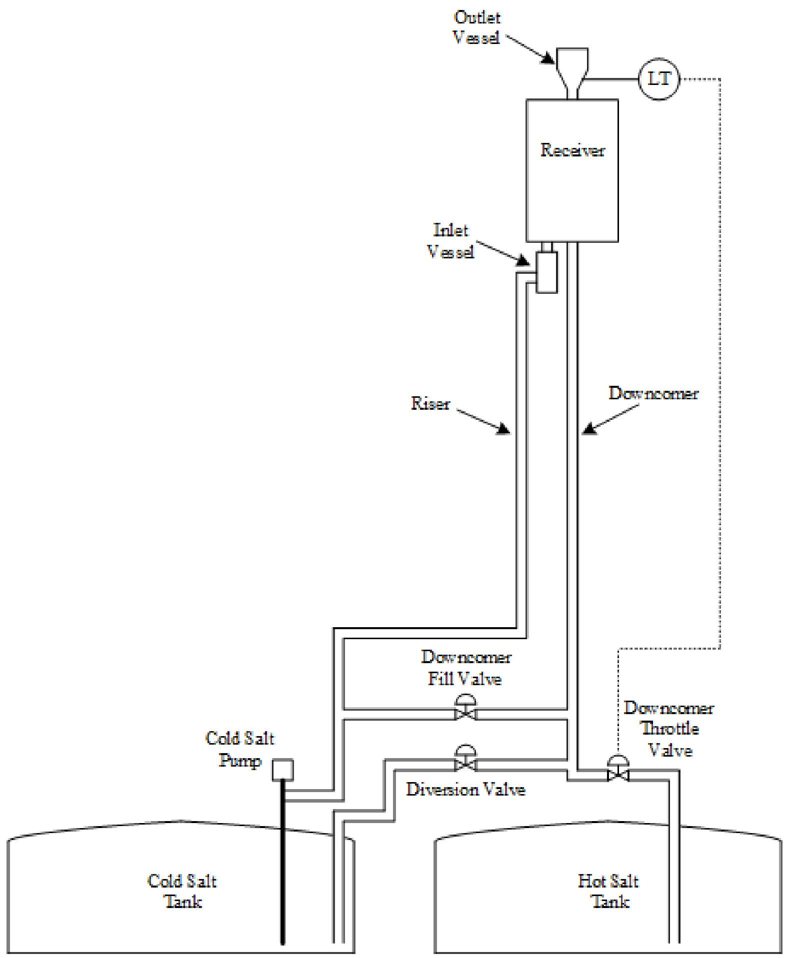

Figure 7-1 Receiver Outlet Vessel and Control Valve at the Solar Two Project

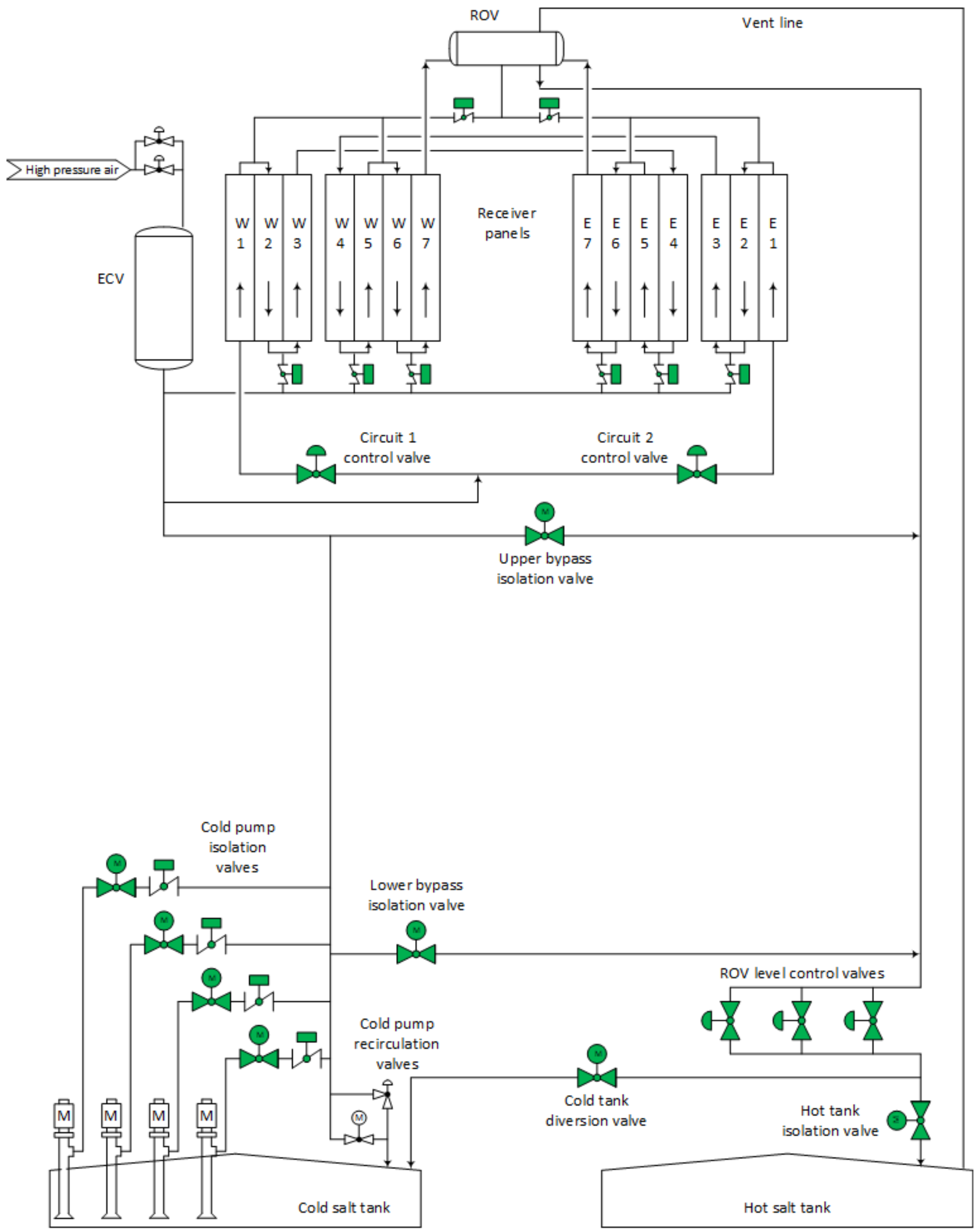

Figure 7-2 Receiver Equipment Arrangement in a Representative Commercial Project

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.

