SolarDynamics LLC

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

DOE Grant Number DE-EE0009810

Volume 1 - Specifications for Parabolic Trough Projects

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1. Introduction, Report Format, Acknowledgements, and Disclaimer

1.1 Introduction

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

1.2 Project Goal

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

1.3 Report Format

The report consists of 3 volumes:

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

Volumes 1 and 2 include discussions of the following topics:

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

- 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 Description

2.1 Introduction

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

2.2 Collector System

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

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

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

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

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

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

- Electric impedance heating for the receiver tubes and the flex hose assemblies is added for preheating and freeze protection

- Electric heat tracing on the collector field piping and valves is added for preheating and freeze protection.

2.3 Heat Transfer Fluids

2.3.1 Organic Fluids

Working Fluid

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

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

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

Therminol System

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

- Expansion vessels to accept the change in volume of the Therminol between the minimum circulation temperature (~20 °C) and the maximum working temperature (393 °C)

- Nitrogen supply system to 1) provide an inert cover gas over the Therminol in the expansion vessels, and 2) pressurize the ullage gas to a nominal value of 10 bar to prevent flashing of the Therminol

- A Therminol cooler, to condense and recycle Therminol vapor released from the expansion vessels

- Activated carbon beds, to capture and adsorb thermal decomposition products of Therminol. The decomposition products are light hydrocarbons, such as benzene. The light hydrocarbons are removed to prevent vapor accumulation in the collector receiver tubes. Once the activated beds are saturated, the activated carbon is shipped offsite for disposal

- A distillation column to separate and remove heavy decomposition products of Therminol. The decomposition process which generates the heavy hydrocarbons proceeds rather slowly, and distillation of the Therminol inventory may not be needed until several years after the start of commercial operation.

2.3.2 Inorganic Fluids

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

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

2.4 Thermal Storage System

2.4.1 Organic Heat Transfer Fluids

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

- The unit cost of the fluid, in $/kWht of stored energy, is too high for commercial feasibility

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

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

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

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

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

- The difference in temperature between the cold Therminol supplied to the collector field (293 °C) and the hot Therminol supplied by the collector field (393 °C) is a modest 100 °C

- To provide a useful difference in temperature between the cold salt tank and the hot salt tank, the approach temperatures at both the cold end and at the hot end of the heat exchanger must both be 'small' (i.e., less than 10 °C) relative to the difference in temperature between the cold Therminol and the hot Therminol.

- The temperature of the Therminol supplied to the steam generator when the plant is operating from storage is lower than the temperature of the Therminol supplied to the steam generator when the plant is operating directly from the collector field. A reduction in the Therminol temperature translates into a reduction in the Rankine cycle efficiency and a reduction in the net plant output. To reduce the performance penalty when the plant is operating from storage, the approach temperature at the hot end of the heat exchanger must be as small as economically practical.

- Commercial plant designs, such as Solana, show the optimum approach temperatures, and the corresponding log mean temperature differences, to be in the range of only 5 to 7 °C. The small temperature differences, in combination with the modest thermal conductivity of nitrate salt, translate into large heat transfer areas and high equipment costs.

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

2.4.2 Inorganic Heat Transfer Fluids

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

2.5 Steam Generation System

2.5.1 Organic Heat Transfer Fluids

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

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

2.5.2 Inorganic Heat Transfer Fluids

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

- Recirculation pumps, which draw suction from the steam drum and supply saturated water to a mixing station upstream of the preheater

- A startup feedwater heater, which is supplied with saturated steam from the drum.

2.6 Electric Power Generation System

The electric power generation system converts the energy in the main and the reheat steam to electric power for delivery to the grid. The system consists of the turbine-generator, air cooled condenser, condensate system, deaerator, feedwater system, water and steam sampling system, turbine lubrication oil system, and associated pumps and rotating equipment.

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

2.7 Balance of Plant

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

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

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

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

2.8 Features of Commercial Projects

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

2.8.1 Site Characteristics

The optimum site would have the following characteristics:

- A contiguous section of land, with a shape and an overall area which allows the collector field to be arranged for 1) the best optical performance, and 2) a field header layout which minimizes the parasitic energy demand for pumping. Nonetheless, it can be noted that a commercial project in California was developed without the benefit of a contiguous site, and the annual electric output has been consistent with the projections in the performance model. The collector field is divided between two separate properties, with piping headers connecting the two sub-fields.

- A nominally flat surface, with a minimum of elevation changes, to reduce the amount of grading for drainage

- Acceptable parameters for the following:
  - Haul distance for soil and waste disposal
  - Distance to road and rail transportation connection points

- A location far enough from adjacent mountain ranges that may otherwise 1) block solar radiation early in the morning and late in the afternoon, or 2) channel winds onto the site.

2.8.2 Labor Availability

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

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

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

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

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

2.8.3 Land

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

2.8.4 Permits

All commercial projects will require land use and construction permits.

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

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

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

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

2.8.5 Solar Radiation

To a first order, the levelized cost of energy is inversely proportional to the annual direct normal solar radiation. The minimum practical annual radiation for a commercial project is in the range of 1,700 to 1,800 kWh/m^2-year. Many desert sites in the US and in North Africa have annual radiations in the range

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

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 3,100 kWh/m^2-year.

2.8.6 Access to Electric Power Transmission

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

2.8.7 Wind

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

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

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

2.8.8 Seismic Conditions

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

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

2.8.9 Range of Plant Sizes

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

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

2.8.10 Range of Storage Capacities

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

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

2.8.11 Water

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

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

2.8.12 Soil Properties

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

- Collector foundations. The foundations for the collector structure can consist of concrete piles, steel piles, screw piles, or concrete footings. The selected approach is based on the strength of the soil, the site design wind speed, the presence, or the absence, of rocks in the soil, and the depth of the ground water

- Thermal storage tank foundations. Essentially all commercial salt tanks use a concrete base mat underneath the floor insulation. This is an economical design, but it requires the weight of the static head of the salt to be less than the allowable bearing load for the soil. A representative soil bearing load is 240 kPa (5,000 lb_f/ft^2), which, in turn, limits the maximum salt depth in the tank to a nominal value of 12 m. If the allowable soil bearing load is less than 240 kPa, or if the salt tank is designed with a inventory height greater than 12 m, friction piles can be installed below the concrete base mat to increase the bearing capacity of the soil. However, the addition of friction piles will significantly increase the cost of the foundation

- Water table. The elevation of the water table should be significantly lower than the elevation of the concrete base mat for the storage tanks. A low water table helps to prevents liquefaction of the soil during an earthquake, and limits the vertical heat flux from the tank foundation into the soil

- Percolation. Following a rain, any standing water would ideally percolate into the soil in less than one day. This reduces the extent of puddles and soft soil, which could otherwise make access to the collector field problematic for maintenance vehicles.

2.8.13 Soiling of Mirrors

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

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

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

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

3.1 Introduction

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

- Energy collection section, consisting of the collector system and the thermal storage system (charging mode)

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

3.2 Operating States for the Energy Collection System

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

- Long Term Hold: The collector field is stowed. The field piping is filled (as there is no mechanism to drain the piping), and the Therminol pumps are off. The oil-to-salt heat exchangers are drained on the salt side, and the salt inventory is divided between the cold tank and the hot tank. The tank standby electric heaters are active to maintain the temperature of the salt inventory above the freezing point

- Overnight Hold / Standby: The collector field is stowed, and the Therminol pumps are circulating oil through the collector field. The oil-to-salt heat exchangers are filled on both the oil- and the salt-sides, but there is no oil flow or salt flow through the heat exchangers. The oil flow is stopped to ensure that potentially cold oil from the collector field does not cause salt to freeze in the heat exchanger. If the temperature of the heat exchanger starts to approach the freezing point of the salt, freeze protection can be provided by 1) activating the electric heat tracing on the heat exchanger, or 2) circulating salt from the cold tank, through the heat exchanger, and back to the cold tank

- Normal Operation: The collector field is tracking the sun, and the Therminol pumps are circulating oil through the collector field. Oil flow rates greater than that required by the steam generator are supplied to the oil-to-salt heat exchanger. The cold salt pump is in operation, supplying salt to the oil-to-salt heat exchanger. The speed of the cold salt pump is adjusted to provide a nominal salt temperature, at the hot end of the heat exchangers, of 385 °C.

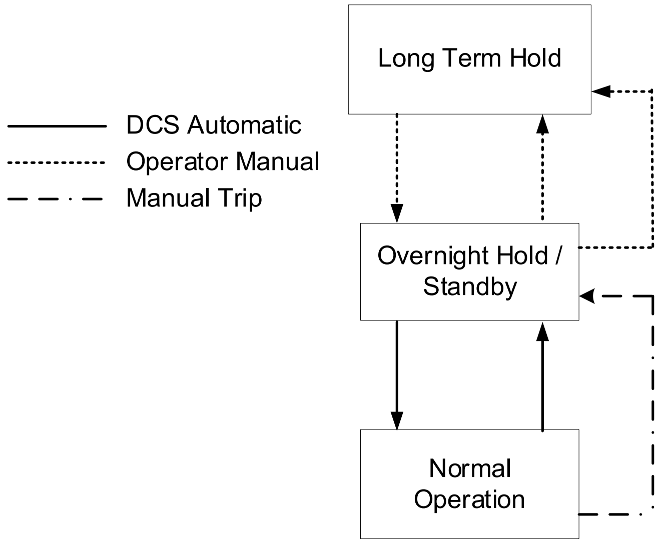

Figure 3-1 States and Transitions for the Energy Collection Section - Organic Fluids

3.3 Transitions Among Operating States for the Energy Collection System

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

- Long Term Hold to Overnight Hold / Standby: The Therminol pumps are started to establish a recirculation flow in the collector field. The oil-to-salt heat exchangers, which are filled on the oil side, are preheated to a nominal temperature of 290 °C by electric heat tracing, oil circulation, or a combination of both. The heat exchangers are filled with cold salt. The cold salt pumps operate in recirculation

- Overnight Hold / Standby to Normal Operation: The collector field is moved from the stow position to the track position. The speed of the Therminol pumps is adjusted to provide a collector field outlet temperature of 393 °C. Flows of Therminol and salt are established in the oil-to-salt heat exchangers, and the speed of the cold salt pump is adjusted to provide a nominal salt temperature of 385 °C at the inlet to the hot tank

- Normal Operation to Overnight Hold / Standby: The collector field is moved from the track position to the stow position. The Therminol flow is bypassed on the oil side of the oil-to-salt heat exchangers, and the cold salt pump is either operated in recirculation or stopped, depending on the length of the hold period

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

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

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

3.4 Operating States for the Energy Conversion System

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

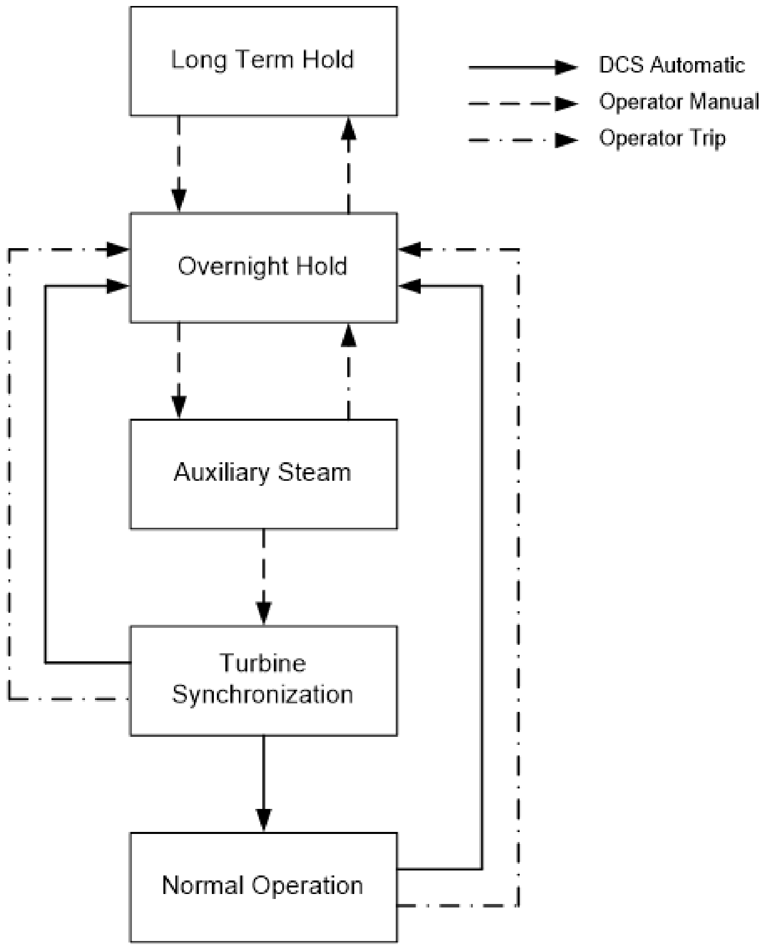

Figure 3-2 States and Transitions for the Energy Conversion Section - Organic Fluids

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

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

Notes:

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

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

- Long Term Hold: The oil-to-salt heat exchanger is drained on the salt side. The steam generator is drained on the water side, and the steam turbine is rotated on the turning gear

- Overnight Hold: The oil-to-salt heat exchanger is filled on the salt side. The cold salt pump is in operation, and salt supplied to the heat exchanger is recirculated to the cold salt tank. The steam generator is filled on both the Therminol and the water/steam sides; however, the steam production rate is 0 kg/sec. The steam turbine is rotated on the turning gear

- Auxiliary Steam: The auxiliary boiler is in service. Sealing steam, after superheating by an electric superheater, is supplied to the turbine shaft seals, and a vacuum is established in the condenser

- Turbine Synchronization: Once the Therminol temperature from the collector field reaches a value consistent with the metal temperatures in the steam generator, oil is supplied to the steam generator. The steam generator is started, and steam is dumped to the condenser until the steam conditions match the startup conditions for the turbine. The turbine-generator is synchronized with the grid, and a minimum turbine output is established

- Normal Operation: The extraction feedwater heaters are placed in service, and the output of the turbine-generator is increased to the rated value. Oil is supplied to the steam generator from the collector field, from the storage system, or from a combination of both.

3.5 Transitions Among Operating States for the Energy Conversion System

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

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

- Overnight Hold to Auxiliary Steam: The auxiliary steam generator is started, a flow of steam to the turbine shaft seals, after superheating by an electric superheater, is established, and a vacuum is drawn in the condenser

- Auxiliary Steam to Turbine Synchronization: The steam generator is placed in service when the temperature of the Therminol from the collector field is consistent with the metal temperatures in the steam generator. Live steam and the cold reheat steam production begin. The steam flows are throttled, and sent to the condenser through the bypass system. As soon as the live steam and the hot reheats steam temperatures satisfy the requirements set by the turbine vendor, a portion of the live steam and hot reheat steam flows are diverted from the bypass system to the turbine. The turbine generator is accelerated to 100 percent speed, synchronized with the grid, and the minimum turbine output is established

- Turbine Synchronization to Normal Operation: The extraction feedwater heaters are placed in service. The output of the turbine follows the thermal energy supplied to the steam generator by the flow of Therminol. The Therminol can be supplied by the collector field, by the storage system, or a combination of each

- Normal Operation to Overnight Hold: The flow of Therminol is reduced to a value consistent with the minimum turbine output. The turbine is then tripped to preserve the metal temperatures at values as high as practical during the subsequent hold period. Live steam and hot reheat steam is bypassed to the condenser until the steam generator temperatures reach the overnight hold condition, at which time the steam generator is tripped

- Overnight Hold to Long Term Hold: The oil-to-salt heat exchangers are drained on the salt-side, and the steam generator heat exchangers are drained on the water-side.

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

4.1 Introduction

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

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

4.2 Operating States for the Energy Collection System

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

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

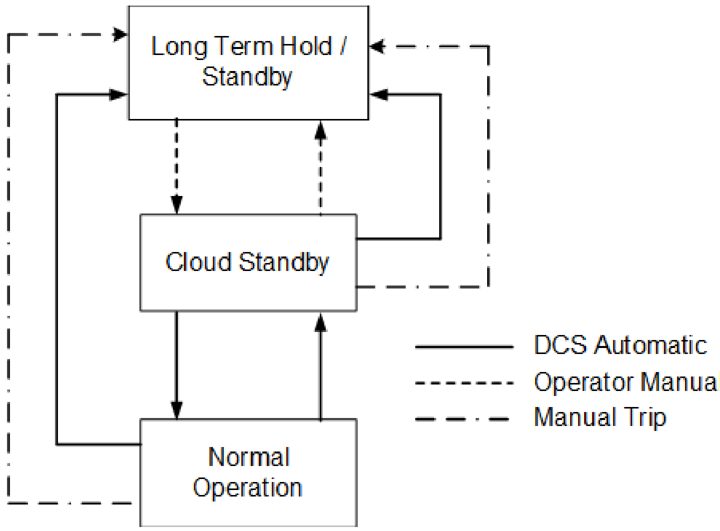

Figure 4-1 Energy Collection Section States and Transitions - Inorganic Fluids

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

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

4.3 Transitions Among the Operating States for the Energy Collection System

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

- Long Term Hold / Standby to Cloud Standby: The aperture normal radiation is not sufficient to establish a collector field outlet temperature equal to the design value. The speed of the cold salt pump is increased to a value which establishes the clear sky flow rate. The collectors are moved from the stow position to the track position, and the flow from the field is recirculated back to the cold tank

- Cloud Standby to Normal Operation: The aperture normal radiation is sufficient to establish a collector field outlet temperature equal to the design value, and the speed of the cold salt pumps is adjusted to reach this set point. When the collector field reaches the crossover temperature (approximately 500 °C), the flow is diverted from the cold tank to the hot tank

- Normal Operation to Cloud Standby: The aperture normal radiation is no longer sufficient to maintain a collector field outlet temperature equal to the design value. The speed of the cold salt pump is increased to a value which establishes the clear sky flow rate. When the collector field falls below the crossover temperature (approximately 500 °C), the flow is diverted from the hot tank to the cold tank

- Normal Operation to Long Term Hold / Standby: The aperture normal radiation is no longer sufficient to maintain a collector field outlet temperature equal to the design value. The collector field is moved to the stow position, and the speed of the cold salt pumps is reduced to the Long Term Hold / Standby condition

- Cloud Standby to Long Term Hold / Standby: The collector field is moved to the stow position, and the speed of the cold salt pumps is reduced to the Long Term Hold / Standby condition.

4.4 Operating States for the Energy Conversion Section

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

- Long Term Hold: The steam generator is drained on both the water side and the salt side, the electric heat trace circuits are inactive, and the electric recirculation water heaters are inactive

- Overnight Hold: The attemperation pump supplies cold salt to the steam generator to maintain heat exchangers temperatures above 275 °C. The steam turbine is rotated by the turning gear. Depending on the requirements of the project, turbine shaft sealing steam may need to be produced to maintain condenser vacuum

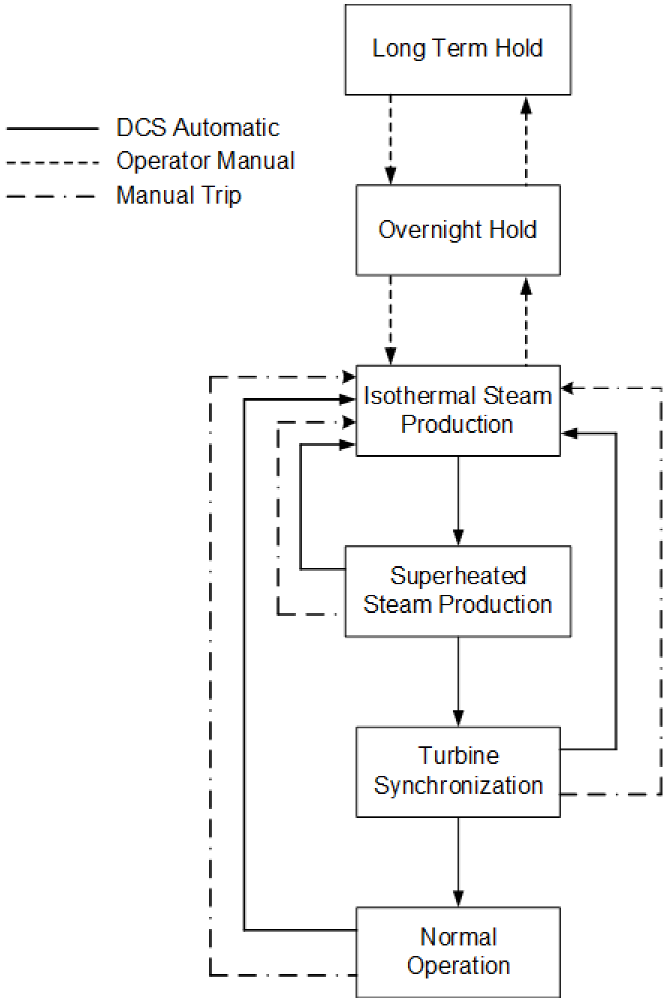

Figure 4-2 Energy Conversion Section States and Transitions - Inorganic Fluids

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

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

- Isothermal Steam Production: The auxiliary steam generator is in service, supplying shaft sealing steam to the turbine. A vacuum is drawn in the condenser. Cold salt is supplied to the hot end of the superheater and to the hot end of the reheater. The salt flows from the cold ends of the two steam heat exchangers combine at a hot salt / cold salt mixing station upstream of the hot end of the evaporator. The mixing station is illustrated in the process flow diagram Figure 6-7 of Volume 3 - Narrative. Thermal energy in the hot salt is sufficient to establish a saturated steam flow rate equal to 20 percent of the design steam flow rate. The saturation pressure in the evaporator is set by the turbine bypass system such that the saturation temperature is equal to the cold salt temperature. The saturated steam flow from the evaporator passes through the superheater and then through the reheater. Since the salt temperature in the superheater and in the reheater is equal to the saturation temperature, no heat transfer occurs in either heat exchanger. However, during the period in which the steam flow rate increases from 0 percent to 20 percent (slightly above the vendor's minimum), non-uniform steam distributions in the heat exchangers are not likely to produce damaging stress distributions

- Superheated Steam Production: The salt mixing station is transferred from the startup point upstream of the evaporator to the normal point upstream of the superheater / reheater. The mixing station is illustrated in the process flow diagram Figure 6-8 of Volume 3 - Narrative. The steam production rates decays slightly to a nominal 16 percent (the vendor's minimum) due to the superheating duties of the superheater and the reheater

- Turbine Synchronization: A live steam flow rate of (Project specific) kg/hr, with a temperature and pressure of (Project specific) ºC and (Project specific) bar, respectively, as specified by the turbine vendor, are established. The turbine-generator is synchronized with the grid. A minimum turbine output of (Project specific) MWe is established to preclude generator trips on reverse power

- Normal Operation: The extraction feedwater heaters are placed in service. A live steam flow rate of (Project specific) kg/hr, with a temperature and pressure of (Project specific) ºC and (Project specific) bar, respectively, are established. For turbine loads less than approximately 90 percent, the turbine operates in fixed pressure control. This ensures that the pinch point temperature in the evaporator is high enough that the salt temperature at the cold end of the preheater remains above the minimum value. For turbine loads greater than 90 percent, the turbine operates in sliding pressure control to maximize the efficiency of the Rankine cycle. Turbine output is set by adjusting the speed of the hot salt pump. The design value for the hot reheat steam temperature is set by adjusting the distribution of the hot salt flow rate between the superheater and the reheater.