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.

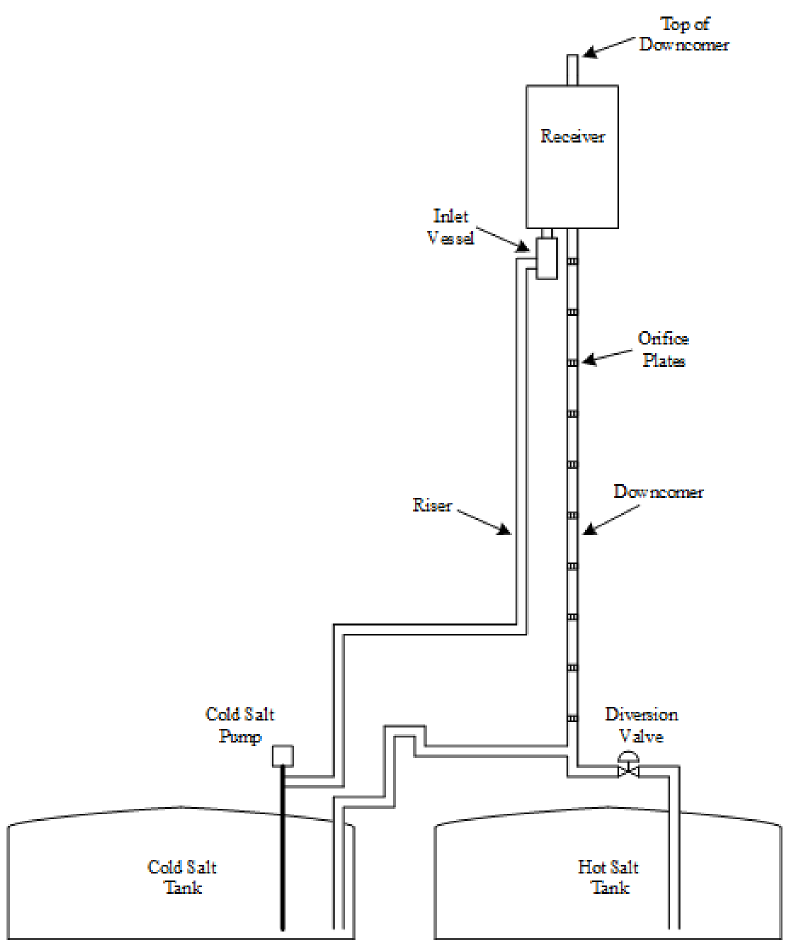

Figure 7-3 Downcomer with Periodic Orifice Plates for Throttling

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 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.

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/m^2. 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/m^2 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 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.

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.

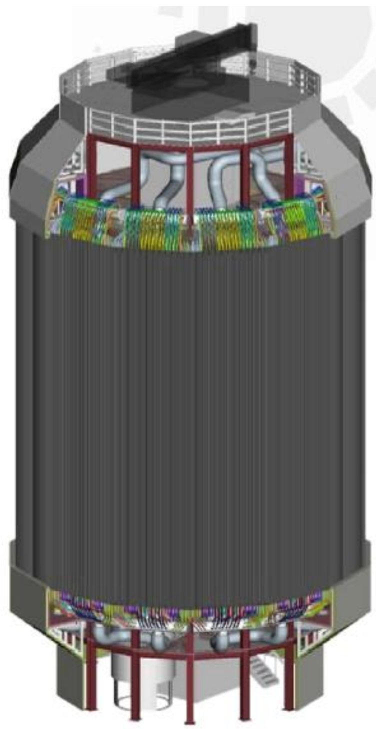

Figure 7-4 External Salt Receiver Arrangement

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

- 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.)

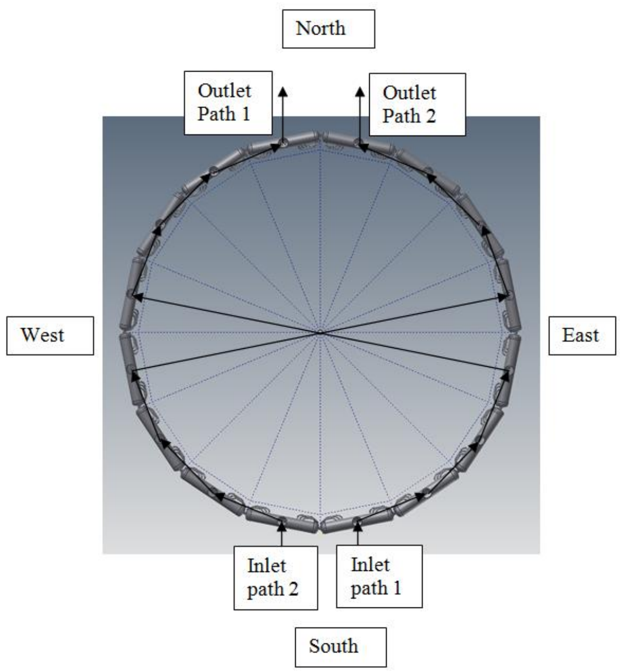

Figure 7-5 Circuit Flow Paths with Single Crossover Between Circuits

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 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/m^2.

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/m^2.

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/m^2. 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:

- 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/m^2.

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

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

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/m^2. 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/m^2 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 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 7-8 [4].

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.