4.5 Transitions Between States for the Energy Conversion Section

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

- Long Term Hold to Overnight Hold: The temperatures of the steam generator heat exchangers and the inter-vessel piping are raised to 275 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, which is representative of the minimum flow rate set by the heat exchanger vendor

- Superheated Steam Production to Turbine Synchronization: When the temperatures of the live steam and the reheat steam meet the startup requirements of the turbine, a portion of the steam in the bypass system is diverted to the turbine, the turbine is accelerated to synchronous speed, the turbine is synchronized with the grid, and output of the turbine is increased to the point where the generator is not at risk of tripping on reverse power. (A note: Different turbine vendors have different mechanisms for starting their turbines.) Steam not required by the turbine flows to the condenser through the bypass system

- Turbine Synchronization to Normal Operation: The extraction feedwater heaters are placed into service, the balance of the steam flow in the bypass system is sent to the turbine, the attemperation flow of cold salt is reduced consistent with an the maximum allowable rate of temperature change for the superheater / reheater, and the flow rate of hot salt is increased to the design value

- Normal Operation to Isothermal Steam Production: The speed of the hot salt pump is reduced, which reduces the output of the turbine. When the turbine reaches the minimum allowable load, the turbine is tripped. Tripping the turbine at the full superheat / reheat temperature maintains the turbine metal temperatures as high as possible during the overnight hold period, which reduces the time, the thermal energy, required for the next startup. However, the steam generator is not tripped when the turbine is tripped, and the continuing steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of (only) cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent

- Isothermal Steam Production to Overnight Hold: The hot salt / cold salt mixing station is transferred from a point upstream of the superheater / reheater to the startup point upstream of the evaporator. During the transition between mixing stations, the superheating duty decreases relative to the evaporation duty. This results in 1) an increase in the steam flow rate to a nominal value of 20 percent, 2) a decay in the metal temperatures of the superheater / reheater to the cold salt temperature. The rate of change in the metal temperatures can be controlled, in part, by adjusting the saturation temperature in the evaporator. During the transition, the goal is to provide steam flow rates that are high enough to 1) ensure a reasonably uniform flow distribution among the tubes, and 2) reduce the potential for damaging temperature and stress distributions in the superheater / reheater

- Turbine Synchronization to Isothermal Steam Production: The transition, although not a normal occurrence, is essentially a generator trip on reverse power. Although the turbine trips, the steam generator is not tripped, and the steam flow is diverted to the condenser. The attemperation pump is started, and temperature of the mixed salt at the inlet to the superheater is reduced consistent with the maximum allowable rate of temperature specified by the steam generator vendor. The process continues until the steam flow rate reaches the minimum allowable specified by the vendor; typically 16 percent. At this point, the mixed salt temperature at the hot ends of the superheater / reheater is necessarily greater than the cold salt temperature; i.e., the thermal energy available in a flow of (only) cold salt from the attemperation pump is not sufficient to support a steam flow rate of 16 percent

- Overnight Hold to Long Term Hold: The attemperation pump is stopped, the salt equipment is drained to the main storage tanks, the electric heat trace system is turned off, the water side of the equipment is drained, and nitrogen is supplied to the heat exchangers for corrosion control.

5. Risk Analysis - Organic Heat Transport Fluids

5.1 Introduction

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

As background to, and a starting point for, the risk assessment, an evaluation of the technical, commercial, and operating risks in commercial solar projects was recently conducted by NREL[2]. The SolarPACES CSP project database[3] was used to identify the projects that are currently in commercial operation, including 76 parabolic trough plants and 14 central receiver projects. Over the course of the project, the project team held about 50 information gathering sessions, representing nearly two-thirds of the plants operating worldwide.

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

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

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

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

[3] http://www.nrel.gov/csp/solarpaces/

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

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

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

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

A review of the table shows the following:

- The salt equipment topics with the highest weight values are the design and the reliability of the oil-to-salt heat exchangers. However, the risks are not at, or near, the top of the list. This is due to the following effects:
  - Only a limited number of the projects use thermal storage
  - The heat exchangers are hydraulically coupled to the outlet of the collector field, and to the outlet of the steam generator. Since the rates of temperature change at the outlet of the field, and at the outlet of the preheater, tend to be modest, the rates of temperature change seen at the oil-to-salt heat exchanger are also modest. In turn, modest rates of change generally result in a limited influence on the low cycle fatigue life of the heat exchanger

- In contrast to the hot salt tanks in central receiver projects, no failures have occurred in either the cold tanks or the hot tanks of parabolic trough projects. This can, in large part, be attributed to 1) the use of carbon steel for both the cold and the hot tanks, and 2) the tanks are partially isolated from daily thermal transients due to the damping effect of the oil-to-salt heat exchangers. A survival hypothesis for carbon steel tanks is discussed in detail in Section 4.3 of Volume 3 - Narrative.

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

5.2 Equipment Risks

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

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

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

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

Notes:

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

Mitigation or Response:

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

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

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

Mitigation or Response:

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

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

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

Mitigation or Response:

1. Replace the pump with the warehouse spare.

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

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

Notes:

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

Mitigation or Response:

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

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

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

Mitigation or Response:

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

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

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

Notes:

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

Mitigation or Response:

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

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

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

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

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

5.2.1 Thermal Storage Tanks

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

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

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

- The tank and the foundation must be cooled to essentially ambient temperature prior to personnel entering the tank and analyzing the failure. The cooling period is on the order of 7 to 10 days due to the large thermal mass of the foundation, and due to a requirement to control intra-tank temperature differentials to values low enough (< 30 °C) to prevent high internal stresses

- The source of the tank failure must be identified. This may include removal of tank sections, and laboratory examinations for items such as corrosion, low cycle fatigue fractures, or weld defects. The time required for laboratory examinations is on the order of one to perhaps a few weeks

- Repairs may involve a wide range of possibilities and conditions. Independent, potentially unrelated activities and associated schedules, include:

  - Procurement and delivery of replacement tank sections: At least several weeks
  - Removal of defective welds and rewelding: One to several weeks
  - Lifting the tank and replacing the foundation insulation: Two to several months
  - Post weld heat treatment of the welds in the thick metal sections: At least one week

- Once repairs are complete, the tank must be preheated prior to refilling with salt. The preheat period is on the order of 7 to 10 days.

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

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

Tank Failures

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

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

- All of the tanks operate at atmospheric pressure

- All of the tanks are supported on insulated foundations

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

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

- Residual welding stresses in the floor reach, and often exceed, yield values. The high stresses lead to plastic deformations of the plates, and the floor has, in essence, developed shallow buckles. The completed floor is a large thin sheet, and its continued resistance to buckling due to compression forces associated with radial temperature gradients, or due to friction loads, is generally low. When the tank is preheated, or when the tank is placed into commercial service, the residual welding stresses start to relax. The relaxation process likely occurs slowly, as the inventory temperatures are well below typical post weld heat treatment temperatures. However, the creep resistance of carbon steel is lower than the creep resistance of stainless steel. As such, the relaxation of the deformations in the floor of a carbon steel tank may be more pronounced than the relaxation of the deformations in the floor of a stainless steel tank. This, in turn, may make the floor of a carbon steel tank more resistant to buckling once commercial service begins

- In central receiver projects, the rate of temperature change in the downcomer can reach values as high as 360 °C/min. The hot salt tank will experience some fraction of this rate depending on 1) the depth of the inventory during the transient, 2) the cold tank/hot tank crossover temperature selected by the operator, and 3) the time required to move the downcomer diversion valves from open to close (or from close to open). The cold tank will also be involved in these transients. However, the level of the inventory in the cold tank, at least during morning startup, is usually much higher than the level of the inventory in the hot tank. This has the effect of damping transient effects in the cold tank. Also, the temperature of the salt entering the cold tank is always higher than the temperature of the inventory. As such, buoyancy effects carry the salt entering the tank away from the floor, which reduces the potential for local compressive forces on the floor

- In parabolic trough projects, the temperature difference between the hot side and the cold side is a nominal 100 °C. In contrast, the temperature difference in central receiver projects is on the order of 275 °C. The smaller temperature difference likely imposes smaller transient effects on the hot tank in a trough project than the hot tank in a central receiver project. The smaller transient effects result in lower transient stresses in the floor

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

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

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

$$Fe \rightarrow Fe^{++} + 2\ e^-$$

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

$$O_2 + 2\ H_2O + 4\ e^- \rightarrow 4\ OH^-$$

Further reactions can then occur, including:

$$Fe^{++} + 2\ OH^- \rightarrow Fe(OH)_2$$
$$2\ Fe(OH)_2 + H_2O + \tfrac{1}{2}\ O_2 = 2\ Fe(OH)_3$$

The hydrated oxides can lose water during dry periods, and revert to anhydrous oxides, such as ferrous oxide, FeO, and ferric oxide, Fe_2O_3 (hematite).

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

[4] Corrosion in Carbon Steels – IspatGuru

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

- The outside of the tank is exposed to nitrate salt. The source of the salt could be 1) vapor condensation from ullage gases passing through a vacuum / pressure relief valve, or 2) salt leakage from pumps, valves, instruments, or piping above the tanks

- Water comes into contact with the outside of the tank. The source of the water could be 1) site flooding during a particularly heavy rain, 2) rain water entering gaps in the tank insulation, or 3) spray water from pressure washers to remove residual salt

- As noted above, process conditions (i.e., hydrostatic loads) can produce stresses which are within the ASME allowable values but still high enough to initiate cracking. In addition, should water come into contact with the tank, local heat transfer will produce local temperature gradients, which, in turn, will generate local thermal stresses. Temperature gradients on the order of 45 °C can produce stresses equal to the ASME allowable value, and a gradient of approximately 115 °C can cause the material to yield.

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

Recommendations

<u>Low Cycle Fatigue</u>

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

- A confirmation that the salt tanks can survive the life of the project.

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

Stress Corrosion Cracking

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

- Ensure that no defects develop in the tank insulation, or in the aluminum jacket covering the insulation, that could expose the exterior of the tank to liquid water

- Remove salt from metal surfaces only by abrasive methods or by media blasting.

5.2.2 Oil-to-Salt Heat Exchanger

Operating Limits

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

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

Process Design

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

- During hold periods, supply cold salt to the cold end of the heat exchanger. Cold salt leaving the hot end of the heat exchanger is returned to the cold tank

- At the beginning of a charge cycle, blend cold Therminol from the discharge of the Therminol pumps with hot Therminol from the collector field at a Therminol mixing station at the hot end of the heat exchanger. This would control the rate of temperature change in the transition from Hold to Charge.

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

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

- The ability of the heat exchangers to accept some level of thermal shocks each day

- Adjusting the speed of the cold salt pump, or adjusting the speed of the hot salt pump, in a feedback loop to try and match the allowable rates of temperature change.

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

Consequences of Leaks

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

- The Therminol passes to the salt side of the heat exchanger, flashes to a vapor, and then collects in the upper portion of the shell. The vapor can restrict salt flow in this area, which can lead to 1) a reduction in the duty of the heat exchanger, or 2) an undesirable flow distribution which, in turn, leads to an undesirable temperature distribution and the development of intra-heat exchanger thermal stresses associated with self-restraint

- The Therminol passes to the salt side of the heat exchanger, flashes to a vapor, and is transported to the ullage space in one of the salt tanks. Over the course of a complete charge / discharge cycle, a portion of the ullage gas mixture of nitrogen and Therminol passes to the Therminol treatment system, where a portion of the Therminol is condensed for reuse. However, equilibrium considerations prevent the recovery of all of the Therminol, and some of the Therminol is lost to the environment through the vents in the treatment system. Replenishment of the Therminol represents an operating expense to the project

- When the leakage rate exceeds an acceptable value, the heat exchanger must be repaired. This involves draining the heat exchanger train, cooling the heat exchangers to ambient, identifying and plugging the tubes which are leaking, preheating the heat exchangers using Therminol from the collector field, and refilling with system with salt. The time required to do so is at least

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

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

Source of Leakage

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

- The friction (i.e., tube expansion) connections between the tube and the tubesheet can relax under daily thermal cycles. Some fraction of the stresses producing relaxation are due to the different thermal response times of the (thin) tubes and the (thick) tubesheet

- The strength welds between the ends of the tube and the face of the tubesheet can act as stress concentration points. Under transient conditions, the end of the welds and the face of the tubesheet will have different thermal response times, which can promote low cycle fatigue damage in the weld region.

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

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

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

Recommendations

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

- Limit the rate of temperature change to the vendor specification of 10 °C/min

- Limit the number and the magnitude of the daily thermal shocks to the vendor specifications of 1 and 60 °C, respectively

- Provide a uniform heat exchanger temperature of 295 °C during hold periods, which provides a consistent and a repeatable condition for daily startup and in response to lengthy cloud transients.

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

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

5.2.3 Salt Valves

Stem Packing Materials

A common commercial stem sealing material is graphite. The material is soft, which allows it to readily conform to the shape of the stem. Also, in the absence of an oxidizing environment, it is thermally stable. However, in salt service, the salt oxidizes the graphite to CO_2, and the packing eventually disappears.

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

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

Stem Leakage

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

Bellows Stem Seals

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

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

- A bellows stem seal is much more expensive than a packing stem seal. Budgetary estimates from a study conducted for DOE indicate that adding a bellows stem seal can triple the cost of a valve

- For isolation service, large globe valves are more expensive than triple offset butterfly valves. (As a technical aside, gate valves and ball valves are often used for isolation. However, nitrate salt oxides all metals, including Stellite. If a gate valve or a ball valve in salt service remains in the closed position for even one to two days, corrosion between the plug and the seat can render the valve permanently immovable. In general, the only types of valves suitable for salt service are ones in which the plug moves in a direction perpendicular to the seat; i.e., globe valves and triple offset butterfly valves.)

- The bellows may, under specific sets if circumstances, rupture. These circumstances include moving the valve with frozen salt in the bellows, or exposing the bellows to high transient pressures generated by a leak in a heat exchanger.

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

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

5.2.4 Heat Tracing

Heat tracing is an expensive item, and there is an economic incentive to install as little heating capacity as possible. A number of commercial projects have adopted, as a design criterion, a heat trace capacity sufficient to provide a maintenance temperature of 275 °C. However, any damage to, or degradation in, the insulation can lead to local freezing and an unscheduled shutdown to thaw the equipment.

Design Criteria for Commercial Projects

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

Recommendations

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

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

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

5.3 Project Risks

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

Table 5-9 Project Risks - Organic Fluids

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

Mitigation or Response:

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

5.3.1 Process Design

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

Process design elements include items such as the following:

- Ensure that the vendor requirements on rates of temperature change, thermal shock, minimum flow rate, and low cycle fatigue can be satisfied. The requirements are generally met through the selection of piping arrangements, pump specifications, valve configurations, valve specifications, control inputs, and control outputs

- Provide sufficient margin on the heat trace capacity to accommodate the inevitable degradation in heating capacity associated with failed cables and damage to the insulation

- Provide an approach to process control that avoids the use of instrument readings that are inaccurate or problematic (pressure, flow) and adopts the use of instrument readings that are accurate and repeatable (temperature, pump speed, valve position)

- Select equipment configurations (materials, fabrication techniques, thermal expansion allowances) that can tolerate some level of operator error

- Provide additional fatigue life for the equipment, above and beyond that required for the life of the project, to accommodate damage to the equipment during commissioning and the first year of operation when the control logic is being developed and refined. Identification of the transient conditions, and the determination of associated thermal stresses, will be difficult to predict. However, even a first-order analysis will often confirm whether a design should last for expected lifetime of the project.

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

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

5.3.2 Personnel Risks

As with process design, operating personnel are not the subject of an equipment specification. Nonetheless, some commercial projects have failed to meet their expected levels of availability due to mistakes made, and aggressive operating approaches taken, by the operating staff. The topic of personnel risks is included in this report, as the actions taken by an operator can have as large an influence on the plant availability as the process design features noted above.

Project Complexity

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

- The oil-to-salt heat exchangers are directly hydraulically coupled to the collector field. Further, in most projects, there is no mechanism for attemperating the temperature of the Therminol entering the heat exchanger (hot end or cold end), or the temperature of the salt entering the heat exchanger (hot end or cold end). As such, all of the transient conditions in the collector field can be imposed on the oil-to-salt heat exchangers

- The heat exchangers in the storage system are large equipment items. Further, the tube-to-tubesheet connections, which can number up to 1,500 in each of 6 series heat exchangers, typically rely on tube welding and tube rolling to seal the high pressure fluid from the low pressure fluid. During startup and shutdown, the thermal transient experienced by the collector field can result in rates of temperature change at or beyond the vendor limit, leading to high values of local thermal stresses, flexing of the tubesheet, relaxation of the tube-to-tubesheet connections, and internal leakage

- Therminol circulation in the collector field occurs whenever the collector field is not in operation; i.e., during overnight hold, during cloudy weather, or when wind speeds are high enough to stow the collector field. This ensures that the temperature of the Therminol in the field headers is nominally the same as the temperature of the Therminol in the heat collection elements. In this way, thermal shocks to the heat collection elements, which have the potential for permanently damaging the receiver tubes, can be avoided. The temperature of the Therminol can decay to values as low as perhaps 100 °C during non-operating periods. Similarly, during non-operating periods, Therminol circulation is maintained in the power block area. However, with salt present in the heat exchangers, the temperature of the Therminol must be maintained at values of at least 275 °C. As such, there are two circulation loops; one for the collector field, and a second for the thermal storage system. The two loops are separated by isolation valves. However, isolation valves do not always provide perfect isolation, and a situation can arise in which cold Therminol from the collector field loop mixes with warm Therminol in the thermal storage loop. Due to the large piping diameters in commercial projects, cold Therminol can flow along the bottom of the warm Therminol lines due to thermal stratification. The cold Therminol, on entering the oil-to-salt heat exchangers, can establish temperature gradients, which in turn can establish stress gradients, which are large enough to permanently damage the heat exchangers

- Heat trace cables can produce outer sheath temperatures in excess of 650 °C. (Design criteria for heat trace systems are discussed in Section 9.8.) On carbon steel equipment, inaccurate control over the duty cycles of cables can lead to graphitization of the steel. Graphitization is the separation of graphite from iron, and the subsequent formation of graphite nodules at the grain boundaries. The resulting structure is brittle, with a low strength and a high susceptibility to failure

- The salt has a nominal melting point of 230 °C. To prevent the salt from freezing, every surface in contact with salt must be electrically heat traced. As might be expected, heat trace systems are expensive, and there is a financial incentive to keep the heat trace capacity as small as possible. Under these conditions, the heat flux provided by the heater cables is on the order of 120 to 130 percent of the steady state heat losses from equipment. As a result, if the thermal insulation becomes moderately damaged, or if small (½ in.) gaps open in the thermal insulation, then the heat trace can no longer maintain the equipment above the salt melting point. The result is frozen salt and plugged equipment. Further, the heat tracing does not have the capacity to overcome heat losses which are higher than design values, and it is not possible to thaw the frozen equipment. To return the equipment to service, the insulation must be repaired. This is followed by a waiting period, which can last from several hours to several days, for the salt to thaw

- Flow instruments in salt service often show differences of 10 to 20 percent between the measured flow and the actual flow. The difference, in some cases, may be due to presence of water in the salt (produced by a leak in the steam generator) and cavitation downstream of the measuring device. Pressure instruments can have better accuracies, but the readings are very sensitive to the quality of the heat trace and the insulation on the instrument standoffs. Flow and pressure measurements are often inputs to the control logic of the distributed control system. But if the flow and pressure readings are inaccurate, then the process control outputs that determine pump speeds and valve positions can result in heat exchanger rates of temperature change that exceed vendor limits by significant margins. If the output signals from the distributed control system lead to oscillations, or lead to other conditions which might damage the equipment, the operators will switch to manual control. The skills of the operator can then have an influence on the fatigue life of the equipment.