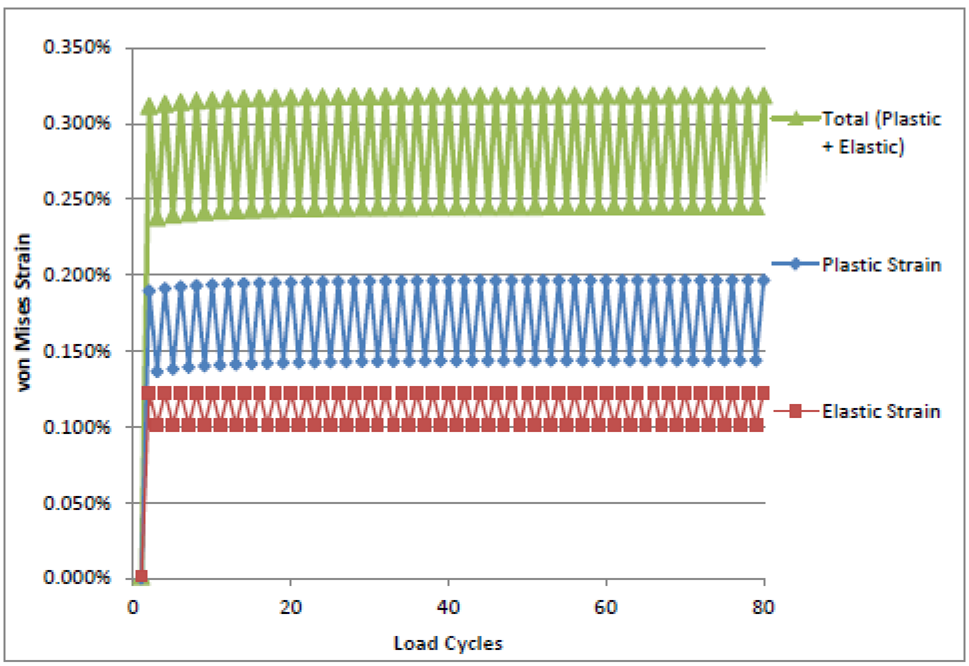

Figure 7-8 Cyclic Tube Strains on Successive Cycles

[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

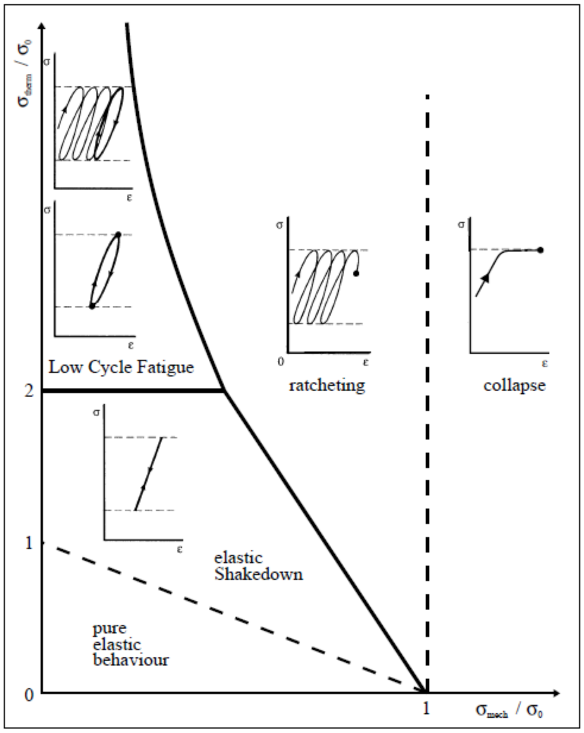

Figure 7-9 Bree Diagram of Primary Mechanical and Secondary Thermal Stresses

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 $(FeNi)_x(CrMo)_y$. 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 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 mm - 0.7 mm = 0.8 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 (Tube diameter * Pi + 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

- 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 7-10 [5].

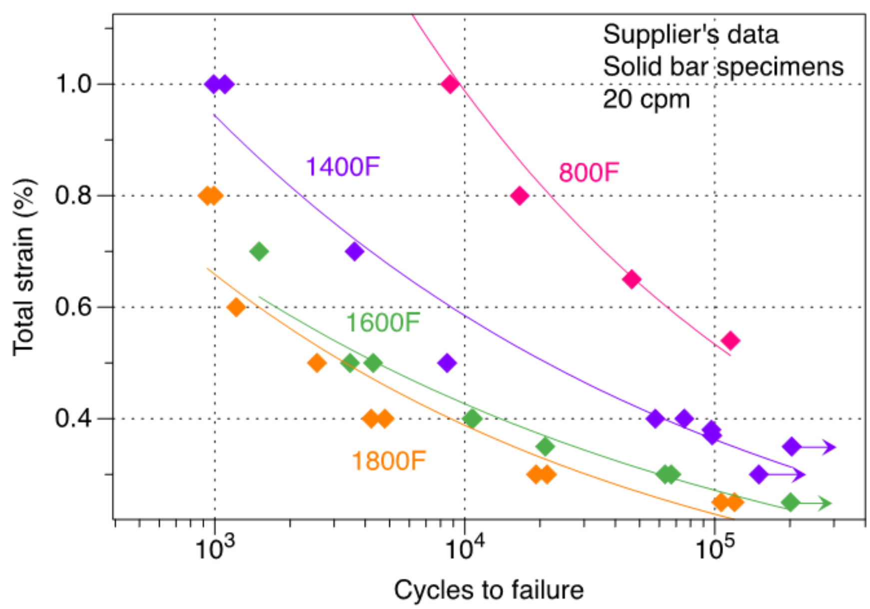

Figure 7-10 Low Cycle Fatigue Data for Alloy 230

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

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.

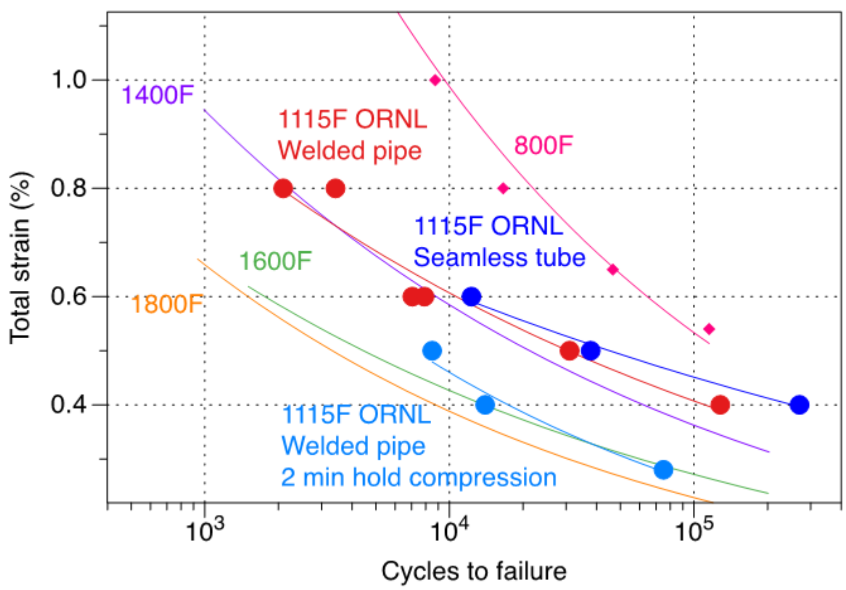

Figure 7-11 Low Cycle Fatigue Data for Alloy 230 with a 2 Minute Hold Time

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:

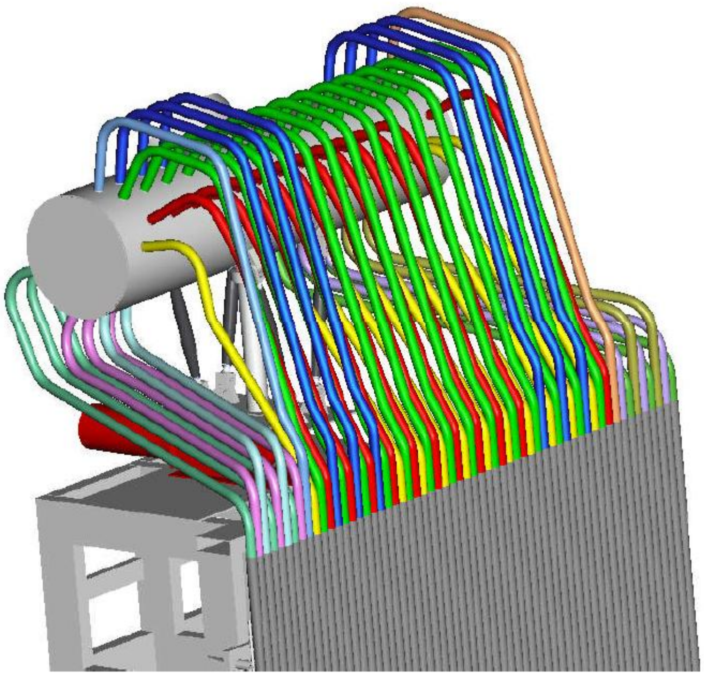

Figure 7-12 Tube-to-Header Connections

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.

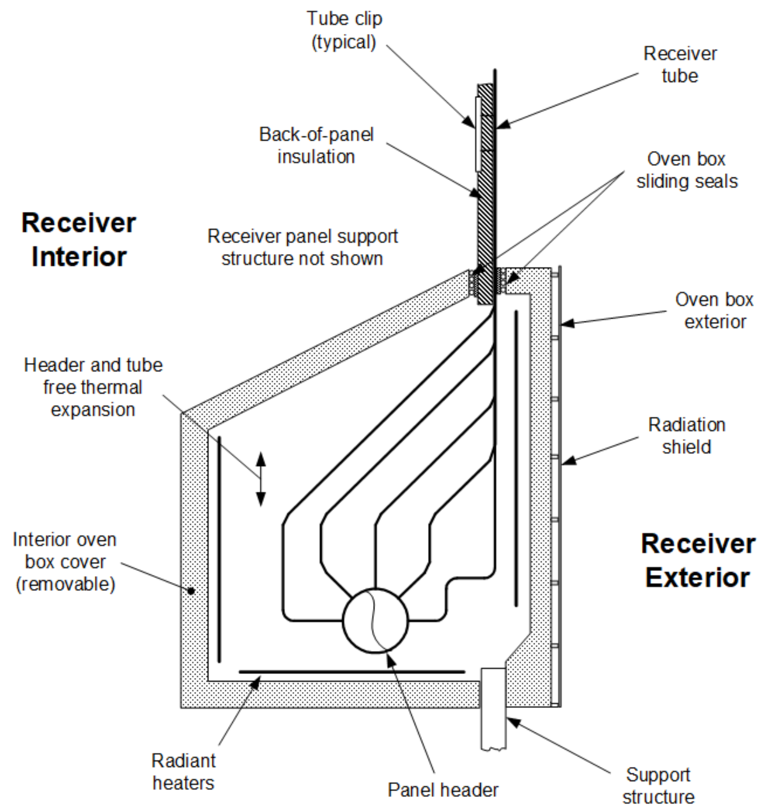

Figure 7-13 Cross Section of Receiver Lower Oven

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.

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/m^2
- 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 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.

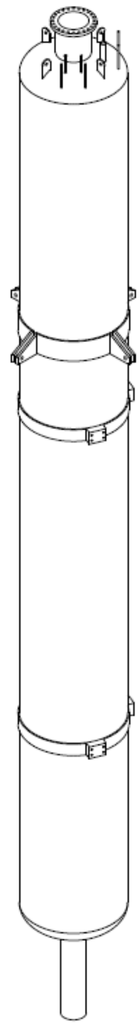

Figure 7-14 Sketch of a Representative Inlet Vessel

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

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 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:

$$H_{Mixed} = \sqrt[3.2]{H_{Natural}^{3.2} + H_{Forced}^{3.2}}$$

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.