Operator Duties

The control logic is initially developed for the plant in an as-new condition. For example, the heat exchangers are in a clean condition, the pump impellers have experienced no wear, the isolation valves seal properly, and the heat tracing on the pipe provides a uniform temperature over the length of the zone.

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

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

Staffing Approach

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

Recommendations

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

- Define the skills needed in each role. For example, an operator may need to have the background of a mechanical engineer, with a basic knowledge of heat transfer, stress analysis, and low cycle fatigue

- Determine the salary required to attract, for example, an engineer to a remote location

- Develop a staffing budget based on the required mix of personnel skills.

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

6. Risk Analysis - Inorganic Heat Transport Fluids

6.1 Introduction

As background to the risk analysis, and as discussed above in Section 5.1, an evaluation of the technical, commercial, and operating risks in commercial solar projects was recently conducted by NREL[6].

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

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

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

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

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

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

A review of the table shows the following:

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

6.2 Equipment Risks

As with the risk analysis for organic fluids, the risk analysis for inorganic fluids is based on an estimate of the daily probability of the event occurring, multiplied by the number of forced outage days to affect repairs or correct the problem. Potential mitigation or maintenance responses are included in the evaluation. The product of probability and outage is then multiplied by a factor of 330 to account for the number of operating days each year to calculate a value for the annual risk.

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

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

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

Notes:

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

Mitigation or Response:

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

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

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

Note:

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

Mitigation or Response:

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

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

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

Mitigation or Response:

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

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

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

Notes:

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

Mitigation or Response:

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

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

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

Mitigation or Response:

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

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

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

Note:

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

Mitigation or Response:

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

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

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

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

Note:

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

6.2.1 Leaks in the Hot Salt Tank

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

Excluding the projects in China, there are 5 hot salt tanks in commercial service: Gemasolar; Crescent Dunes; Cerro Dominador (2 tanks); and Noor III. The largest risk to the availability of a project is the low cycle fatigue life of the hot salt tank. All commercial projects, with the exception of the Cerro Dominador project in Chile, use a single hot tank. Further, all of the salt delivered by the solar field passes through the hot salt tank. Should the hot tank fail, there is no longer a source of thermal energy for the steam generator. As a result, the plant cannot generate electric power, and the project revenues drop to zero.

Consequences of a Leak

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

- The tank and the foundation must be cooled to essentially ambient temperature prior to personnel entering the tank and analyzing the failure. The cooling period is on the order of 7 to 10 days due to the large thermal mass of the foundation, and due to a requirement to control intra-tank temperature differentials to values low enough (< 30 °C) to prevent high internal stresses.

- The source of the tank failure must be identified. This may include removal of tank sections, and laboratory examinations for items such as corrosion, low cycle fatigue fractures, or weld defects. The time required for laboratory examinations is on the order of one to perhaps a few weeks.

- Repairs may involve a wide range of possibilities and conditions. Independent, potentially unrelated activities and associated schedules include:

  - Procurement and delivery of replacement tank sections: At least several weeks
  - Removal of defective welds and rewelding: One to several weeks
  - Lifting the tank and replacing the foundation insulation: Two to several months
  - Post weld heat treatment of the welds in the thick metal sections: At least one week

- Once repairs are complete, the tank must be preheated prior to refilling with salt. The preheat period is on the order of 7 to 10 days.

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

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

Failure Mechanisms

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

Recommendations

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

6.2.2 Leaks in the Steam Generator

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

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

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

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

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

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

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

Thermal Cycles

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

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

- An allowable rate of temperature change of 10 °C/min

- An allowable thermal shock (difference between the temperature of the fluid entering the heat exchanger and the metal temperature at the entrance to the heat exchanger) of 60 °C. Typically, one thermal shock per day is allowed.

- A number of startup cycles consistent with the life of the project, in combination with a safety factor. For example, a project life of 30 years implies a nominal cycle life of 30 years * 1 startup cycle per day = 10,000 cycles. To this is added a safety factor to account for difficulties in exactly predicting the low cycle fatigue damage associated with each startup cycle, and to provide a margin against a process design which does not exactly satisfy the vendor limits on rate of temperature change and thermal shock. A representative safety factor is in the range of 3 to 4.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Feedwater Chemistry

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

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

Hematite has a much lower solubility in water than magnetite, thus providing significantly better resistance to flow accelerated corrosion. Flow accelerated corrosion is caused by local removal of protective oxide formed on a steel surface by turbulent water flow in single-phase conditions, or by saturated steam flow in two-phase conditions. Loss of the protective oxide leads to rapid decrease in the thickness of the underlying component. To limit single-phase corrosion, an oxidizing feedwater chemistry is essential to ensure the formation of hematite, and thereby reduce the dissolution of magnetite. To limit two-phase corrosion, a pH increase across the entire Rankine cycle is required to push alkalinity into the liquid phase.

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

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

- When the leakage rate exceeds an acceptable value, the heat exchanger must be repaired. This involves draining the steam generator train, cooling the heat exchangers to ambient, identifying and plugging the tubes which are leaking, preheating the heat exchangers, and refilling with system with salt. The time required to do so is at least 1 week. The expense to the project, for both the repairs and the loss in revenue, is on the order of $1,000,000.

- The feedwater passes to the salt side of the heat exchanger, and is eventually lost to the environment through the vents in the storage tanks. This represents an increased demand on the water treatment system.

- During overnight hold, the water level in the drum must be replenished. If the leakage rate exceeds some threshold value, the temperature of the salt at the cold end of the preheater falls below a minimum value. To compensate, additional energy is supplied to the steam generator by 1) operating the electric water heaters normally used for startup, or 2) starting the hot salt pump, and adding a small quantity of hot salt to the flow of cold salt from the attemperation pump. Daily operation of the electric heaters can increase the annual auxiliary electric energy demand by 8 GWhe. Supplying hot salt to the steam generators during overnight hold results in less thermal energy available for electric power production.

- Prior to filling with salt, the steam generator is preheated using the electric water heaters. The combined capacity of the heaters is large (3 MWe), but it is not infinite. If the thermal demand for preheating, in combination with the thermal demand for preheating the makeup water to the drum, exceeds the capacity of the electric heaters, then it is no longer possible to raise the temperature of the heat exchangers to a safe fill temperature of, say, 275 °C. If so, the project may decide to thermally shock the heat exchangers to return the plant to commercial operation.

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

Dissolved Oxygen Control in the Rankine Cycle

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

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

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

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

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

Recommendations

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

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

6.2.3 Salt Pumps

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

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

Impedance Heating of Receiver Tubes

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

Heat Trace Heating of Field Piping

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

Schedule for Thawing the Collector Field

As discussed in Section 10.9.2, it is prohibitively expensive to provide a permanent impedance heating system for the collector field. As such, the project uses a number of portable impedance heating systems, that can be moved from one loop to the next as thawing progresses. Representative times to thaw all of the receivers in a commercial plant range from 15 to 30 days.

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

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

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

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

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

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

6.2.4 Salt Valves

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

6.2.5 Heat Tracing

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

6.3 Project Risks

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

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

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

Mitigation or Response:

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

6.3.1 Process Design

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

6.3.2 Selection of Operating Personnel

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

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

- The equipment on the hot side is stainless steel. Unlike carbon and ferritic steels, stainless steel is susceptible to a range of corrosion mechanisms, including intergranular stress corrosion cracking, stress relaxation cracking, and pitting corrosion.

- The difference in temperature between the cold side (295 °C) and the hot side (~520 °C) is a meaningful 225 °C. This difference is roughly 2½ times the difference in a parabolic trough project using organic heat transfer fluids. During steam generator startup and shutdown, inaccurate blending of cold salt with hot salt has the potential to thermally shock the equipment well beyond the capabilities of the equipment to withstand the thermal shock.

- The heat exchangers in the steam generator are large equipment items, with thick wall sections to withstand pressure as high as 145 bar. Further, the tube-to-tubesheet connections, which can number up to 3,000 in each of 2 steam generator trains, rely on a combination of friction (tube expansion) and welds (strength welds between the ends of the tubes and the face of the tubesheet) to seal the high pressure fluid from the low pressure fluid. During startup and shutdown, inaccurate blending of cold salt with hot salt can result in rates of temperature change well beyond the vendor limit, leading to high values of local thermal stresses, flexing of the tubesheet, relaxation of the tube-to-tubesheet connections, and internal leakage.

- During overnight shutdown of the Rankine cycle, the condenser is often opened to the atmosphere. The goal is to reduce the auxiliary energy consumption required to maintain condenser vacuum for several hours each day. Exposing the condenser hotwell to the atmosphere increases the dissolved oxygen in the condensate. Prior to daily startup, the dissolved oxygen and the pH values must be set to values required by the water treatment system. Any errors in setting the dissolved oxygen and pH levels can lead to long-term problems with flow accelerated corrosion, fouling of the heat transfer surfaces in both the feedwater heaters and the steam generator, and potentially pitting corrosion in the evaporator.

Recommendations

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

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

- During the period required to refine the logic in the distributed control system, provide guidance to the operators to protect equipment from low cycle fatigue damage during startup, shutdown, and transient conditions

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

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

7. Operating Requirements - Organic Heat Transport Fluids

7.1 Introduction

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

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

7.2 Oil-to-Salt Heat Exchangers

7.2.1 Requirements

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

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

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

7.2.2 Process Arrangement

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

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

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

- Salt from the hot salt tank is supplied to a salt steam generator.

Therminol Steam Generator

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

- Therminol from the collector field, at a temperature of 393 °C, is supplied directly to the steam generator. The Rankine cycle operates at the design values for the live steam pressure (100 bar), live steam temperature (379 °C), reheat steam temperature (379 °C), gross cycle efficiency (0.380), and gross generator output (100 percent)

- When the thermal output from the collector field exceeds the design duty of the steam generator, the additional energy is delivered to the storage system through the oil-to-salt heat exchanger

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

Salt Steam Generator

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

- When the collector field is in operation, Therminol from the collector field, at a temperature of 393 °C, is supplied to the oil-to-salt heat exchangers. Salt at the hot end of the heat exchanger, at a temperature of 388 °C, is supplied to the hot salt tank. Salt from the hot tank, in turn, is supplied to the salt steam generator. The Rankine cycle is designed to operate under full load conditions with a steam generator source temperature of 388 °C

- When the collector field is not in operation, salt from the hot tank is supplied to the steam generator. Since the steam generator source temperature remains constant at 388 °C, the Rankine cycle operates at the design rating and efficiency.

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

Commercial Selection

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

- The storage capacities in commercial projects are generally in the range of 6 hours. On an annual basis, the Rankine cycle can operate for approximately 50 percent more hours directly from the field rather than from the storage system. This, in turn, favors an approach which can exploit the higher Rankine cycle efficiency available with a steam generator source temperature of 393 °C

- With a Therminol steam generator, only that portion of the thermal energy from the field which exceeds the design rating of the steam generator needs to be directed to the storage system. This results in a smaller oil-to-salt heat exchanger than the case with a salt steam generator, in which all of the thermal energy from the collector field needs to be directed to the storage system

- Therminol steam generators have demonstrated much higher annual availabilities than salt steam generators, and reaching the availability targets used in the financial model is perhaps the most important contributor to a successful project.

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

7.2.3 Startup and Shutdown Procedures

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

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

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

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

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

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

7.2.4 Possible Revisions to the Operating Procedures

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

Morning Startup

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

- During morning startup, when the temperature of the Therminol from the collector field exceeds the temperature of the cold salt, a flow of Therminol is established at the hot end of the heat exchanger. The initial Therminol flow rate is equal to, or greater than, the minimum flow rate defined by the vendor

- As morning startup progresses, the temperature of the Therminol from the field increases. This, in turn, results in an increase in the temperature of the salt at the hot end of the heat exchanger. If the temperature of the salt is less than the temperature of the hot tank, the salt is recirculated back to the cold tank

- When the temperature of the salt at the hot end of the heat exchanger is within 5 °C of the salt temperature in the hot tank, the salt flow is directed to the hot tank, and the recirculation flow to the cold tank is stopped. The heat exchanger is now in normal operation, and the flow of cold salt is adjusted to provide the desired approach temperature at the hot end of the heat exchanger.

Transition from Charge to Discharge

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

- The design longitudinal temperature profile has been established along the heating length

- The salt flow rate and the Therminol flow rate are at values no lower than the minimum allowable values specified by the vendor.

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

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

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

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

- When the cold salt pump is tripped, the coast down time of the pump should match the closing period for the Therminol isolation valve

- When the hot salt pump is started, the acceleration time for the pump should match the opening period for the Therminol isolation valve

- When the hot salt pump is started, the speed of the pump, and the percent open of the Therminol isolation valve, should be pre-selected values.

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

- Remove the longitudinal temperature profile by reducing the flow of Therminol to the hot end of the heat exchanger, and simultaneously increasing the flow of cold salt to the hot end of the heat exchanger. The two flow rates would be set to decrease the metal temperature at the hot end of the heat exchanger by the allowable rate set by the vendor (a nominal 10 °C/min). Relatively low temperature salt leaving the hot end of the heat exchanger would be recirculated back to the cold tank. When the flow of Therminol reaches the minimum value set by the vendor, and with the cold salt pump operating at the design speed, the approach temperature at the hot end of the heat exchanger would be a modest 5 °C or so. At this point, the flow of Therminol can be stopped, and the heat exchanger will assume a uniform temperature equal to the cold salt temperature. The speed of the cold salt pump is reduced, and pump is then tripped

- A flow of cold Therminol is established at the cold end of the heat exchanger. The temperature of the Therminol is nominally the same as the temperature of the cold salt; as such, no heat transfer offers. The flow rate of Therminol is the same order of magnitude as the design flow rate. Therminol from the hot end of the heat exchanger is recirculated back to the suction side of the Therminol pumps

- A flow of cold salt is established at the hot end of the heat exchanger by means of a bypass line. Salt leaving the cold end of the heat exchanger flows to the cold tank. The flow rate of cold salt is at least equal to the minimum allowable flow rate specified by the vendor

- The hot salt pump is started, with a speed which develops a head that is somewhat less than the cold salt pressure at the hot end of the heat exchanger. The pump is protected from overheating by the flow through the minimum flow recirculation valve

- The speed of the hot salt pump is increased to the point where the discharge pressure equals the cold salt pressure. The discharge check valve on the pump opens, and hot salt starts to mix with cold salt at the hot end of the heat exchanger

- The speed of the hot salt pump is increased at a rate which results in an increase in the mixed salt temperature at a rate equal to the allowable rate for the heat exchanger. The speed of the cold salt pump remains constant. As such, the flow rate on the salt side is always at least equal to the minimum allowable flow rate

- When the flow rate of hot salt is nominally equal to the flow rate of cold salt, the speed of the cold salt pump is reduced at a rate which increases the mixed salt temperature at the hot end of the heat exchanger at the allowable rate of temperature change specified by the vendor. When the flow rate of cold salt reaches a value of zero, the cold salt pump is tripped

- The flow rate of Therminol is adjusted to provide the desired approach temperature at the cold end of the heat exchanger. At this point, the normal longitudinal temperature profile in the heat exchanger has been reestablished.

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

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

Evening Shutdown

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

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

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

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

- During part load conditions, the steam turbine normally operates under sliding pressure conditions to improve the expansion efficiency. However, shortly before the turbine is tripped, the operating mode can be switched from sliding pressure to constant pressure at the design value of 100 bar. This has the effect of increasing the temperature of the Therminol at the cold end of the oil-to-salt heat exchanger

- The turbine is tripped at load of perhaps 10 percent. Based on GateCycle models of the Rankine cycle part-load performance in a representative commercial project, the temperature of the Therminol at a load of 10 percent is on the order of 265 °C, as noted in Figure 7-1. A Therminol temperature of 265 °C is about 30 °C below the design value. However, given an approach temperature of perhaps 5 °C at the cold end of the oil-to-salt heat exchanger, the salt temperature at the cold end of the heat exchanger remains in an acceptable range

- After the turbine is tripped, the steam generator remains in operation, sending steam to the condenser through the bypass system. Auxiliary steam is supplied to the deaerator to provide a level of feedwater heating. Therminol, at a temperature of 260 to 265 °C, returns to the oil-to-salt heat exchanger

- The cold salt pump is started, with a speed which develops a head that is somewhat less than the hot salt pressure at the hot end of the heat exchanger. The pump is protected from overheating by the flow through the minimum flow recirculation valve

- The speed of the cold salt pump is increased to the point where the discharge pressure equals the hot salt pressure. The discharge check valve on the pump opens, and cold salt starts to flow in the salt bypass line around the heat exchanger. The flow of cold salt start to mix with the flow of hot salt at the hot end of the heat exchanger

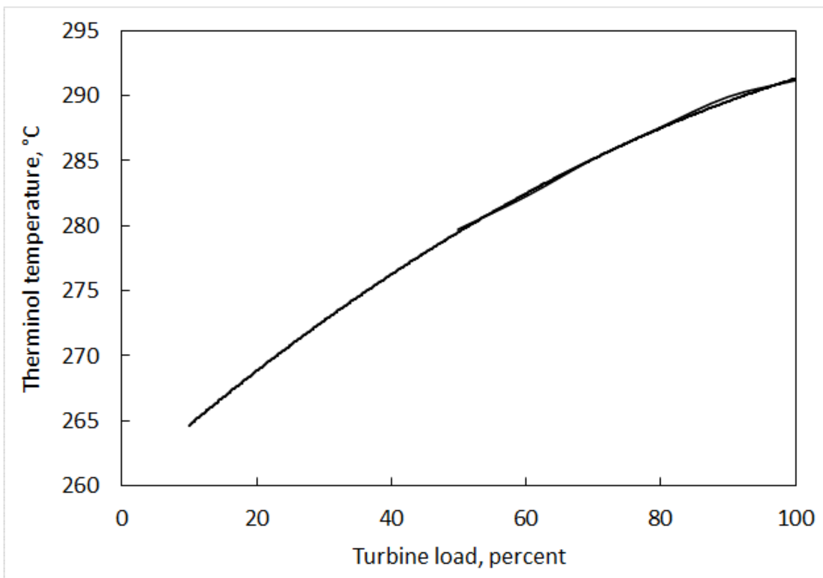

Figure 7-1 Therminol Temperature as a Function of the Turbine Load

- The speed of the cold salt pump is increased at a rate which results in an decrease in the mixed salt temperature at a rate equal to the allowable rate for the heat exchanger. The speed of the hot salt pump remains constant. As such, the flow rate on the salt side is always at least equal to the minimum allowable flow rate

- As the temperature of the mixed salt at the hot end of the heat exchanger decreases, the temperature of the Therminol returning to the steam generator also decreases. However, if the saturation pressure in the evaporator is fixed at 100 bar by the turbine bypass system, then the temperature of the Therminol at the cold end of the steam generator either remains constant or increases slightly; i.e., the Therminol heat transfer line pivots about the pinch point

- The speed of the cold salt pump is increased to the design value. At some point in the transition, the discharge pressure of the hot salt pump falls below the discharge pressure of the cold salt pump, and the check valve in the discharge line from the hot pump closes. The pump is protected from overheating by the flow through the minimum flow recirculation valve. The hot salt pump is then tripped

- At this point, only cold salt is flowing through the heat exchanger. However, the flow of cold salt is near the design value, while the flow of Therminol is near the minimum allowable value. With a Therminol temperature of approximately 265 °C at the cold end of the heat exchanger, little heat transfer occurs, and the entire exchanger soon approaches a uniform temperature which is slightly less than the temperature of the cold salt

- The flow of Therminol is stopped by closing the isolation valve. The cold salt pump is stopped, the positions of the salt isolation valves are changed, and the cold salt pump is restarted. This establishes a flow of salt to the cold end of the heat exchanger. The speed of the cold salt pump is set to the overnight hold value. Cold salt from the hot end of the heat exchanger returns to the cold tank through the heat exchanger bypass line.

7.3 Thermal Storage System

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

7.3.1 Performance Requirements

Cold Salt Tank

The cold salt tank performs 3 primary functions:

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

- During a storage charge cycle, supplies salt, at a nominal temperature of 293 °C, to the cold end of the oil-to-salt heat exchanger

- During a storage discharge cycle, receives salt, at a nominal temperature of 293 °C, from the cold end of the oil-to-salt heat exchanger.

The normal operating temperature of the tank is the temperature of the salt at the cold end of the oil-to-salt heat exchanger (295 to 305 °C, depending on the approach temperatures selected for the heat exchanger). However, the design temperature of the tank is typically selected to be 350 °C. This is a representative value for the temperature of the salt added to the tank during the salt melting process.

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/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 3 primary functions:

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

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

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/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

Distribution Ring Header

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

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

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

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

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

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

- The flows on both the oil side and the salt side are reduced to the minimum, and the heat exchanger is tripped. The goal is to preserve the longitudinal temperature profile for operation the next day. Since the salt flow is tripped, there is essentially no effect on the inventory temperature in the cold tank.

- Alternately, the temperature of the entire heat exchanger can be reduced to the cold salt temperature by adding a flow of cold oil to the flow of hot oil supplied to the hot end of the heat exchanger. Since the change in the metal temperature at the hot end of the heat exchanger is about 100 °C, the time required to cool the heat exchanger is on the order of 100 °C / 10 °C/min = 10 min. The effect on the inventory temperature in the cold tank will be low because 1) the salt flow rate is at the maximum turndown ratio for the heat exchanger, and 2) the cold tank will be full, or close to full, at the end of a discharge cycle.

Ring Header Supports

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

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

Alternately, in other projects, the supports are placed on, but are not welded to, the floor. 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

- Introduce salt at locations inside the roof or near the top of the wall. This helps to isolate receiver startup transients from the floor, and eliminates the potential of the sharing of forces between the header supports and the floor. The ring header would be replaced with an array of spray nozzles to avoid problems with trying to support the ring header from the roof or from the top of the wall. A note: This approach to flow distribution in the hot tank was demonstrated at the Solar Two project.

- Clearly, introducing salt at the top of the inventory will produce a vertical stratification in the inventory due to 1) a permanent layer of stagnant salt located below the elevation of the suction bells for the salt pumps, and 2) continuous transfer of heat into the foundation. To prevent or to remove the stratification, some form of bulk mixing will be required. Candidates methods might include the following:
  - Pump recirculation, which supply salt to a grid of injection points near the floor
  - Gas compressors, which draw suction from the ullage space in the tank and supply the ullage gas to a grid of injection points near the floor
  - Tank-scale mixing devices, such as a large vertical plate rotating about a vertical axis in the tank, or a large horizontal perforated plate, which vertically traverses the inventory.

7.3.2 Equipment Design

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

At projects sites representative of commercial projects, the allowable soil bearing load is on the order of 240 kPa (5.000 lb_f/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 carbon steel used in both the cold tank and the cold salt tank, a post weld heat treatment is required by the Section VIII of the Code for section thicknesses greater than 38 mm (1.5 in.). Specialty contractors can provide heat treatment services, but the process is not without risks:

- The treatment temperature is high enough (650 °C) that the strength of the material decreases to the point where creep deformations are possible

- The iron in the steel can chemically reduce the carbon dioxide in the air. Carbon is infused into the surface of the metal, which increases the strength, but reduces, the ductility of the material.

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

8. Operating Requirements - Inorganic Heat Transport Fluids

8.1 Introduction

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

- Selection of design parameters, such as temperatures, flows, allowable fluxes, rates of temperature change, low cycle fatigue lives, and design Codes

- Evaluation of (Capital cost) versus (Capital cost + Operating cost). The evaluation considers items such as material selection, fabrication techniques, and installed redundancy

- Outline of hardware recommendations and specifications.

8.2 Collector Field

8.2.1 Performance Requirements

Temperatures

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

Collector Field Outlet Temperature

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

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

- 5.75 m collector aperture width

- 0.75 collector optical efficiency

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

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

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

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

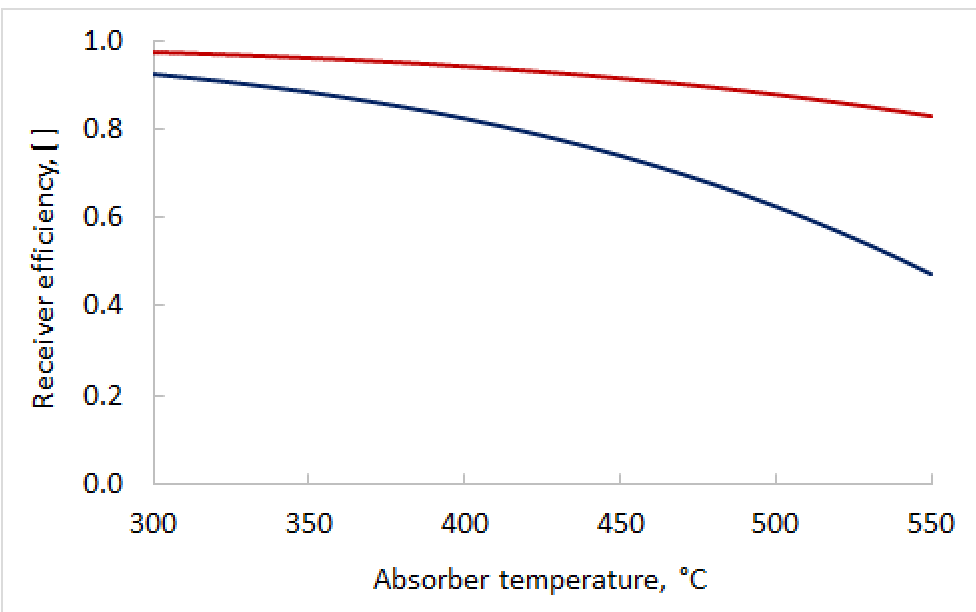

Figure 8-1 Receiver Efficiency as a Function of Absorber Temperature

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

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

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

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

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

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

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

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

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

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

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

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

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

<u>Collector Field 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 540 °C, a representative live and reheat steam temperature is 515 °C. These temperatures, in turn, result in live steam pressures of 120 bar to a maximum of perhaps 130 bar

- The efficiency of the Rankine cycle is improved by increasing the final feedwater temperature. However, raising the final feedwater temperature increases the salt temperature at the cold end of the preheater in the steam generator, which, in turn, reduces the temperature difference between the cold salt tank and the hot salt tank. Optimum final feedwater temperatures are typically in the range of 220 to 240 °C.

- To provide a heat transfer area in the evaporator of the steam generator that can be safely warranted by the vendor, a pinch point of 5 °C is typically selected

- To help prevent water evaporation (steaming) at the hot end of the preheater during low load operation, the preheater is typically designed with a 5 °C approach to saturation.

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

Pressures

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

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

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

Flow Velocities

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

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

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

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

Turn Down Ratio

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

Thermal Efficiency

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

- Direct normal radiation (300 to 1,065 W/m^2)

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

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

- 300 °C absorber temperature
- 1,065 W/m^2 aperture normal radiation
- 0 m/sec wind speed.

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

Fatigue Life

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

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

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

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

8.2.2 Equipment Design

Materials

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

Redundancy

The length of a collector loop is based on thermodynamic properties of the working fluid, the temperature rise across the loop, and limits on minimum and maximum flow velocities. In commercial projects, the product of the aperture width times the loop length is on the order of 4,000 to 6,000 m^2.

The total aperture area in a commercial project is a function of the plant rating, the capacity of the storage system, and the site weather characteristics. Representative aperture areas can range from 300,000 m^2 to more than 1,000,000 m^2. As such, the number of loops can range from 75 to several hundred.

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

8.3 Thermal Storage System

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

8.3.1 Performance Requirements

Cold Salt Tank

The cold salt tank performs 5 primary functions:

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

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

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

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/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 (540 °C). However, the design temperature of the tank is typically selected to be the high-high trip temperature for the receiver (600 °C).

The exterior of the tank is provided with thermal insulation to limit the thermal losses to values in the range of 80 to 120 W/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:

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

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

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

Alternately, in other projects, the supports are placed on, but are not welded to, the floor. 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

- Introduce salt at locations inside the roof or near the top of the wall. This helps to isolate receiver startup transients from the floor, and eliminates the potential of the sharing of forces between the header supports and the floor. The ring header would be replaced with an array of spray nozzles to avoid problems with trying to support the ring header from the roof or from the top of the wall. A note: This approach to flow distribution in the hot tank was demonstrated at the Solar Two project.

- Clearly, introducing salt at the top of the inventory will produce a vertical stratification in the inventory due to 1) a permanent layer of stagnant salt located below the elevation of the suction bells for the salt pumps, and 2) continuous transfer of heat into the foundation. To prevent or to remove the stratification, some form of bulk mixing will be required. Candidates methods might include the following:
  - Pump recirculation, which supply salt to a grid of injection points near the floor
  - Gas compressors, which draw suction from the ullage space in the tank and supply the ullage gas to a grid of injection points near the floor
  - Tank-scale mixing devices, such as a large vertical plate rotating about a vertical axis in the tank, or a large horizontal perforated plate, which vertically traverses the inventory.

8.3.2 Equipment Design

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

At projects sites representative of commercial projects, the allowable soil bearing load is on the order of 240 kPa (5.000 lb_f/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 ASME Boiler and Pressure Vessel Code. For the stainless steel used in the hot salt tank, there are no fabrication restrictions noted in the Code. However, for the carbon steel used in the cold salt tank, a post weld heat treatment is required for section thicknesses greater than 38 mm (1.5 in.). Specialty contractors can provide heat treatment services, but the process is not without risks:

- The treatment temperature is high enough (650 °C) that the strength of the material decreases to the point where creep deformations are possible

- The iron in the steel can chemically reduce the carbon dioxide in the air. Carbon is infused into the surface of the metal, which increases the strength, but reduces, the ductility of the material.

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

8.4 Steam Generation System

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

8.4.1 Process Requirements

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

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

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

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

Daily Startup and Shutdown

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

- During overnight hold, cold salt from the attemperation pump flows through each of the heat exchangers. The cold salt is returned to the cold tank

- In the first phase of startup, hot salt is mixed with cold salt at a point upstream of the hot end of the evaporator. This establishes a flow of saturated steam in the superheater and in the reheater equal to about 20 minimum of the design flow rate. The flow rate is slightly greater than the minimum allowable flow rate specified by the vendor (16 percent). No heat transfer occurs in either the superheater or the reheater during this phase

- In the second phase of startup, the relative flow rates of hot salt and cold salt remain fixed, and the salt mixing station is moved from a point upstream of the hot end of the evaporator to the normal point upstream of the hot ends of the superheater / reheater. Superheating now occurs in the superheater and in the reheater. Due to the superheating duties, the steam flow rate decreases to 16 percent

- The relative proportions of hot salt and cold salt are adjusted by changing the relative speeds of the hot salt pump and the attemperation pump. When the flow of cold salt has reached a value of zero, the steam generator is in normal operation

- The process is essentially reversed to shut down the steam generator.

Minimum Flow Rate

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

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

- A preheater recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the preheater. The combination of the feedwater flow rate plus the recirculation flow rate always satisfies the vendor requirement

- An evaporator recirculation pump draws suction from the steam drum, and supplies saturated water to a mixing station at the cold end of the evaporator. The combination of the feedwater flow rate plus the recirculation flow rate can be used to satisfy the vendor requirement

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

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

Maximum Allowable Rate of Temperature Change

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

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

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

8.4.2 Performance Requirements

The steam generator must meet the following performance requirements:

- The saturated steam at the turbine control valves complies with the steam quality standards set by the turbine vendor

- The heat exchanger designs meet the Tubular Equipment Manufacturers Association requirements for items including flow distribution, tube vibration, and tubesheet temperature gradients, and allowances for spare tubes

- The heat exchangers have a fatigue life consistent with at least 10,000 startup and shutdown cycles (i.e., 30 years * 330 cycles per year).

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

8.4.3 Equipment Design

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

- The evaporator uses natural or forced recirculation, in combination with a separate steam drum. A solar project necessarily cycles each day, and the condensate water chemistry in the condenser hotwell or condensate tank typically degrades during the overnight hold period. Reestablishing the water chemistry during morning startup for a drum-type evaporator is less complex and less time consuming than reestablishing the more stringent water chemistry required for a single-pass-to-superheat steam generator

- A preheat heat exchanger, separate from and upstream of the water side of the evaporator, performs two functions:
    - Raises the temperature of the feedwater close to the saturation point, which allows the evaporator to operate under isothermal conditions on the water side

    - 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.

9. Functional / Prescriptive Specifications - Organic Heat Transport Fluids

9.1 Introduction

Plant and equipment specifications fall into two broad categories:

- Functional specifications describe what the system or the equipment needs to do, consistent with the minimum legal requirements of the local jurisdictions. The details of how this is to be accomplished is developed by the engineering contractor. This allows the engineering contractor to define the process and the equipment designs such that the functional requirements can be met at the lowest cost

- Prescriptive specifications, which are developed by the Owner, prescribe to the engineering contractor how the functional requirements of the project are to be met. Prescriptive elements include items such as the process design, Code Section selection, fabrication techniques, materials selection, and component details. This arrangement ensures that the favorable experience from a previous project is repeated, and helps to avoid situations in which an inexperienced contractor repeats mistakes from earlier projects.

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

- The Owner must have a thorough understanding of the process and the technical features of the plant. For example, the heat trace designs in commercial projects often use 1 dual-element thermocouple in each zone as the input to the temperature set point controller. However, a heat trace design, which reflects the experience from previous projects, might use 2 to 4 thermocouples, distributed along the length of the zone, as inputs to the controller. This arrangement can help to detect defects in the insulation, local areas with temperatures below the freezing point of the salt, local areas which are receiving too much heat input from adjacent zones, and leakage past isolation valves.

- The Owner is prepared to accept what is likely to be an increase in the capital cost as a means of achieving the goals set in the performance and in the financial models. For example, the Owner may require the use of all-welded construction in salt heat exchangers. This fabrication approach is more expensive than the conventional approach; i.e., tube rolling followed by strength welding of the tube to the tubesheet. However, an all-welded design is more tolerant of the transient stresses associated with 1) daily startup and shutdown, and 2) mistakes made by the operators. In turn, the improvement in plant availability more than compensates for the marginal increase in the cost of the heat exchangers due to an all-welded design.

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

9.2 Design Codes and Standards

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

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

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

9.3 Nitrate Salt Specification

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

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

- Binary nitrate salt (solar salt) mixture, which is 60 weight percent sodium nitrate ($NaNO_3$) and 40 weight percent potassium nitrate (KNO_3)

- Ternary nitrate salt mixture, which, with contributions of 30 weight percent lithium nitrate ($LiNO_3$), 18 weight percent sodium nitrate ($NaNO_3$), and 52 weight percent potassium nitrate (KNO_3), offers the lowest melting point.

- Ternary nitrate/nitrite salt mixture, the most common of which is Hitec®, with contributions of 7 weight percent sodium nitrate ($NaNO_3$), 40 weight percent sodium nitrite ($NaNO_2$), and 53 weight percent potassium nitrate (KNO_3)

- A ternary nitrate salt mixture, Hitec XL®, with contributions of 42 weight percent calcium nitrate ($Ca(NO_3)_2$), 15 weight percent sodium nitrate ($NaNO_3$), and 43 weight percent potassium nitrate (KNO_3)

- A quaternary nitrate salt mixture, reviewed by Sandia National Laboratories, with contributions of 22 weight percent lithium nitrate ($LiNO_3$), 17 weight percent calcium nitrate ($Ca(NO_3)_2$), 18 weight percent by weight sodium nitrate ($NaNO_3$), and 43 weight percent potassium nitrate (KNO_3)

- A quinary nitrate/nitrite salt mixture, patented by Sandia National Laboratories, with contributions of 13 weight percent lithium nitrate ($LiNO_3$), 10 weight percent by weight sodium nitrate ($NaNO_3$), 12 weight percent potassium nitrate (KNO_3), 29 weight percent sodium nitrite ($NaNO_2$), and 36 weight percent potassium nitrite (KNO_2).

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

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

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

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

9.3.1 Grades

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

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

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

chlorine forms at the surface of the metal in contact with the salt. Through diffusion, chlorine congregates at the boundary between the inside of the corrosion scale and the parent metal. The chlorine reacts with the metallic elements, forming metal chlorides. The metal chlorides, which have high vapor pressures, evaporate and diffuse back out through the corrosion scale layer [9]. The metal chlorides, in contact with the salt, are oxidized to various metal oxides. This releases the chlorine to repeat the corrosion process. The chlorides act, in effect, as catalysts to accelerate the oxidation rates of the parent metal. In the paper by Grabke, the process was explained specifically in reference to iron based alloys, where various oxides, such as hematite (Fe_2O_3) or magnetite (Fe_3O_4), are formed.

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

A typical technical grade specification should include the following items:

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

9.3.2 Total Chloride Concentration

An example of the calculation of the total chloride ion concentration in a mixture is as follows: If the chloride ion concentrations in the sodium nitrate and potassium nitrate are 0.85 and 0.20 weight percent, respectively, then the chloride ion concentration in the mixture is (0.6 weight fraction $NaNO_3$)*(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:

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

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

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

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

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.

9.3.3 Allowable Contaminants

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

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

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

9.3.4 Supply Options

The salt may be supplied in one of two forms:

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

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

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

$$Mg(NO_3)_2 \Rightarrow MgO + 2\ NO_2 + ½\ O_2$$

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.

9.3.5 Deviations from a 60/40 Mixture

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

Other properties, specifically the density, can be estimated by adjusting the 60/40 properties based on molar volumes, as described by Bradshaw [10].

9.4 Nitrate Salt Handling and Melting Specification

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

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

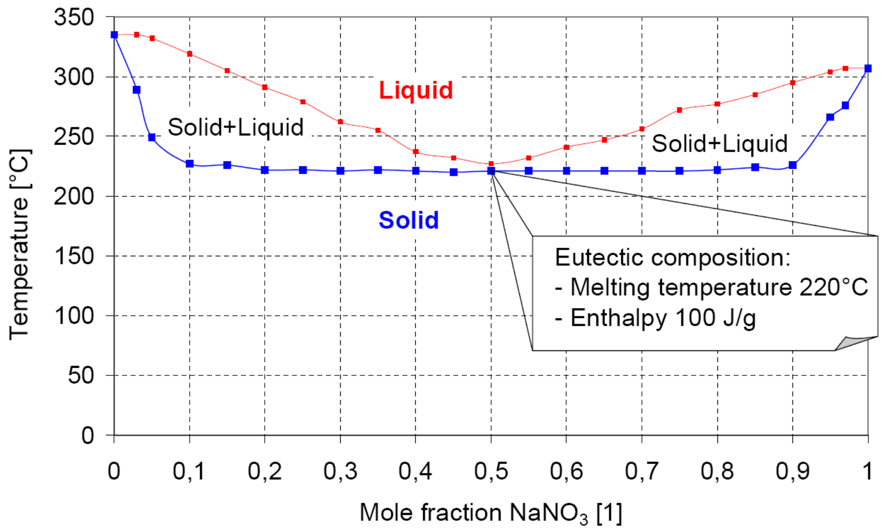

Figure 9-1 Melting Point of Binary Nitrate Salt Mixtures

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

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

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

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.

In a trough project, the salt, after melting, is loaded into the storage tanks. However, the temperature of the salt is perhaps in the range of 275 to 350 °C, depending on the capacity of the tank electric heaters. As such, the magnesium decomposition rate is much slower than would occur at a temperature of 550 °C; certainly on the time span of months. Further, the water that is adsorbed by the salt in transit and in storage does not leave the salt during, and immediately after, the melting process. Anecdotal evidence suggests that the time required to release the water from the salt is measured in weeks. As such, within the liquid inventory of the salt, 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 in the form of a gas. Further, the gas can condense on any surface with temperatures below about 80 to 90 °C. Since nitric acid has a dew point higher than water vapor, any warm surface on which the vapor condenses will essentially be pure acid. Nitric acid samples collected from a commercial project during commissioning showed pH values in the range of 1 to 2.

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

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

- To prevent damage to the carbon steel equipment which is exposed to the ullage gas in the storage tanks, the equipment must be maintained at a temperature of at least 250 °C for a period of several months after the salt is loaded into the storage tanks. A temperature of 250 °C prevents the condensation of both nitric acid and salt vapors.

- If the equipment cannot be maintained at a temperature of 250 °C, then the equipment must be fabricated from a material which is resistant to nitric acid corrosion, such as stainless steel.

9.5 General Salt System Equipment

9.5.1 Piping

Design Codes

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

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

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

Materials

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

Connections, Sizes, and Schedules

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

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

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

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

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

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

Flanges

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

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

Fittings

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

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

Pipe Supports, Anchors, and Guides

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

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

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

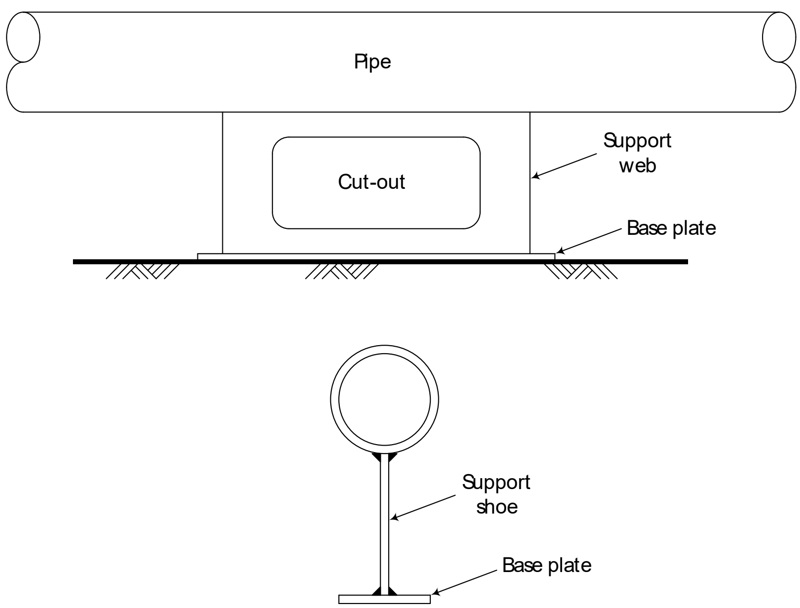

Figure 9-2 Pipe Support, Anchor, and Guide

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

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

9.5.2 Control and Isolation Valves

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

Process Design

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

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

Extended Bonnets

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

Valve Types

A salt corrosion layer develops on the plugs and the seats of all valves. For gate and ball valves, the two sealing surfaces slide across one another. When the corrosion layers develop, particularly with the valves in the closed position, the corrosion layers can lead to binding and erratic valve motion. As such, gate and ball valves are not suitable for salt service. For globe and offset butterfly valves, the plugs move perpendicular to the seats, and the development of the corrosion layers are less likely to interfere with the motion of the valve.

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

Stem Seals

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

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

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

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

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

- Fiberglass-filled Teflon® washers.