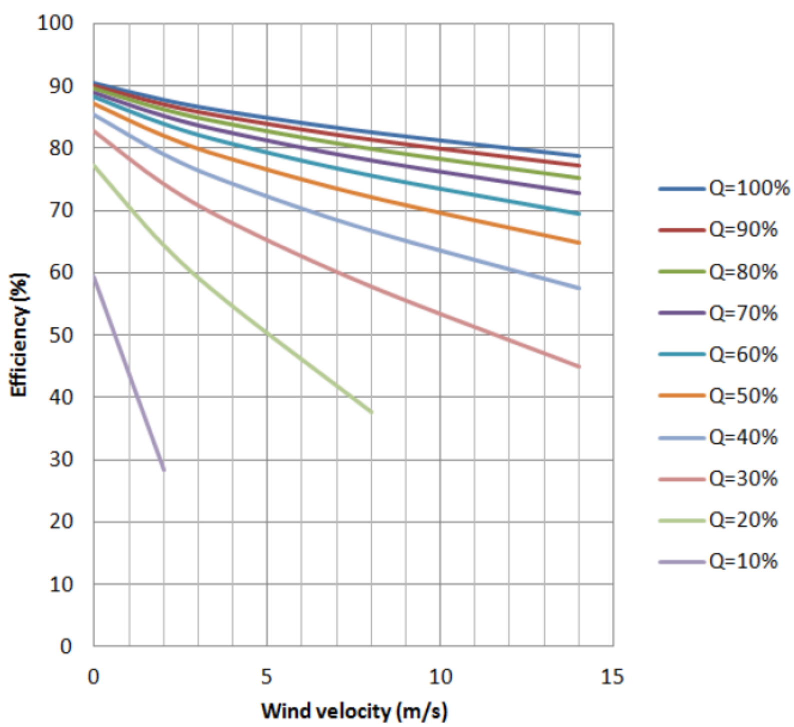

Figure 7-15 Receiver Efficiency as a Function of Input Duty and Wind Speed

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:

- 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/m^2, 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 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 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 MWht, 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.

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.

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 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 m^3/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 attemperation 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.

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.

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

- 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 °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 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.

Materials

<u>Hot Tank</u> 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 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, 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.

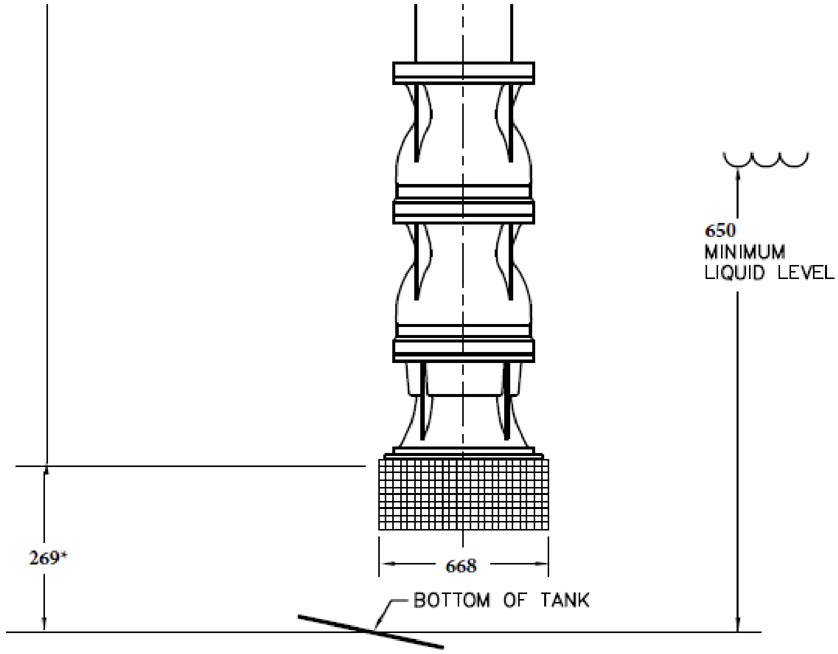

Figure 7-17 Salt Pump Clearance and Submergence

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/ft^2). 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.

To a first order, the thicknesses of the various wall sections can be calculated from the hoop stress equation; i.e., Thickness, m = (Static height, m) * (Density, kg/m^3) * (Acceleration due to gravity, m/sec^2) / (2 * 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.

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.

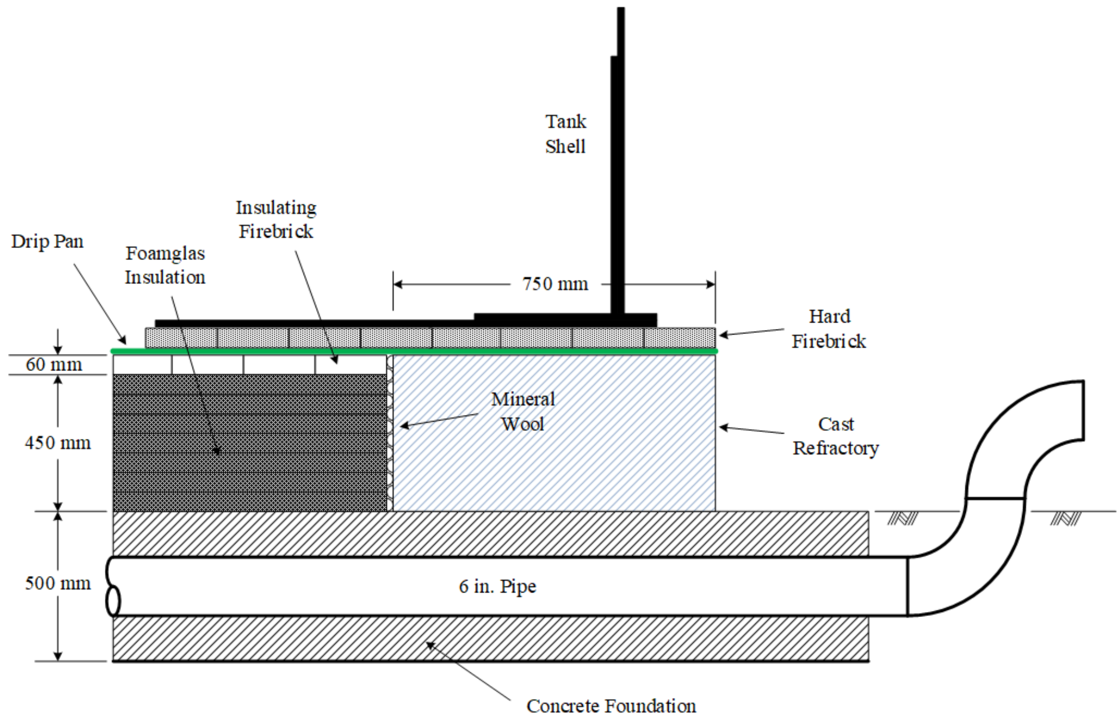

Figure 7-18 Representative Cross Section of Tank Foundation

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.