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

Bonnet Gaskets

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

9.5.3 Check Valves

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

- Should a pump trip, the check valves prevents a reverse flow through the pump, and protects against an overspeed condition in reverse rotation

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

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

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

9.6 Oil to Salt Heat Exchanger

9.6.1 Equipment Design

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

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

9.6.2 Process Design

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

- During Hold periods, cold salt is supplied to the cold end of the heat exchanger. The flow of Therminol through the heat exchanger is suspended to reduce the potential for cold Therminol from the collector field freezing salt in the heat exchanger. As such, no heat transfer occurs, and the salt leaving the hot end of the heat exchanger is returned to the cold tank through a recirculation line.

- Included in the design is a Therminol mixing at the hot end of the heat exchanger. At the beginning of a charge cycle, cold Therminol from the discharge of the Therminol pumps is mixed with hot Therminol from the collector field at the mixing station. The blending process controls the rate of temperature increase in the heat exchanger during the transition from Hold to Charge, and reduces the magnitude of any thermal shocks.

- Also included in the design is a salt mixing station at the hot end of the heat exchanger. At the end of a Discharge period, cold salt from the cold salt pump is mixed with hot salt from the hot salt pump at the mixing station. The blending process controls the rate of temperature decrease in the heat exchanger during the transition from Discharge to Hold.

- During the Hold period, the flow rate of salt is set to a value which is equal to, or greater than, the minimum flow rate specified by the vendor. During normal operation, the heat exchanger is always operated at a combination of salt flow rate and Therminol flow rate that satisfies the vendor flow rate requirement. As a consequence, the minimum flow rate criterion on the salt side is automatically satisfied both during the Hold period, and during the transition from Discharge to Hold.

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

9.7 Thermal Storage Tank Weld Examinations

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

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

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

To improve the calculated fatigue life of the floor, the reduction factors should be 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.

9.8 Heat Tracing

9.8.1 System Description

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

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

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

9.8.2 Scope of Supply

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

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

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

9.8.3 Electric Heat Tracing System Design Basis

Zone Definitions

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

In general, zone lengths for piping should be as long 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 9-7 Pressure Transmitter in a Vertical Process Line, and in Figure 9-8 Pressure Transmitter in a Horizontal Process Line.

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

If different operating modes subject a pipe or a vessel to different static fluid heights, then the boundary between the zones must end above the liquid level. For example, recirculation from a salt pump at a demonstration solar project produced a static height of about 9 m in a riser. If the heat trace zone in the lower section of the riser had ended below an elevation of 9 m, then the stagnant salt above the top of the zone would have frozen in a matter of a few hours. Further, if the zone spans a region which is not exposed to salt at the top, but is flooded below, then a heat transfer analysis must be conducted to confirm that the temperature in the upper portion of the zone does not exceed safe levels.

Zone Thermal Capacities

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

- An economic analysis, comparing the capital cost of the heat trace capacity with the system startup time. One example is a salt line, which is normally drained at the end of an operating day. Prior to startup the following day, the temperature of the piping must be increased to the salt fill temperature. If the line has a high heat trace capacity, then the preheat time will be short, the preheat energy will be low, but the capital cost will be high. (The reverse also applies.) As such, there is an optimum heat trace capacity that minimizes the sum of (the equivalent capital cost of the preheat energy) and (the capital cost of the heat tracing).

- An Owner's assessment of the degree to which the heat trace capacity must accommodate an expected degradation in the cable heating capacity or an increase in the heat losses through the equipment insulation

- An Owner's assessment of the reliability of the heat trace cables, and the requirement for spare installed cables.

The nominal set of design parameters include the following:

- Empty salt pipes, empty valve bodies, and valve bonnets can be preheated from ambient temperature to 275 °C in 12 hours

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

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

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

Control and Set Point Temperatures

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

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

Component Redundancy

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

- 100 percent for cables on the piping from the heat exchangers. This segment of piping operates under daily thermal cycles, and the failure rates of the heat trace cables are expected to be 'moderate to high'

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

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

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

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

Component Requirements

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

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

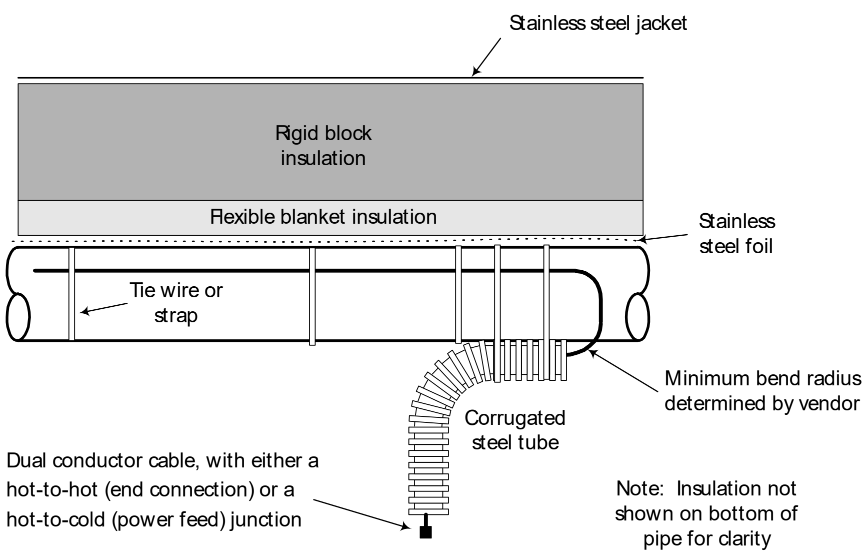

Figure 9-3 Heat Trace Arrangement on Pipe at End of Zone

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

Valve Zones

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

- The unit mass, in kg/m, of a valve body is much higher than the unit mass of the adjacent pipe. As such, if a time is specified to preheat the piping system to 275 °C, the heat input per meter for a valve body must be much higher than the heat input per meter for the pipe

- The type of insulation on a valve body is often different from the type of insulation on the adjacent pipe. As such, under steady state conditions, the heat loss per meter of valve body will be different than the heat loss per meter of the pipe

- The unit heat losses, in W/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. The upper value ensures that Teflon or other organic materials in the stem packing, if used, do not decompose.

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

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

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

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

Temperature Sensor Installation

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

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

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

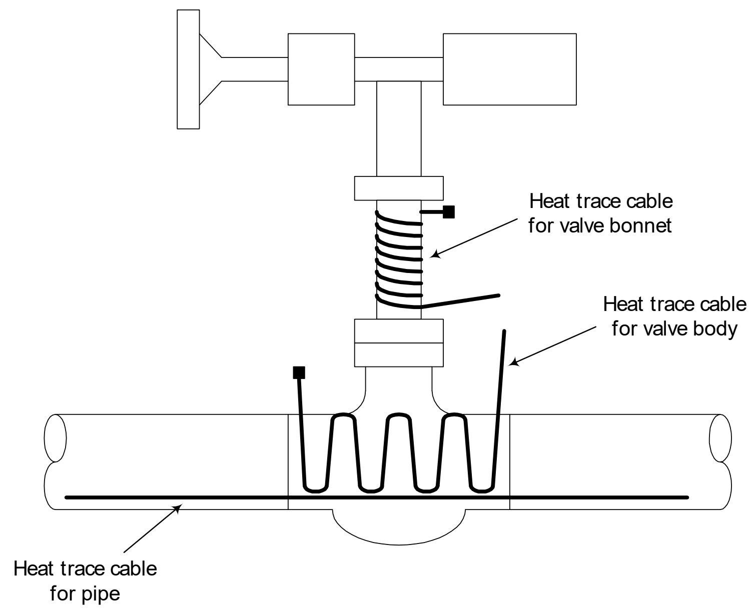

Figure 9-4 Typical Heat Trace Cable Arrangement on Valves

- For long pipes, operating in cyclic service, as many as 6 to 8 dual-element thermocouples may be needed to understand the temperature distribution throughout the zone. The heat trace circuits would be controlled based on a low select feature

- For pipes operating with salt flowing in some modes, but switching to static salt in other modes, a minimum of 2, and as many as 6, dual-element thermocouples may be needed to maintain set point temperatures during all of the modes.

9.8.4 Equipment Heat Tracing

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

Pressure Safety Valves

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

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

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

Vortex Shedding Flow Meters

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

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

Tank Vents

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

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

Tank Vacuum / Pressure Relief Valves

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

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

Tank Level Gauges

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

Capillary Lines for Diaphragm Pressure Transmitters

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

Salt Pumps

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

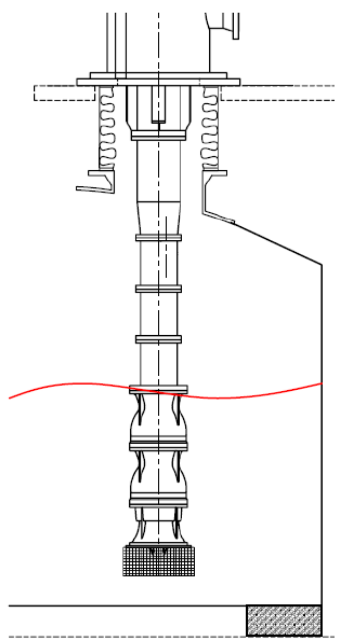

Figure 9-5 Elevation View of Salt Pump

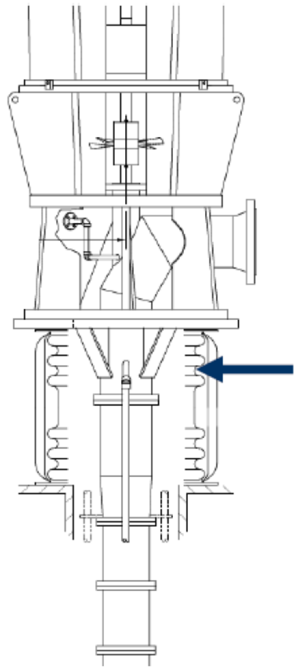

Figure 9-6 Detail of Salt Pump Discharge Head and Bellows

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

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

9.9 Salt Pumps

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

Commercial tank designs have volumes on the order of 17,000 m^3. The mass of steel in the tank, and to a first order the price, reaches a minimum value with a height-to-diameter ratios of 1. As such, the optimum tank would have a height and a diameter of 28 m. However, API 650-style tanks are supported by the parent soil beneath the foundation base mat. A representative allowable soil bearing load at many project locations is 240 kPa (5,000 lb_f/ft^2), which effectively limits the allowable salt inventory depth to about 12 m. As such, overall pump shaft lengths of 14 to 16 m are expected.

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

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

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

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

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

<u>Installed Redundancy</u>

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

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

9.10 Salt Instruments

9.10.1 Pressure Instruments

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

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

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

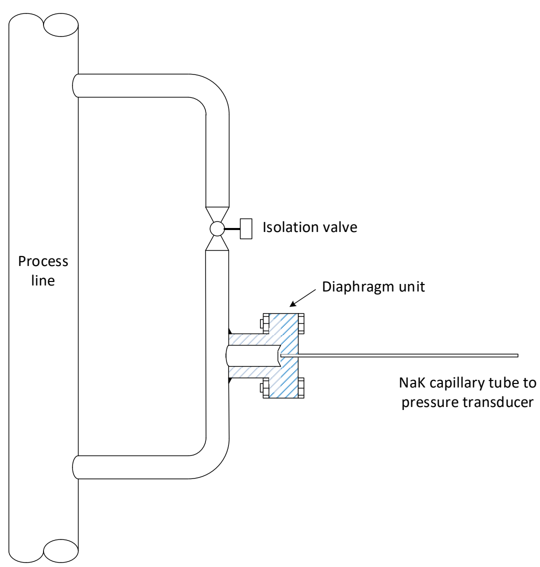

Figure 9-7 Pressure Transmitter in a Vertical Process Line

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

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

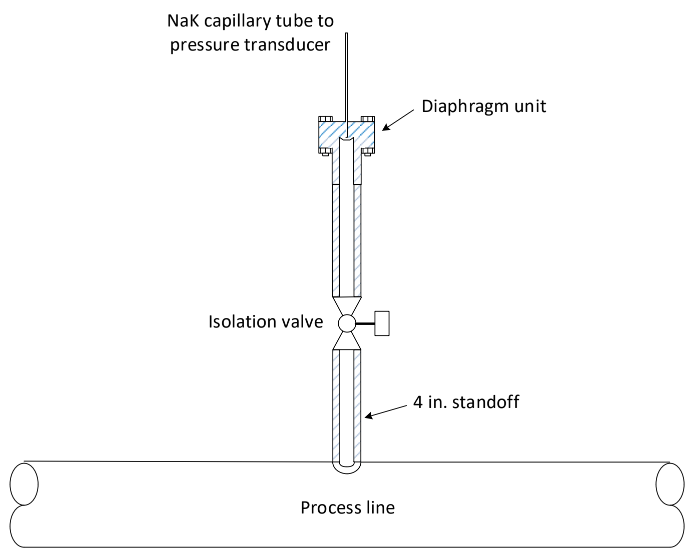

Figure 9-8 Pressure Transmitter in a Horizontal Process Line

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

9.10.2 Flow Instruments

Candidate Designs

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

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

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

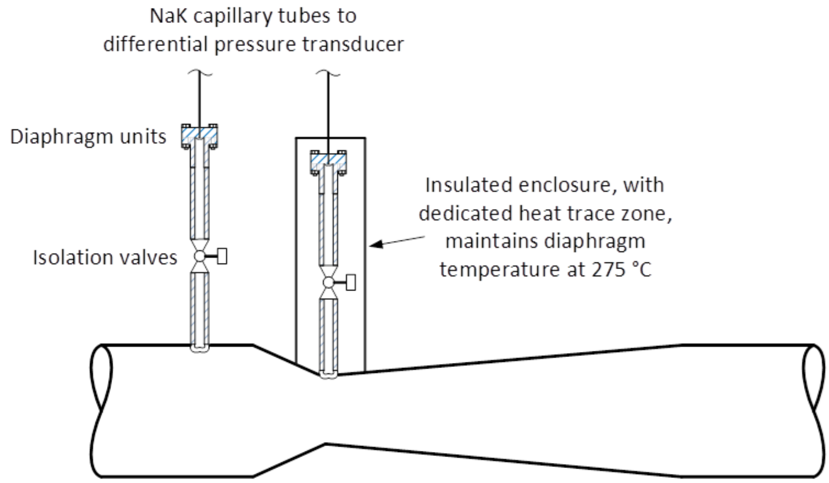

Figure 9-9 Flow Venturi with Diaphragm Pressure Transmitters

Venturis, flow nozzles, and orifice plates are available from a number of commercial vendors, and can operate over the full range of temperatures and flow rates expected in a commercial project. However, it 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

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

9.10.3 Temperature Instruments

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

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

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

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

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

9.10.4 Level Instruments

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

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

10. Functional / Prescriptive Specifications - Inorganic Heat Transport Fluids

10.1 Introduction

Plant and equipment specifications fall into two broad categories:

- Functional specifications describe what the system or the equipment needs to do, consistent with the minimum legal requirements of the local jurisdictions. The details of how this is to be accomplished is developed by the engineering contractor. This allows the engineering contractor to define the process and the equipment designs such that the functional requirements can be met at the lowest cost

- Prescriptive specifications, which are developed by the Owner, prescribe to the engineering contractor how the functional requirements of the project are to be met. Prescriptive elements include items such as the process design, Code Section selection, fabrication techniques, materials selection, and component details. This arrangement ensures that the favorable experience from a previous project is repeated, and helps to avoid situations in which an inexperienced contractor repeats mistakes from earlier projects.

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

- The Owner must have a thorough understanding of the process and the technical features of the plant. For example, the heat trace designs in commercial projects often use 1 dual-element thermocouple in each zone as the input to the temperature set point controller. However, a heat trace design, which reflects the experience from previous projects, might use 2 to 4 thermocouples, distributed along the length of the zone, as inputs to the controller. This arrangement can help to detect defects in the insulation, local areas with temperatures below the freezing point of the salt, local areas which are receiving too much heat input from adjacent zones, and leakage past isolation valves.

- The Owner is prepared to accept what is likely to be an increase in the capital cost as a means of achieving the goals set in the performance and in the financial models. For example, the Owner may require the use of all-welded construction in salt heat exchangers. This fabrication approach is more expensive than the conventional approach; i.e., tube rolling followed by strength welding of the tube to the tubesheet. However, an all-welded design is more tolerant of the transient stresses associated with 1) daily startup and shutdown, and 2) mistakes made by the operators. In turn, the improvement in plant availability more than compensates for the marginal increase in the cost of the heat exchangers due to an all-welded design.

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

10.2 Design Codes and Standards

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

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
HEI	Heat Exchanger Institute
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
IBC	International Building Code

10.3 Nitrate Salt Specification

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

- Binary nitrate salt (solar salt) mixture, which is 60 weight percent sodium nitrate ($NaNO_3$) and 40 weight percent potassium nitrate (KNO_3)

- Ternary nitrate/nitrite salt mixture, the most common of which is Hitec®, with contributions of 7 weight percent sodium nitrate ($NaNO_3$), 40 weight percent sodium nitrite ($NaNO_2$), and 53 weight percent potassium nitrate (KNO_3)

- A ternary nitrate salt mixture, Hitec XL®, with contributions of 42 weight percent calcium nitrate ($Ca(NO_3)_2$), 15 weight percent sodium nitrate ($NaNO_3$), and 43 weight percent potassium nitrate (KNO_3)

- A ternary nitrate salt mixture, which, with contributions of 30 weight percent lithium nitrate ($LiNO_3$), 18 weight percent sodium nitrate ($NaNO_3$), and 52 weight percent potassium nitrate (KNO_3), offers the lowest melting point among the ternary nitrate mixtures (~120 °C)

- A quaternary nitrate salt mixture, reviewed by Sandia National Laboratories, with contributions of 22 weight percent lithium nitrate ($LiNO_3$), 17 weight percent calcium nitrate ($Ca(NO_3)_2$), 18 weight percent sodium nitrate ($NaNO_3$), and 43 weight percent potassium nitrate (KNO_3)

- A quinary nitrate/nitrite salt mixture, patented by Sandia National Laboratories, with contributions of 13 weight percent lithium nitrate ($LiNO_3$), 10 weight percent sodium nitrate ($NaNO_3$), 12 weight percent potassium nitrate (KNO_3), 29 weight percent sodium nitrite ($NaNO_2$), and 36 weight percent potassium nitrite (KNO_2).

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

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

With an organic working fluid in the collector field, the maximum temperature is limited to a nominal 392 °C based on the thermal stability of the oil. With an inorganic working fluid, the maximum temperature of the working fluid must be greater than 392 °C to offset the complexity and the cost of the electric heating system. The benefits of the higher temperature are 1) an increase in the efficiency of the Rankine cycle, 2) a reduction in the unit cost of the storage system, in $/kWht, due to an increase in the difference between the cold tank and the hot tank temperatures, and 3) a reduction in the cost of the storage system due to a switch from an indirect system to a direct system; i.e., eliminating the oil-to-salt heat exchangers. Previous studies have shown that the collector field outlet temperature should be at least 450 °C to offset the cost of the electric heating systems [11].

Binary Nitrate Salt

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

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

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

Hitec

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

$$2\ NO_2^- \rightarrow 2\ NO + 2\ O^{--}$$

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

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

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

Hitec XL

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

$$Ca(NO_3)_2 \rightarrow Ca^{++} + 2\ NO_3^-$$

$$NO_3^- \rightarrow NO_2^- + ½\ O_2$$

$$2\ NO_2^- \rightarrow 2\ NO + 2\ O^{--}$$

$$Ca^{++} + O^{--} \rightarrow CaO$$

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

On a related point, the relative stability of salt mixtures containing lithium nitrate ($LiNO_3$) (discussed in the section below) is greater than mixtures containing calcium nitrate ($Ca(NO_3)_2$), as indicated by the maximum temperature at which decomposition products became appreciable. The trend in stability of ternary mixtures followed the correlation for single salts established by Stern [12]. This correlation is based on a parameter of the cation species consisting of the square root of the covalent metallic radius, r, divided by the effective nuclear charge, Z. Stability increases as the cation parameter increases, which implies that electrostatic effects between the cations and the polyatomic nitrate anions are important. The values of this parameter for the alkali and alkaline earth metals of interest are given in Table 10-1.

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

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

The stability of ternary mixtures appears to be determined primarily by the least stable constituent. The types of cations in the salt mixtures had relatively little effect on the nitrate / nitrite / oxygen equilibrium reaction (i.e., $NO_3^- \leftrightarrow NO_2^- + ½ O_2$) over the range of compositions investigated. In contrast, decomposition to form corrosive oxide ions was strongly influenced by the types of cations present. This suggests that cations may interact more strongly with nitrite or oxide ions than with nitrate ions.

Ternary Nitrate Salt

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

- There are no nitrite contributions to the mixtures

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

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

Nonetheless, as summarized in Table 10-2, ternary mixtures containing $LiNO_3$ can produce significant concentrations of oxide ions [13]. As a point of reference, the 60/40 binary salt reportedly contains less than 1 * 10^{-2} molal total oxide ions under similar conditions of temperature and atmosphere.