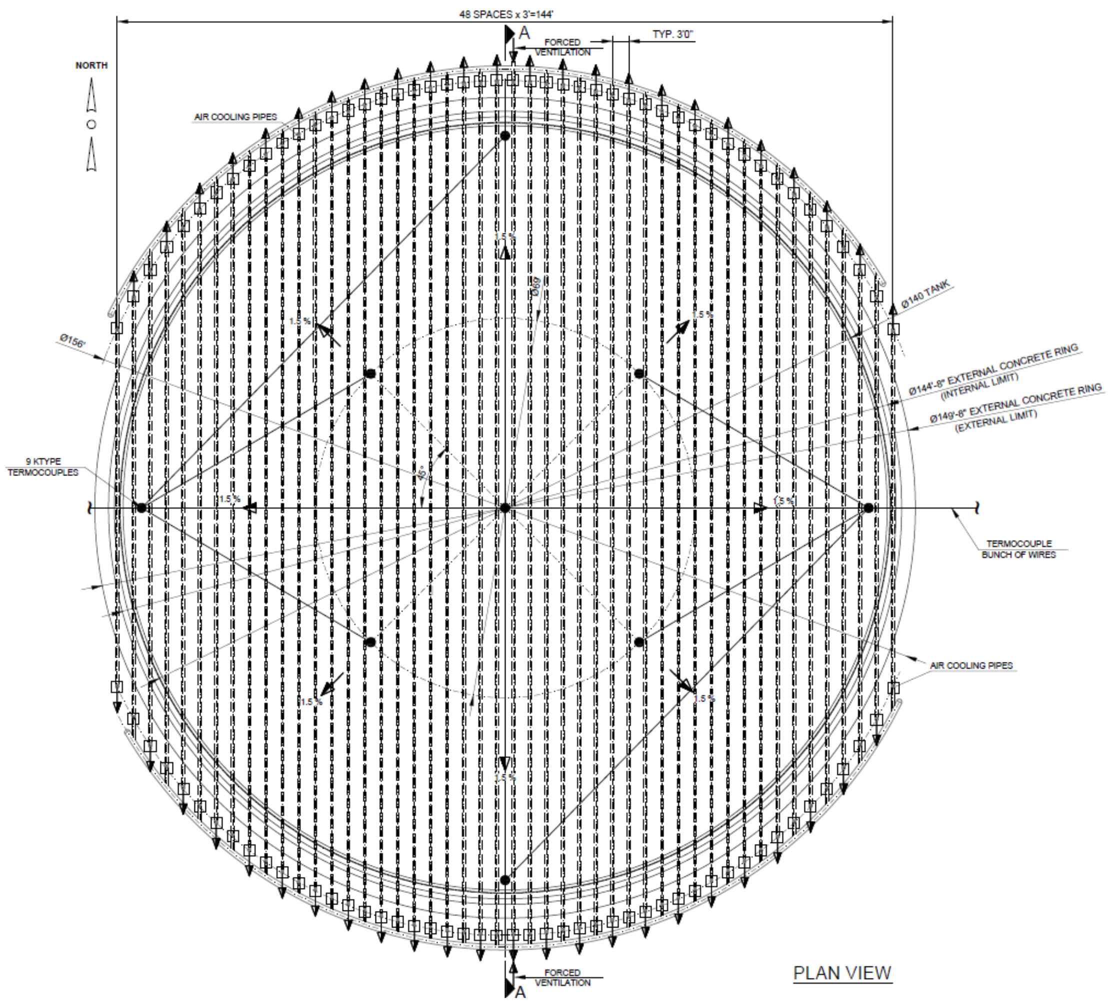

Figure 7-19 Foundation Cooling Air Passage Layout

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.

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 N_2, O_2, H_2O, CO_2, and NOx. The H_2O and the CO_2 will react to form carbonic acid, H_2CO_3, 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.

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

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

$$\sigma_{r_1} = \gamma E\left(\frac{1}{R^2}\int_0^R Tr\,dr - \frac{1}{r_1^2}\int_0^{r_1} Tr\,dr\right)$$

$$\sigma_{t_1} = \gamma E\left(-T + \frac{1}{R^2}\int_0^R Tr\,dr + \frac{1}{r_1^2}\int_0^{r_1} Tr\,dr\right)$$

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., 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-sec^2 (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/m^3
H is the depth of the salt, m
g is the acceleration due to gravity, m/sec^2
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

- The temperature of the inventory changes by an amount (~ 15 °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 °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.