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

$LiNO_3$:$NaNO_3$:KNO_3, weight percent	Melting point, °C	Operating temperature, °C	NO_2^- concentration, 10^{-2} molal	Oxide concentration, 10^{-2} molal
12 : 18 : 70	200	500	8.5	1.1
		550	20.6	1.2
20 : 28 : 52	160	500	11.1	1.2
		550	30.0	2.9
27 : 33 : 40	150	500	12.6	7.6
		550	27.8	11.0
30 : 18 :52 [1]	120	500	12.4	11
		550	27.0	20

Note 1: Eutectic mixture

It should be noted that the above data were developed with an ullage gas composition of 100 percent O_2. This was done to prevent the formation of lithium carbonate from the CO_2 normally present in atmospheric air. However, if O_2, rather than air, is used as the ullage gas, this forces the following equation to the left:

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

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

$$2\ NO_2^- \rightarrow 2\ NO + 2\ O^{--}$$

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

In central receiver projects, the long term nitrite concentration is in the range of 2 to 3 weight percent [14], and the associated oxide concentrations (which, to some degree, are unknown) result in acceptable corrosion rates for carbon, ferritic, and austenitic steels. A nitrite concentration of 2.5 percent by weight is equivalent to a molal concentration of $5.2 * 10^{-2}$. As indicated in Table 10-2, to achieve comparable corrosion rates between a ternary nitrate salt (using an ullage cover gas of O_2) and the 60/40 binary nitrate salt (using an ullage cover gas of air), the lithium contribution in the ternary salt must be held to values less than 20 percent. This in turn, results in a melting point in the range of 180 to 200 °C.

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

Quaternary Salt

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

Quinary Salt

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

Salt Selection

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

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

The remaining candidate salts include the following:

- The 60/40 binary nitrate salt mixture, with a low cost and a high thermal stability, but with a high freezing point of 235 °C

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

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

- The cost for freeze protection in a plant with a fluid melting point of ~190 °C is not significantly lower than the cost for freeze protection in a plant with a fluid melting point of 235 °C

- There is a considerable data base, totaling several hundred plant years, on the use of the binary salt in commercial service

- The 60/40 binary salt is immune from the swings in the commodity prices of lithium carbonate seen in the past few years.

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

- Grades: Section 9.3.1

- Total Chloride Concentration: Section 9.3.2

- Allowable Contaminants: Section 9.3.3

- Supply Options: Section 9.3.4

- Deviations from a 60/40 Mixture: Section 9.3.5.

10.4 Nitrate Salt Handling and Melting Specification

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

10.5 General Salt System Equipment

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

A description of the high temperature (500+ °C), stainless steel piping for trough projects using inorganic heat transfer fluids is discussed below.

10.5.1 Piping

Design Codes

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

<u>Collector Field Piping</u>

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

- The design temperatures of the hot salt piping in a trough project using an inorganic heat transfer fluid (510 to 540 °C) are similar to the design temperature of the hot salt piping in a central receiver project (565 °C)

- If a project is to meet the availability values typically adopted for project financing, then the field piping in a trough project must have an availability comparable to the availability of the salt piping in a central receiver project; i.e., at least 99.5 percent.

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

- The weld examinations are less rigorous

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

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

<u>Steam Generation System and Rankine Cycle</u>

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

- The cold salt and the hot salt piping to, within, and from the steam generator

- The water/steam piping segments in and around the steam generator and the turbine-generator, which operate at combinations of high temperature and high pressure.

Materials

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

Stainless Steel Material Specification

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

<u>Stabilization Heat Treatment of Type 347H Piping</u>

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.

Post Weld Heat Treatment of Type 347H Pipe and Fittings

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

10.5.2 Flanges

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

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

10.6 Heat Tracing

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

Cable Redundancy

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

- 100 percent for the cables 1) on the hot salt piping from the collector field, and 2) the hot salt piping to the steam generator. These segments of piping operate under daily thermal cycles, and the failure rates of the heat trace cables are expected to be ‘moderate to high’

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

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

10.7 Salt Pumps

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

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

10.8 Salt Instruments

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

10.9 Collector Field

10.9.1 Solar Collector Assemblies

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

The principal differences in the hardware lie in the connections between the rotating heat collection elements and the fixed piping. In a project using Therminol, the most common connection is a ball joint. A cross section of a representative design is shown in Figure 10-1. A threaded retainer loads the two seals against the inner ball. A packing material, consisting of grease and graphite flakes, is injected between the two seals to form a liquid-tight boundary. However, all of the candidate salts discussed in Section 10.3 are oxidizing materials. Over time (days to months) the salt reacts with the grease and the graphite to form CO_2. The packing eventually disappears, and the joint begins to leak. Considerable research effort has been devoted to finding either an organic or an inorganic packing material that compatible with salt and provides a liquid-tight boundary, but to date this has proven unsuccessful.

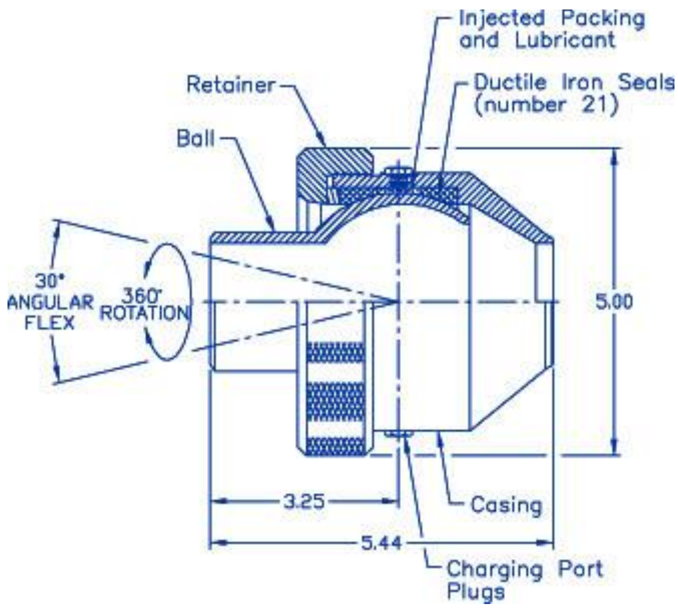

Figure 10-1 Cross Section of a Representative Ball Joint

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

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

A representative flex hose assembly, in its shipping container, is shown in Figure 10-2. The flex hose assembly, installed on a prototype collector, is shown in Figure 10-3. The upper portion accommodates the thermal expansion and contraction of the receiver tubes, and the two lower section accommodate the collector rotation by bending.

10.9.2 Impedance Heating of Receiver Tubes and Flex Hose Assemblies

The absorber tube for the receiver is enclosed in a glass envelope, and a vacuum is drawn in the annular space between the receiver tube and the glass envelope. The receiver tubes require some form of heat addition, other than from solar radiation, for the following purposes:

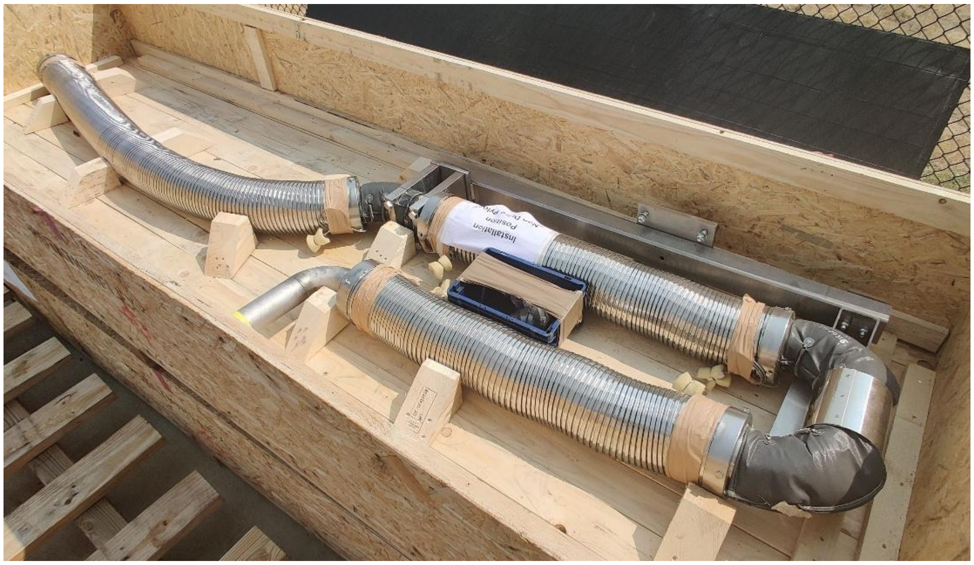

Figure 10-2 Flex Hose in Shipping Container

Figure 10-3 Flex Hose Assembly Installed on a Prototype Collector

- The receiver tubes must be preheated prior to filling with salt to prevent a thermal shock which could break the glass-to-metal seal

- Should salt freeze in the receiver tubes, heat must be added to melt the frozen sections.

Two forms of electric heating have been considered for design of the receiver:

- A conventional heat trace cable, installed on the non-illuminated inside of the receiver tube

- A direct current power supply, which passes a high current through the receiver tubes. The current is converted to heat by the familiar I^2R effect.

The industry has generally not pursued the first approach, for the following reasons:

- The electric cable providing power to the heat trace cable must penetrate the vacuum boundary. The seal for the cable must remain vacuum-tight for decades under thermally cycling conditions. To date, a suitable design for the seal has yet to be identified

- There is a commercial market for parabolic troughs using Therminol as the working fluid; however, there is no commercial market for parabolic troughs using salt as the working fluid. As such, the limited number of commercial receiver suppliers have little motivation to develop a reliable seal for the electric connection.

The impedance heating approach is feasible for receiver tubes due to a combination of a small metal cross section area and a relatively high resistivity for stainless steel. However, based on local jurisdictional requirements, the maximum voltage is limited to either 80 V, based on IEEE requirements [15], or is limited to 50 V, based on NEC Article 110.27 [16].

The low voltage allowances result in the following features of an impedance heating system:

- A representative power input to the receiver tubes is 250 W/m. This translates to electric currents in the range of 400 to 600 Amps, depending on the receiver diameter and the wall thickness. The large electric currents result in large transformers, switchgear, and electric cables for the distribution of power to the trough structures, compounded by the need to keep the voltage drops in the range of only a few Volts. For example, a representative size of the electric power cables is illustrated at the top of Figure 10-3, where the combined cross section areas of the three cables approaches the diameter of the receiver tube

[15] "IEEE Recommended Practice for Electrical Impedance, Induction, and Skin Effect Heating of Pipelines and Vessels", Institute of Electrical and Electronics Engineers, Inc., Standard 844-1991, March 1991

[16] 2017 NEC edition of NFPA 70, '110.27(A) Live Parts Guarded Against Accidental Contact'

- The receiver tube and the flex hose connections are supported by the collector structure. The structure has a much lower electric resistivity than the receiver tubes and the flex hoses. As such, the electric isolation of the receiver tubes and the flex hoses must be essentially perfect. Otherwise, the heating currents will shunt around the receiver tubes and the flex hoses, which decreases the heat input to the receiver tubes and the flex hoses

It can be noted that the electric resistance of the receiver tubes is different than the electric resistance of the flex hose assembly. If the receiver tubes and the flex hoses are to be preheated at the same rate, then separate power supplies, operating at different currents and voltages, will be required for the receiver tubes and the flex hose assemblies.

Design Requirements

As the design of the impedance heating system is fundamentally an electrical problem, each collector loop is considered as a series of electrical resistance elements. The impedance heating system may connect at any point between the resistance elements shown in Figure 10-4, and may be composed of one or multiple power supplies based on the geometry

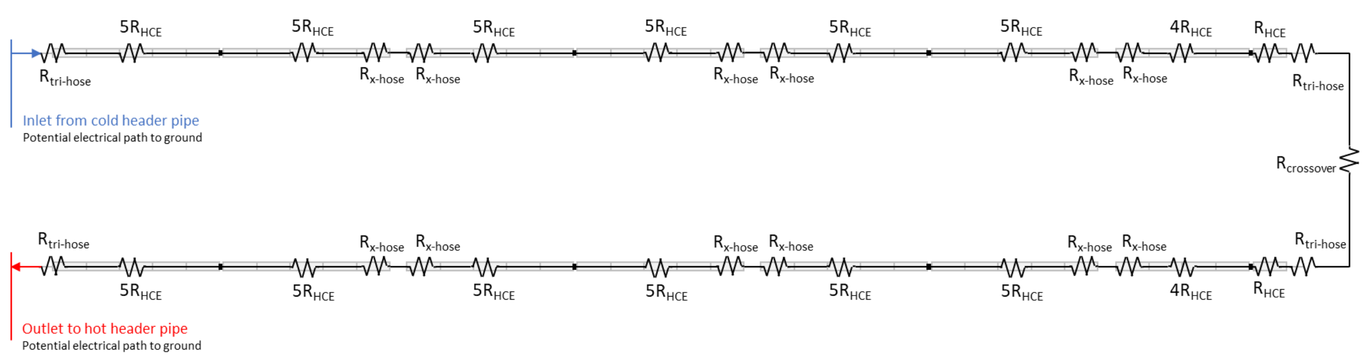

Figure 10-4 Electrical Resistance Network a Collector Loop

The main challenge with impedance heating systems is eliminating paths to ground. The receiver tubes, the flex hose assemblies, and the fixed piping that compose the electrical circuit in Figure 10-4 have numerous paths to ground through the collector structure, and all of the paths need to be electrically isolated. In addition, the field header piping represents additional paths to ground that cannot be easily isolated, but must be considered in the wiring configuration.

Permanent Systems

A commercial parabolic trough plant may be made up of several hundred collector loops. In principle, permanent impedance systems can be installed on each loop, and the plant could recover from a freeze event in approximately 3 weeks by heating the field piping, the receiver tubes, and the flex hose assemblies simultaneously. In practice, such a system would be limited by the capacity of the plant auxiliary transformers to distribute power from the grid to the various electric heating loads. One study estimates that a maximum of 10 collector loops could be melted at any one time based on the typical capacity of auxiliary transformers in commercial projects [17]. This would result in a freeze recovery time approaching 100 days for the very largest solar fields.

As might be expected, given the large sizes of the electric equipment required for impedance systems, the estimated capital cost to provide a permanent system on each loop is quite high, and may approach $\$80/m^2$. This is approximately 50 percent of the unit cost of the collector field, and would render the approach infeasible.

Mobile Systems

An alternate approach to a permanent impedance heating system is a mobile system. The system consists of a number of trailers towed by trucks. On each trailer is a generator, a step-down transformer, and electric cables stored on a series of reels. The cables provide temporary connections between the step-down transformer and permanent electrical connection points in the collector loop.

The truck and each trailer have a common ground bus for connecting the collector structure to the generator neutral. A ground cable also connects to a local permanent grounding well that is bonded to the plant permanent grounding grid.

The estimated cost of a mobile system is on the order of \$1,500,000 [18]. An economic analysis will determine the optimum combination of the number of impedance heating systems and the time to recover from a frozen collector field. For the complete plant, the estimated unit cost of the mobile heating systems is on the order of \$5 to $\$10/m^2$ of collector area.

Electric Power Supply Arrangements

The arrangement of the electric connections between the power supply and the receiver tubes, or between the power supply and the flex hose assemblies, has, as its principal goal, the elimination of

[17] C. Prieto, A. Rodríguez-Sánchez, F. J. Ruiz-Cabañas and L. F. Cabeza, "Feasibility study of freeze recovery options in parabolic trough collector plants working with molten salt as heat transfer fluid", *Energies*, vol. 12, no. 12, pp. 1-20, 2019
[18] Shininger, R. (Solar Dynamics LLC, Broomfield, CO), "Simplified Melting and Rotation Joint Technology", DOE SunShot Technology to Market Award Number DE-EE0008140, August 2022

current leakage into the supporting structure. Two configurations have been considered in an effort to reach this goal; midpoint cable; and staggered polarity.

Midpoint Cable The midpoint cable wiring method, illustrated in Figure 10-5, is described in a conceptual design study developed for NREL [19], and it is described in a freeze protection study which considers a reduced order model to estimate the melt time [12]. This method involves wiring the positive terminal in the middle of the element to be heated, and splitting the negative terminal between the two ends. In this fashion, the ends of two adjacent circuit will always be at the minimum voltage, and the maximum voltage is at the electrical center of the circuit. This wiring configuration can also be advantageous for avoiding current leaks to ground.

In the Figure, the difference between the supply voltage (80 V) and the voltage supplied to the receiver tubes (72 V) is the volage drop in the electric cables between the power supply and the receiver tubes. Similarly, the difference between the outlet voltage from the receiver tubes (8 V) and the voltage at the power supply (0 V) is the voltage drop in the electric cables between the receiver tubes and the power supply.

Staggered Polarity The staggered polarity method is described in a report looking into alternate methods for freeze protection and recovery [17]. This method involves wiring multiple power supplies in series with opposing polarity to limit the maximum voltage of the total circuit. Only the output terminal of the power supply immediately adjacent to the header piping is grounded. With the staggered polarities and voltage drop of the interconnecting portable cables, the maximum voltage from any section of piping to ground is less than the maximum voltage across any of the power supply terminals.

To illustrate the concept, a collector loop in a potential commercial project is shown in Figure 10-6.

The electric resistance of the loop is represented above in Figure 10-4. A series of portable power supply trailers are placed within the loop, as shown in Figure 10-7.

[19] K. &. Associates, "Engineering & Evaluation of a Molten Salt HTF in a Parabolic Trough Solar Field," NREL Contract No. ANN-30441-04, Final Report, 2002

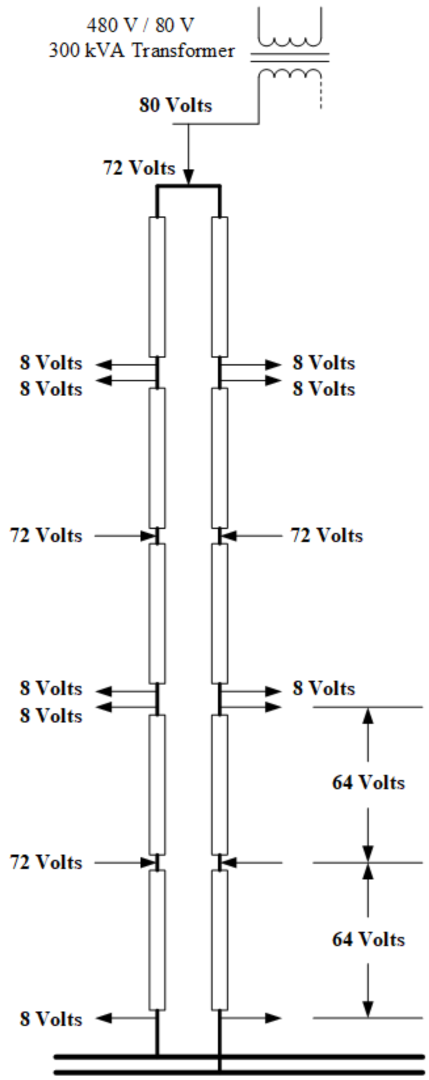

Figure 10-5 Midpoint Cable Wiring Arrangement

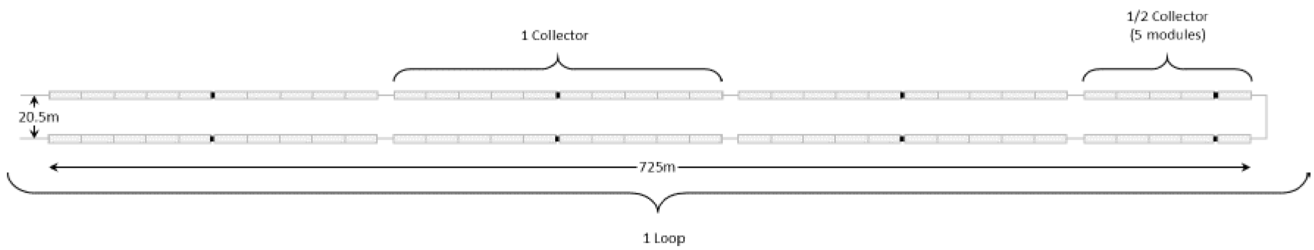

Figure 10-6 Collector Loop in a Potential Commercial Project

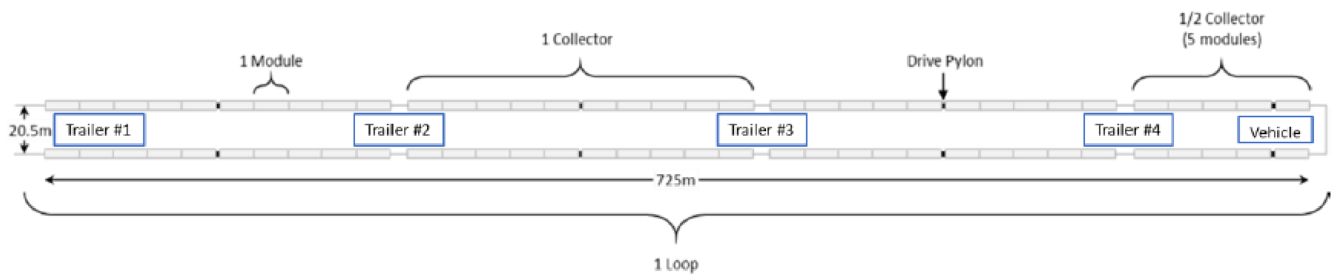

Figure 10-7 Portable Power Supplies for the Impedance Heating System

A conceptual arrangement for the power distribution is shown in Figure 10-8. The green outlines represent the trailers shown in Figure 10-7.

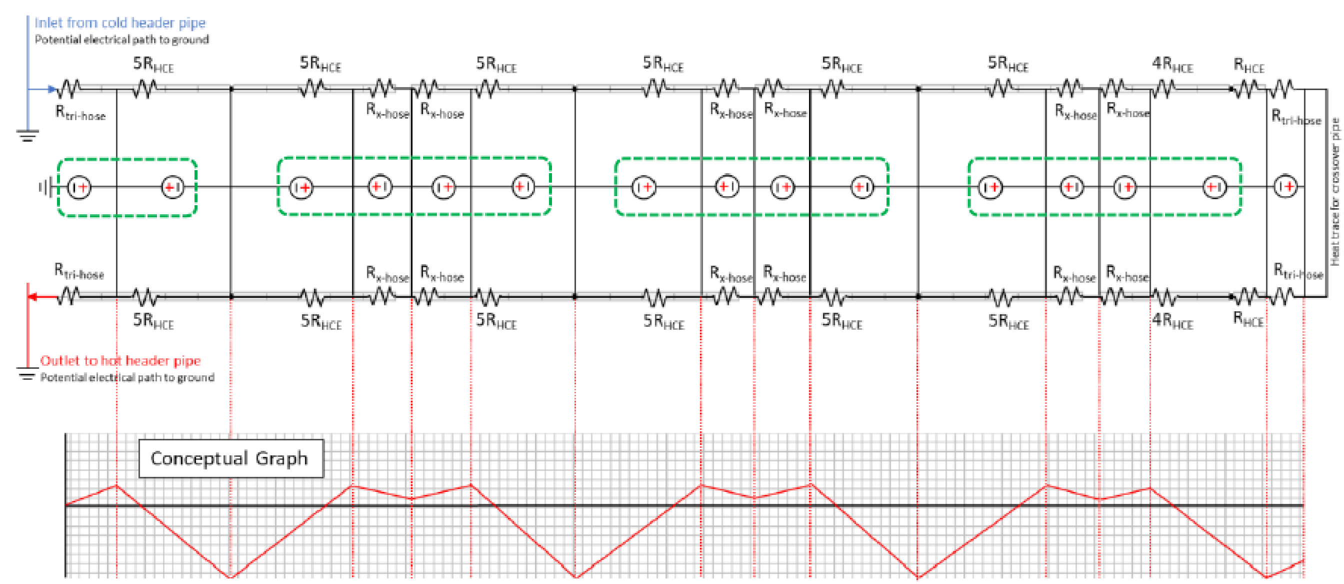

Figure 10-8 Conceptual Power Distribution with Staggered Polarity

10.9.3 Resistance Heating of the Field Piping

The resistance heating system for the field piping follows the basic design approach discussed in Section 9.8.

As discussed in Section 2.5.3 of Volume 3, Narrative, the heat tracing on the field piping will, at some point in the life of the project, be required to recover the field piping from a frozen condition. This process, as one might imagine, is rather lethargic, with thaw periods ranging from 1 week to 1 month, depending on the diameter of the piping.

Heat tracing is expensive, and the design duty of the system is never generous. As such, the heating rate of the pipe is sensitive to the condition of the pipe insulation. In the plant design phase, an economic analysis will be conducted to determine the optimum combination of the design duty of the heat trace system and the recovery time from a frozen condition. The uncertainties in the analysis are likely be high, as there are no data on the frequency of the freezing events in commercial projects. Nonetheless, the design capacity of the heat tracing should assume some level of degradation in the thermal resistance of the pipe insulation. This is to prevent large deviations between the expected thawing period and the actual thawing period due to slow, but expected, degradation in the quality of the insulation.

10.10 Thermal Storage System

The thermal storage system in a trough project using an inorganic heat transfer fluid is nominally the same as the thermal storage system in a central receiver project. A discussion of the specifications for the thermal storage equipment is presented in Section 6.10 of Volume 2 - Specifications for Central Receiver Projects.

10.11 Steam Generation System

The steam generation system in a trough project using an inorganic heat transfer fluid is nominally the same as the steam generation system in a central receiver project. A discussion of the specifications for the steam generator equipment is presented in Section 6.12 of Volume 2 - Specifications for Central Receiver Projects.

10.12 Steam Bypass System and Air Cooled Condenser Capacities

The criteria for selecting the capacities of the steam bypass system and the air cooled condenser in a trough project using an inorganic heat transfer fluid is nominally the same as the criteria used in a central receiver project. A discussion of the specifications for the steam bypass system and the air cooled condenser is presented in Section 6.14 of Volume 2 - Specifications for Central Receiver Projects.

11. Current State of the Art - Organic Heat Transport Fluids

11.1 Introduction

The current state of the art in the salt equipment of parabolic trough projects using organic heat transfer fluids is presented in the sections below. Topics include configurations, working fluids, temperatures, rates of temperature change, efficiencies, materials, heat exchangers, pumps, instruments, and valves. The discussion is, in essence, a compilation of functional and prescriptive specifications that incorporate lessons learned from successful commercial plants.

11.2 Configurations

A schematic diagram of a parabolic trough plant with thermal storage is illustrated in Figure 11-1. The Therminol components are shown in red, and the salt components are shown in green. A schematic diagram, showing the salt thermal storage system and the Therminol steam generators, is shown in Figure 11-2.

An elevation view of the storage tanks and the oil-to-salt heat exchangers is shown in Figure 11-3. As discussed in below in Section 11.4, the optimum log mean temperature for the heat exchanger is in the range of 5 to 7 °C. Due to the modest thermal conductivity of nitrate salt, this results in a longer heating length than can be provided in one shell. A representative upper limit on the length of commercial heat exchanger tubes is 30 m (100 ft.), which leads to a requirement for 6 heat exchangers in series.

A plan view of the tanks and the heat exchangers is illustrated in Figure 11-4.

11.3 Working Fluids

The working fluid in the collector field, and the working fluid on the oil side of the oil-to-salt heat exchanger is the eutectic mixture of diphenyl oxide and biphenyl. It is commercially available from Eastman Chemical under the trade name Therminol VP-1.

The working fluid in the thermal storage system, and the working fluid on the salt side of the oil-to-salt heat exchanger, is a mixture of sodium nitrate ($NaNO_3$) and potassium nitrate (KNO_3). The most common blend is 60 percent by weight sodium nitrate and 40 percent by weight potassium nitrate (64 mole percent sodium nitrate and 36 mole percent potassium nitrate).

A note: This mixture is not the eutectic, which would be a blend of 50 mole percent sodium nitrate and 50 mole percent potassium nitrate. Since the mixture is not the eutectic, the blend has a melting range, rather than a single melting point, as shown in Figure 11-5.

A 60/40 mixture has been adopted in commercial projects because the sodium nitrate is less expensive than the potassium nitrate, and the moderate deviation from the eutectic does not result in a large increase in the liquidus temperature.

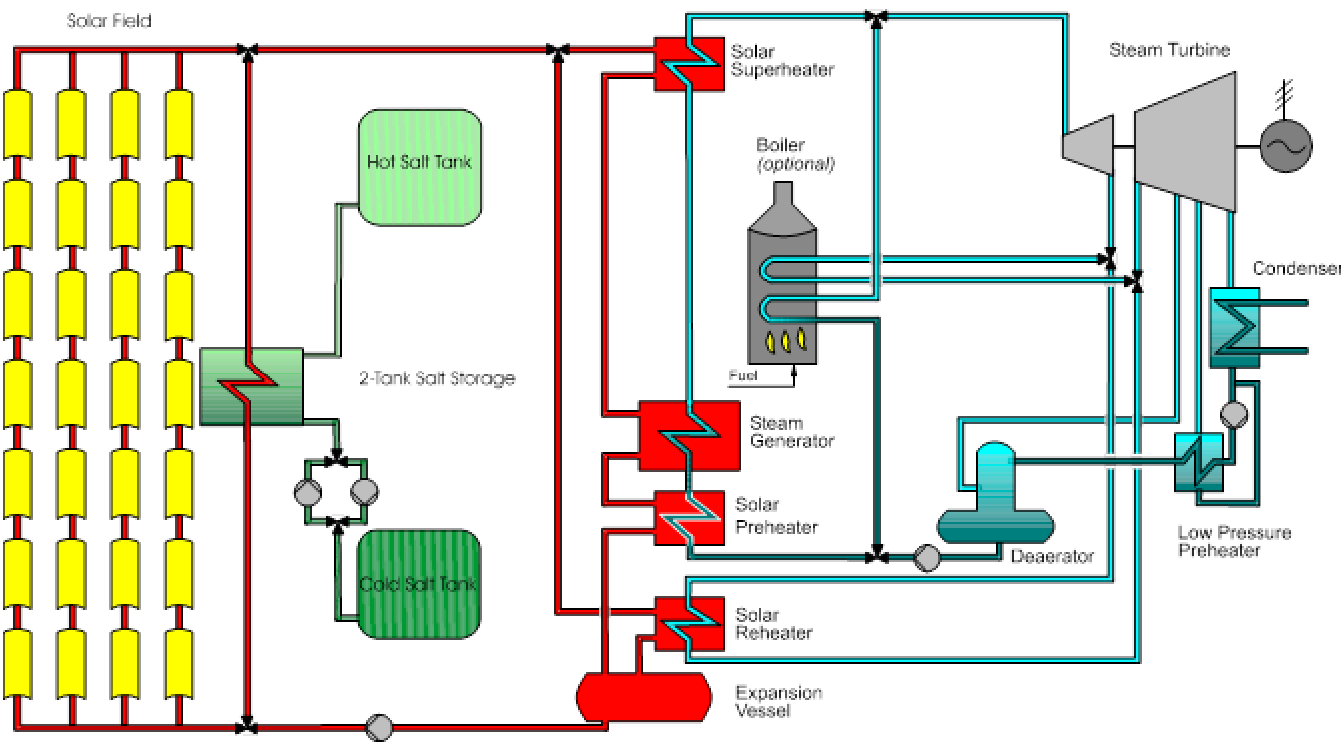

Figure 11-1 Schematic Diagram of a Parabolic Trough Plant with Nitrate Salt Thermal Storage

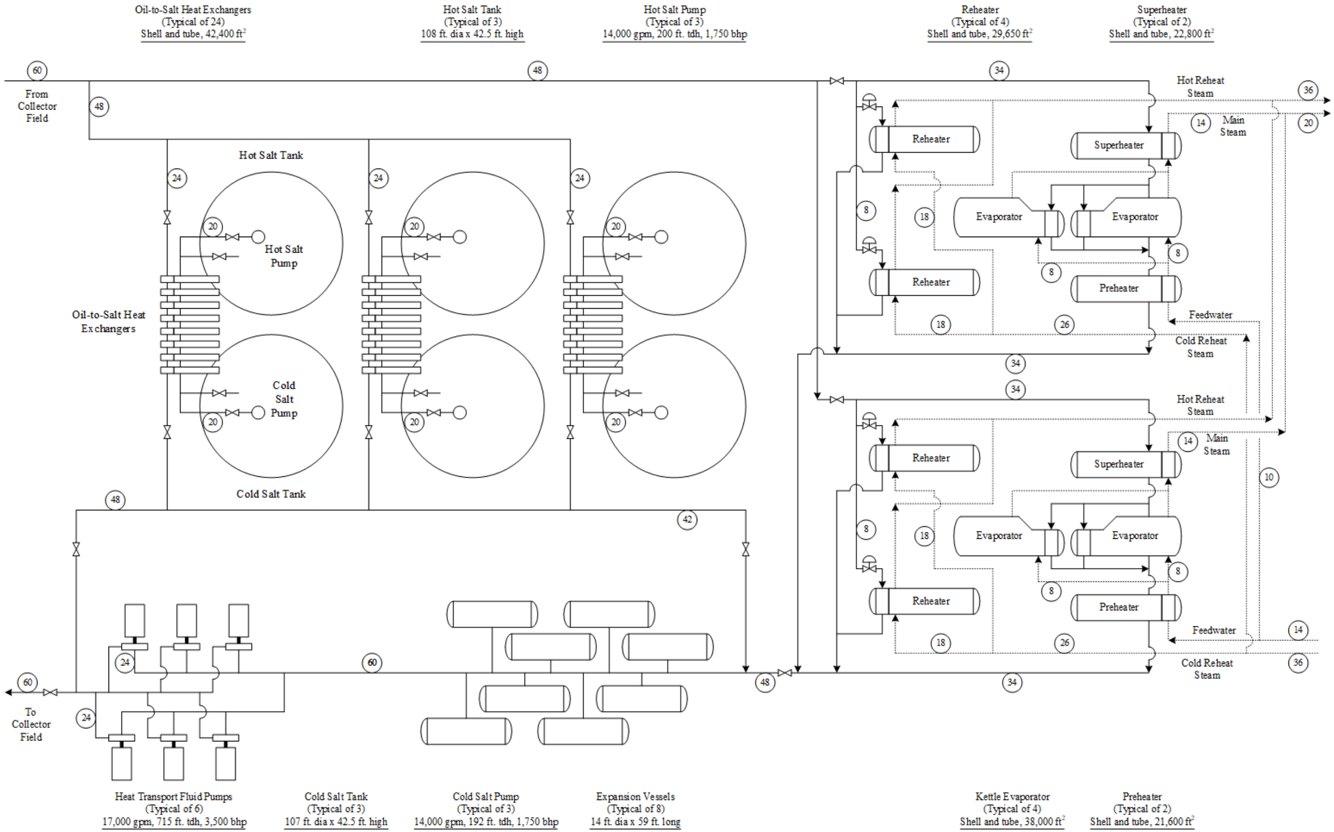

Figure 11-2 Schematic Diagram of Salt Thermal Storage System and Therminol Steam Generators

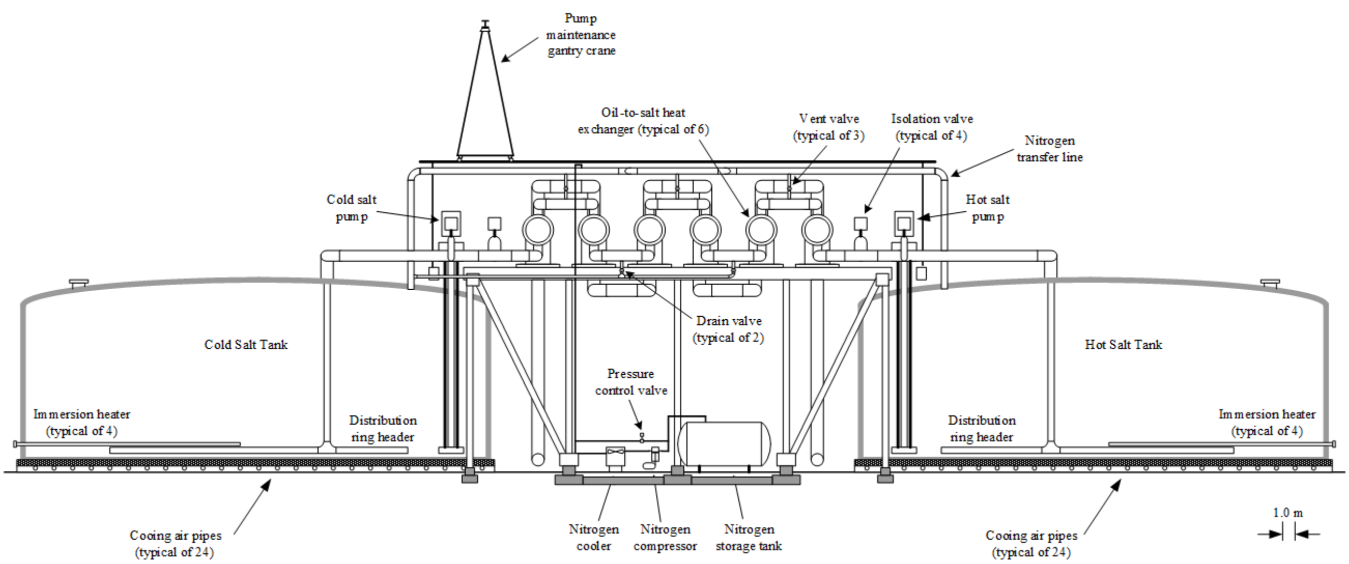

Figure 11-3 Elevation View of Thermal Storage Tanks and Oil-to-Salt Heat Exchangers

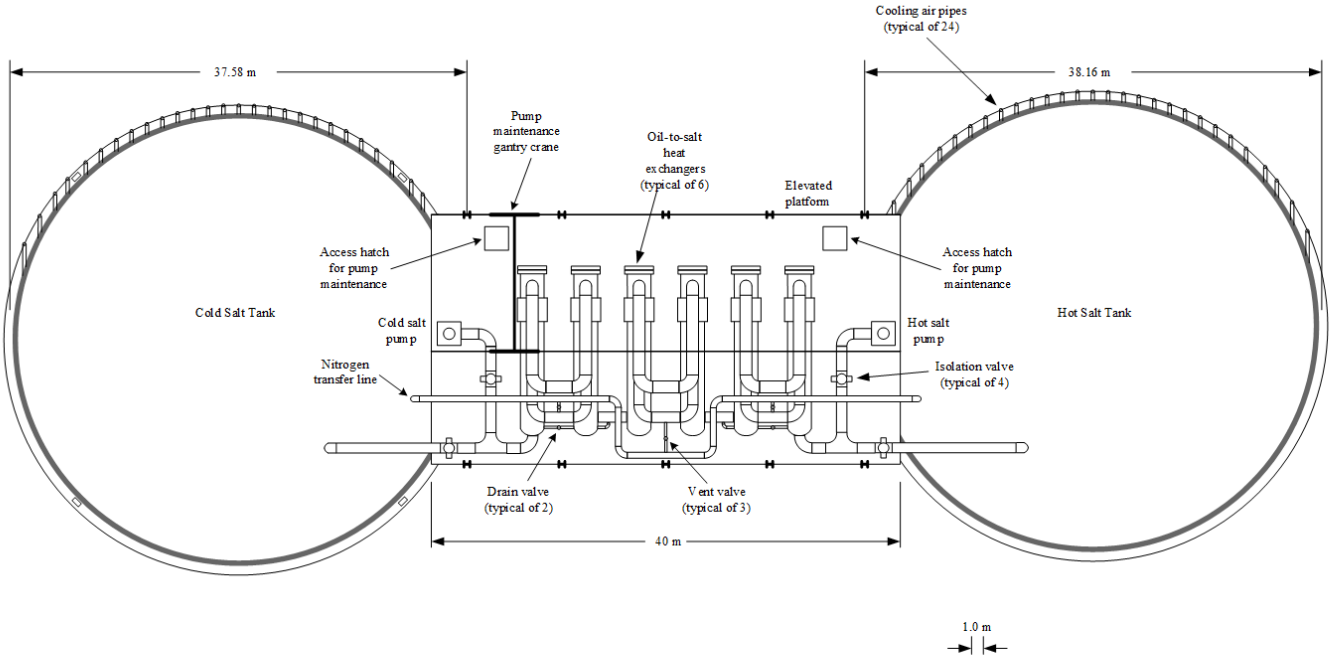

Figure 11-4 Plan View of Thermal Storage Tanks and Oil-to-Salt Heat Exchangers

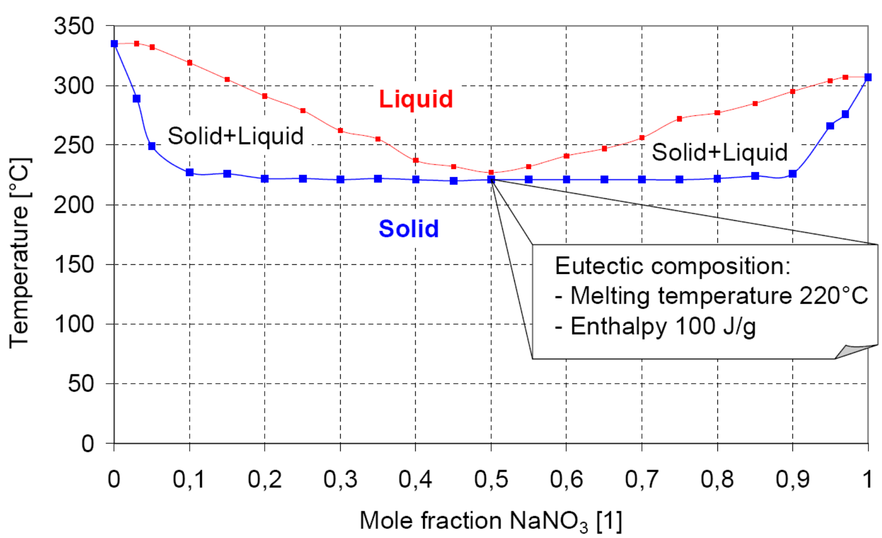

Figure 11-5 Melting Point of Binary Nitrate Salt Mixtures

11.4 Process Temperatures

11.4.1 Therminol

As with any power plant, the efficiency of the Rankine cycle is maximized by operating at the highest pressure ratio across the steam turbine, and at the highest final feedwater temperature, as practical.

In commercial parabolic trough projects, the source temperature for the Rankine cycle is limited by the maximum allowable operating temperature for the Therminol in the collector field. A representative value is currently 392 °C. At this temperature, the Therminol slowly decomposes, producing a range of light hydrocarbons, such as benzene, and a range of heavy hydrocarbons. The benzene is removed on a semi-continuous basis by exposing the Therminol to a free surface in the expansion vessels, collecting the gaseous benzene released from the liquid, adsorbing the benzene in an activated carbon bed, and transporting the carbon offsite for disposal. The heavy hydrocarbons are separated from the Therminol in a distillation column, and the heavy components are removed from the system in a blowdown stream. The maximum operating temperature of 392 °C is essentially a compromise between power plant efficiency and the operating expense of periodic Therminol makeup.

With a source temperature of 392 °C for the steam generator, the live steam temperature is on the order of 379 °C based on approach temperatures typical of commercial steam generators. With a live steam temperature of 379 °C, and the potential for reheating the steam to 379 °C in a reheater, the live steam pressure can be as high as 100 bar and still acceptable steam qualities (90 to 91 percent) at the exhaust of the low pressure turbine.