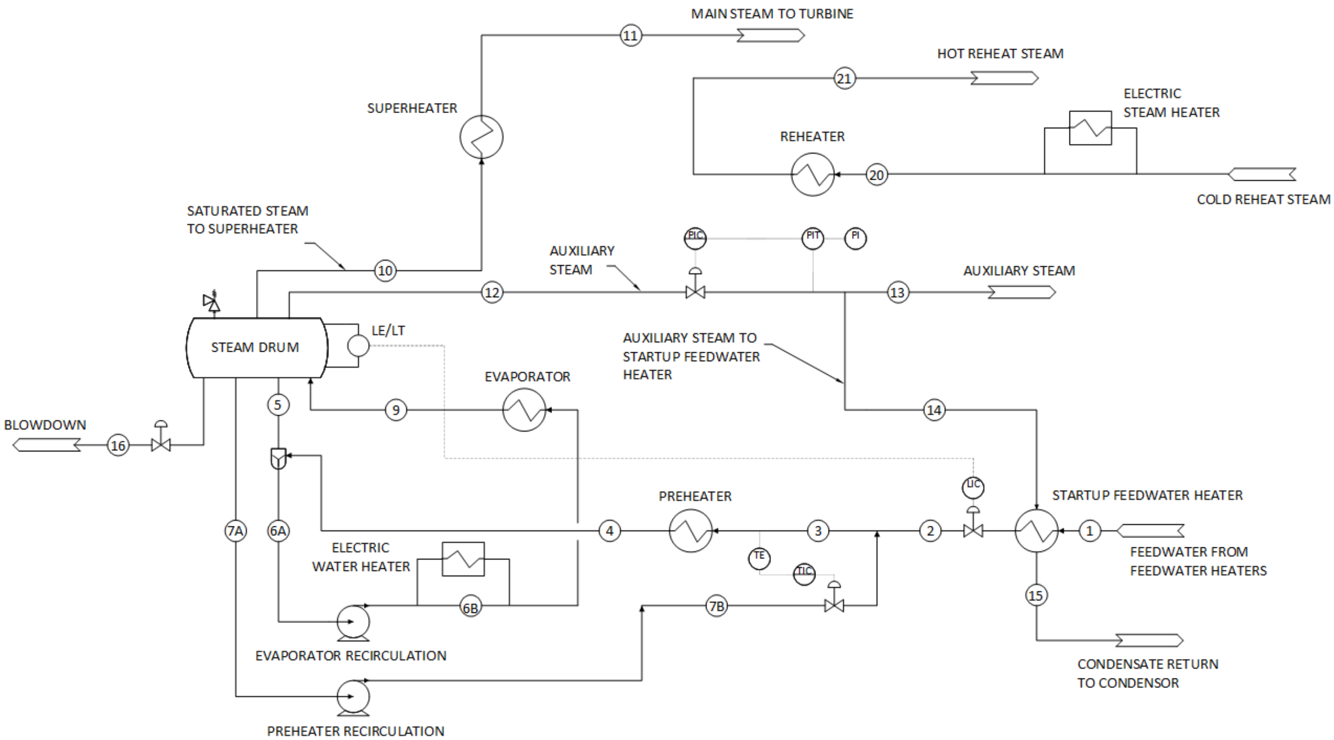

Figure 7-20 Flow Diagram on the Water/Steam Side of the Steam Generator

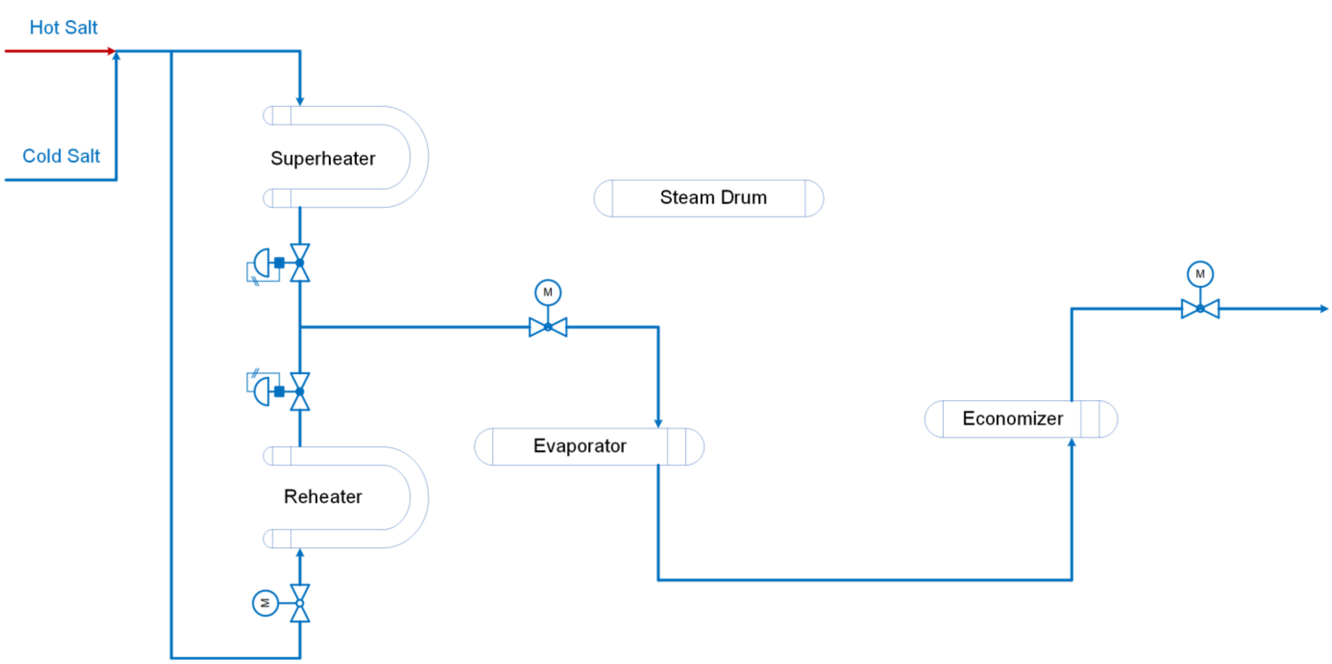

Figure 7-21 Flow Diagram on the Salt Side of the Steam Generator

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.

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

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.

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

- 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
- 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 7-22, and the intra-heat exchanger salt valve arrangement is shown in Figure 7-23.

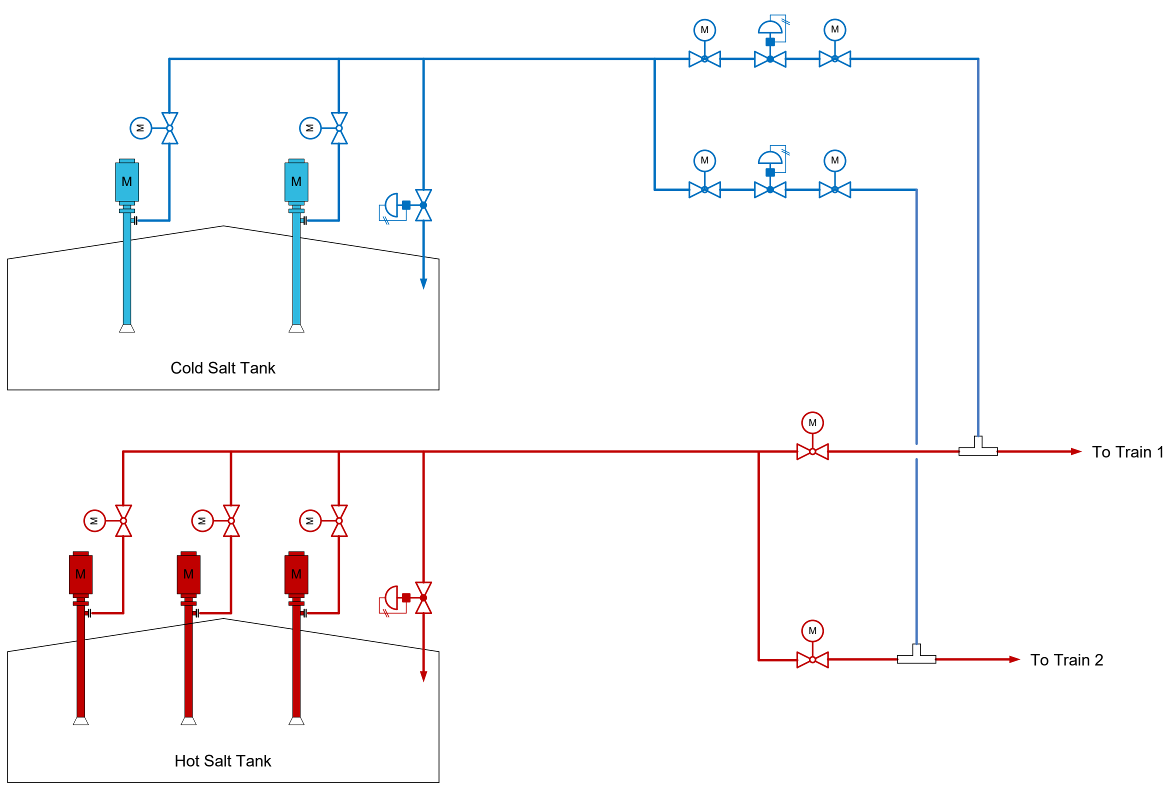

Figure 7-22 Salt Pump Arrangement in a Representative Commercial Project

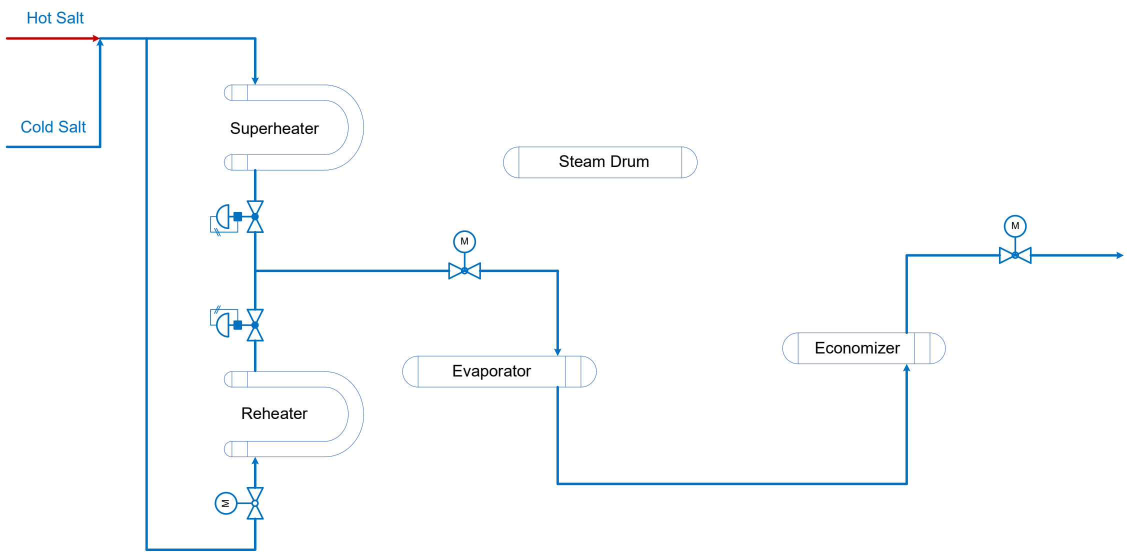

Figure 7-23 Intra-Heat Exchanger Salt Valve Arrangement in a Representative Commercial Project

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 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.

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/m^3
- 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 ($NaNO_3$), and 36 mole percent potassium nitrate (KNO_3), which is equivalent to 60 weight percent $NaNO_3$, and 40 weight percent KNO_3.

Note: The mixture is not the eutectic. The eutectic is a mixture of 50 mole percent $NaNO_3$, and 50 mole percent KNO_3. For solar applications, the fraction of $NaNO_3$ is increased to reduce the cost of the salt. Increasing the $NaNO_3$ 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.

<u>Temperature range</u> The salt mixture can be used over a temperature range of 260 °C to approximately 621 °C.

<u>Freezing point</u> As temperature decreases, the mixture starts to crystallize at 238 °C, and is completely solid at 221 °C.

<u>Isotropic compressibility ($NaNO_3$) at the melting point</u> $2 * 10^{-10}$ m^2 / N.

Heat of fusion (based on the average of heat of fusion of each component) h_{sl} = 161 kJ/kg

Change in density upon melting $\Delta V / V_{solid} = 4.6\% \Rightarrow V_{liquid} = 1.046\ V_{solid}$

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:

$$\rho\ (lb_m / ft^3) = 131.2 - 0.02221 * T\ (°F)$$
$$\rho\ (kg / m^3) = 2090 - 0.636 * T\ (°C)$$

Specific heat, as a function of temperature:

c_p (Btu / lb_m- °F) = 0.345 + ($2.28*10^{-5}$) * T (°F)
c_p (J / kg - °C) = 1443 + 0.172 * T (°C)

Absolute viscosity, as a function of temperature:

μ (lb_m / ft - hr) = 60.28440 - 0.17236 * T (°F) + ($1.76176*10^{-4}$) * $(T (°F))^2$ - ($6.11408*10^{-8}$) * $(T (°F))^3$
μ (mPa - sec) = 22.714 - 0.120 * T (°C) + ($2.281 * 10^{-4}$) * $(T (°C))^2$ - ($1.474*10^{-7}$) * $(T (°C))^3$

Thermal conductivity, as a function of temperature:

k (Btu / hr - ft - °F) = 0.253208 + $6.26984 * 10^{-5}$ * T (°F)
k (W / m - °C) = 0.443 + $1.9 * 10^{-4}$ * T (°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