With a saturation pressure in the evaporator of about 105 bar, and with a typical commercial pinch point temperature in the evaporator (5 °C) and a typical approach-to-saturation temperatures at the hot end of the preheater (5 °C), the temperature of the Therminol at the cold end of the preheater is essentially defined as 294 °C.

In the storage system, the hot salt temperature is determined by the storage charge condition, and is equal to the design Therminol temperature (392 °C) minus the approach temperature at the hot end of the oil-to-salt heat exchanger. A representative approach temperature is 5 °C, and results from an economic analysis of the storage system, which compares the cost of the oil-to-salt heat exchangers with the cost of the storage inventory. As such, the design temperature of the hot salt tank is a nominal 386 °C.

Conversely, the cold salt temperature is determined by the storage discharge condition. During storage discharge, Therminol is supplied to the steam generator from the storage system at a temperature of 386 °C. Reducing the source temperature to the steam generator reduces the mass flow rate of live / reheat steam, and reduces the live / reheat steam temperatures. Since the turbine normally operates in sliding pressure mode, with the control valves open to 100 percent, reducing the live steam flow rate reduces the live steam pressure to values on the order of 92 bar. Reducing the live steam pressure reduces the saturation temperature in the evaporator, which cascades down to a reduction in the Therminol temperature at the cold end of the preheater to a value of approximately 287 °C. A heat transfer analysis on the oil-to-salt heat exchanger shows a corresponding cold salt temperature of 292 °C, and this becomes the temperature of the cold salt tank.

Note that the difference in temperature between the hot tank and the cold tank is a modest 94 °C. This has important implications for the reliability of the storage tanks, as discussed in Section 4.3 of Volume 3 - Narrative.

11.4.2 Silicone Fluids

In addition to Therminol, there is a range of alternate heat transfer fluids based on silicone rather than hydrocarbons. Two high temperature candidates include:

- Syltherm 800, a dimethyl polysiloxane, from Dow Corning Corporation

- Helisol 5A, a polydimethylsiloxane, from Wacker Chemie AG.

According to the suppliers, the silicone fluids can operate at bulk fluid temperatures that are 10 to 25 °C higher than for Therminol. The higher fluid temperatures can, in principle, offer the following benefits:

- Decrease the unit cost ($/kWht) of the storage system, by increasing the difference in temperature between the hot salt tank and the cold salt tank

- Increase the efficiency of the Rankine cycle, by increasing the temperature of both the live steam and the reheat steam, which allows an increase in the live steam pressure.

A comparison of some of the thermophysical properties of Therminol, Syltherm 800, and Helisol 5A are shown in Table 11-1.

The thermal conductivities, the specific heats, and the viscosities of the 3 fluids are similar, and differences in the designs and the performance of the collector field, the storage system, and the steam generator are likely to be modest. However, the minimum pressure in any equipment containing heat transfer fluid must always be at least equal to the vapor pressure to ensure that the fluid remains in the liquid state. The maximum allowable working pressure for commercial heat collection elements is on the order of 40 bar. As such, the maximum allowable pressure drop in the collector field is as follows:

- Therminol: 40 bar - 11 bar = 29 bar

Table 11-1 Comparison of Heat Transfer Fluid Properties

	Therminol	Syltherm 800	Helisol 5A
Temperature limit, °C			
Bulk	392	400	425
Film	425	~ 425	~ 450
Vapor pressure, bar	11	14	16
Autoignition temperature, °C	615	385	358
Thermal conductivity, W/m-°C, at 400 °C	0.075	0.064	0.057
Specific heat, kJ/kg-°C, at 400 °C	2.6	2.7	2.5
Viscosity, mPa-sec, at 400 °C	0.14	0.25	0.11

- Syltherm 800: 40 bar - 14 bar = 26 bar
- Helisol 5A: 40 bar - 16 bar = 24 bar.

The allowable pressure drop influences the length of a collector loop. As such, projects using Syltherm 800 or Helisol 5A may require shorter loops than a project using Therminol, which, in turn, can lead to an increase in the number of loops and a second-order increase in the cost of the field piping.

As discussed in Section 10.9.1, differential movement between the heat collection elements and the support structure is accommodated by ball joints. The reliability of the joints is generally very good; however, if the joint does develop a leak, then the heat transfer fluid is exposed to the ambient. The autoignition temperature of Therminol (615 °C) is well above the design fluid temperature (392 °C), and the fluid does not automatically start to burn following a leak. However, the autoignition temperature for the silicone fluids (~ 375 °C) is below the design fluid temperature (400+ °C). As such, a leak is likely to start a fire.

The higher operating temperatures available from silicone fluids may offer benefits which translate to a reduction in the levelized cost of energy. However, the benefits are likely to be modest, particularly in light of the hundreds of project-years of commercial operating experience with Therminol.

11.5 Collector Structure and Reflectors

A trough collector consists of an elongated horizontal structure in the shape of a linear parabola. The structure supports a series of mirrors, each of which represents a segment of a parabola.

At the focal line of the parabola is a linear receiver. The receiver consists of a metal tube, to which is applied a selective coating. The coating provides a high absorptivity (95 to 96 percent) in the visible spectrum, and a low emissivity (≤ 0.095 percent at 400 °C) in the infrared spectrum.

The metal tube is surrounded by a glass envelope. The envelope has anti-reflective coatings applied to both the inside and the outside of the glass. Air is removed from the annular space between the metal tube and the glass envelope to reduce convection losses. This necessitates the use of a metal bellows, and a glass-to-metal seal, to accommodate the differential thermal expansion between the metal receiver tube and the glass envelope. A representative gas pressure in the annular space 0.0001 mm Hg (0.013 Pa). Getters, which are metallic substances that adsorb gas molecules, are installed in the vacuum space to absorb hydrogen and other gases that permeate into the annular space over time.

Each structure tracks the sun by rotation about a horizontal axis. The drives for the structure are typically a pair of hydraulic rams.

The mirrors are made from a low iron float glass that is silvered on the back and then covered with several protective coatings. The mirrors are formed into a parabolic trough by heating the glass, and then sagging, over parabolic molds in specialty ovens. Ceramic pads are used for mounting the mirrors to the collector structure with a specialty adhesive.

The width of each collector structure ranges from 5 m to 8.2 m, depending on the vendor. A number of collectors are bolted to one another to form a collector assembly. Each assembly has a drive, and the length of the assembly ranges from 100 m to 250 m, again depending on the vendor.

Commercial receivers are available with tube outer diameters of 70, 80, and 90 mm. This results in concentration ratios (collector aperture divided by tube diameter) in the range of 60 to 110. The selected concentration ratio is a function of 1) the optical accuracy of the structure and the mirrors, 2) process temperatures, and 3) allowable fluid velocities in the tubes.

11.5.1 Optical Efficiencies

A representative set of optical efficiencies for the structure and the mirrors is listed in Table 11-2, and a representative set of optical efficiencies for the receiver tube are shown in Table 11-3.

The product of the collector optical efficiency times the receiver optical efficiency is on the order of 0.756. In essence, for every 1,000 photons normally incident on the collector aperture, 756 are absorbed by the receiver tube.

Table 11-2 Representative Collector Optical Performance Factors

Item	Factor
Tracking error and twist	0.990
Geometric accuracy	0.980
Mirror reflectivity	0.940
Mirror cleanliness	0.950
Product	0.866

Table 11-3 Representative Receiver Optical Performance Factors

Item	Factor
Dust on glass envelope	0.980
Bellows shading of tube	0.963
Glass envelope transmissivity	0.963
Tube absorptivity	0.960
Product	0.872

11.5.2 Heat Losses and Thermal Efficiencies

Thermal losses from the receiver tube include a combination of convection, conduction, and radiation heat transfer. A thermal model of the receiver is illustrated in Figure 11-6. The calculation of the thermal losses, which is rather involved, is detailed in a report from NREL [20].

Examples of calculated receiver efficiencies and unit heat losses (W/m), as functions of the absorptivity of the selective surface and the average heat transfer fluid temperature, are shown in Figure 11-7. Due to the very low gas pressure in the annular space between the receiver and the glass envelope, convection losses from the receiver are generally low. As such, the principal loss mechanism is radiation, which is responsible for the parabolic shapes in the efficiency and heat loss curves as functions of the average fluid temperature.

If Therminol is the heat transfer fluid, then the design collector field outlet temperature is 393 °C. With a maximum receiver tube temperature of perhaps 400 °C, the selective surface chosen for the tubes has a

[20] Forristall, R., (National Renewable Energy Laboratory, Golden, Colorado), 'Heat Transfer Analysis and Modeling of a Parabolic Trough Solar Receiver Implemented in Engineering Equation Solver', NREL/TP-550-34169, October 2003

combination of absorptivity of 0.96 and emissivity of 0.09 to 0.095 which maximizes the efficiency for temperatures up to 400 °C [21].

Examples of calculated receiver efficiencies and unit heat losses, as functions of the status of the receiver tube and the wind speed, are shown in Figure 11-8. The values represent receiver tubes for use with Therminol as the heat transfer fluid. The receiver tube can be 1) in an as-new condition with the vacuum intact, 2) in a partially degraded condition with air in the annular space between the tube and the glass envelope (lost vacuum), and 3) in a fully degraded condition with a broken glass envelope.

As noted in Figure 11-8, unit heat losses, at an average tube temperature of 250 °C and over a range of wind speeds, are on the order of the following:

- 150 to 300 W/m with the receiver tubes in an as-new condition
- 300 to 500 W/m with air in the annular space between the tube and the glass envelope
- 1,100 to 2,300 W/m with a broken glass envelope.

[21] 'Optical and thermal characterization of parabolic trough solar receivers, Shandong Huiyin Energy Co., Ltd., 903, Test Report', Foundación CENER-CIEMAT, Departamento de Energía Solar Térmica, Laboratorio de Ensayos Solares Térmicos, Test Report 30.3429.0-2 Annex 6, 31/08/2018

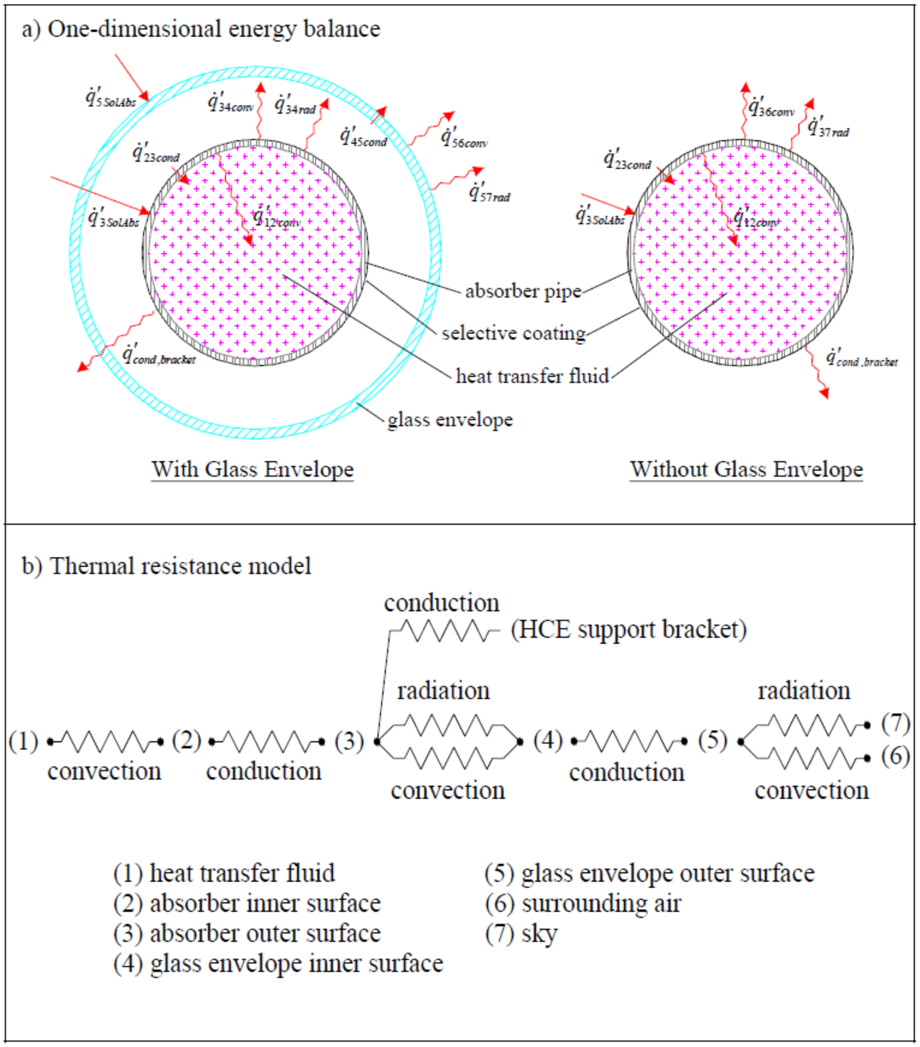

Figure 11-6 Convection, Conduction, and Radiation Heat Exchange in a Receiver Tube

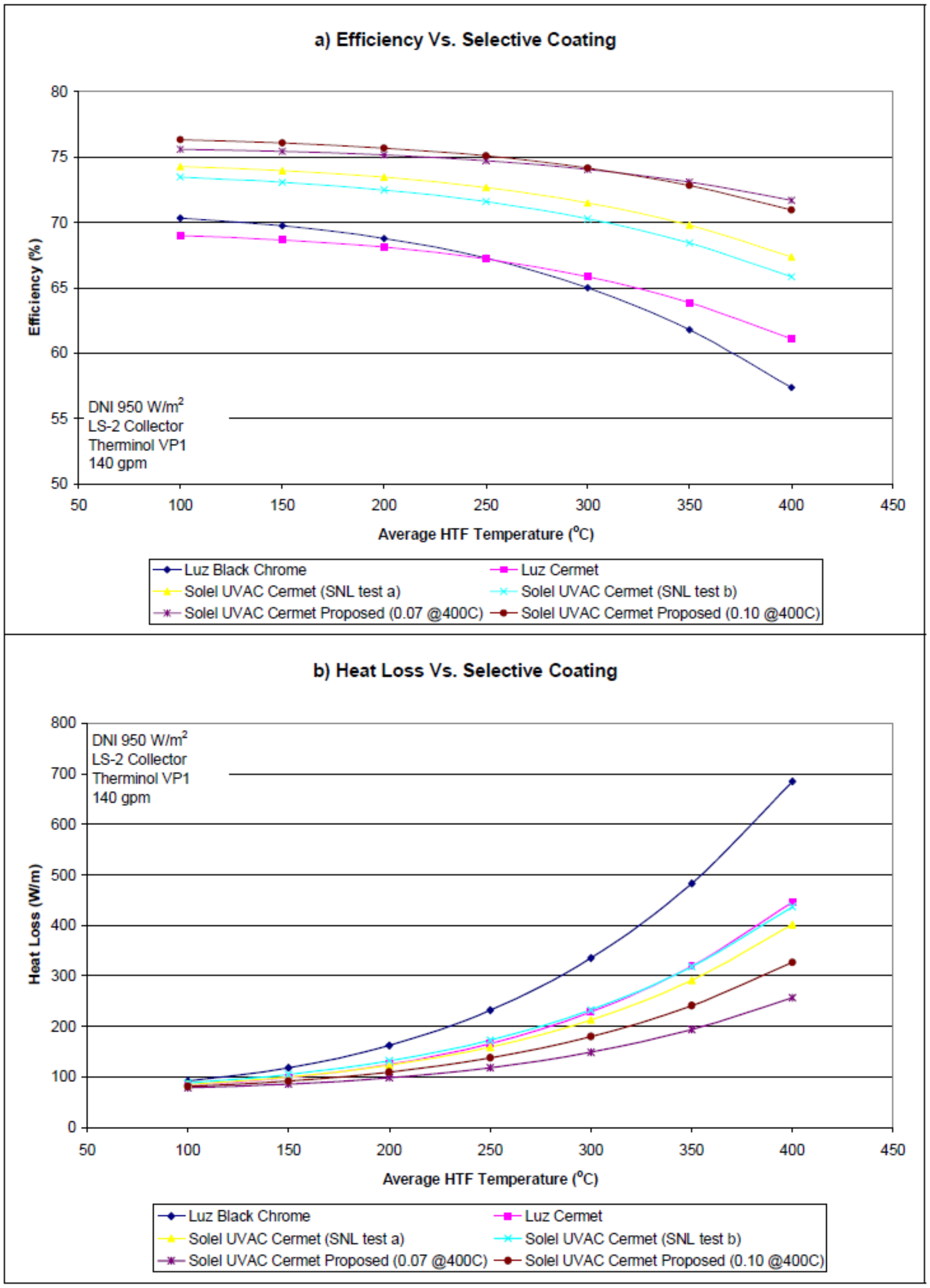

Figure 11-7 Receiver Efficiency and Heat Loss as Function of the Absorber Coating

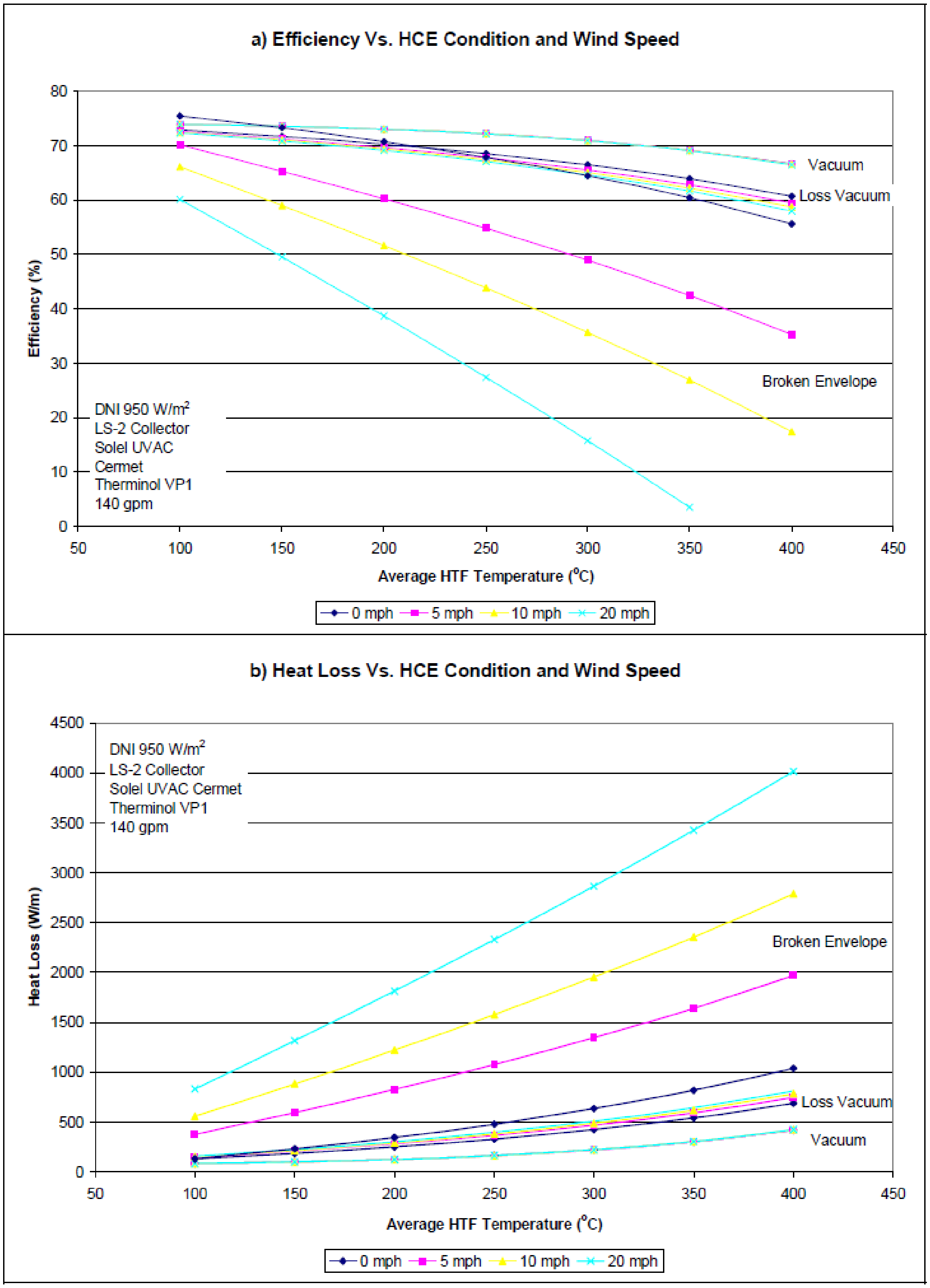

Figure 11-8 Receiver Efficiency and Heat Loss as Function of the Tube Condition

11.5.3 Collector Drives and Controls

The collector assemblies rotate around the horizontal axis to track the sun. The axis can be aligned either North/South or East/West. The latter offers the best annual sunlight-to-thermal efficiency. However, the former is normally adopted for commercial projects, as it offers a better daily efficiency during the important summer months.

The axis of rotation is located at the collector center of mass to minimize the required tracking power. The drive system uses hydraulic rams to position the collector. A closed loop tracking system relies on a sun sensor for the +/- 0.1 degree accuracy required to properly focus the sun on the receiver tubes. The tracking is controlled by a local controller located on each collector assembly. The local controller also monitors the heat transfer fluid temperature and reports operational status, alarms, and diagnostics to the main solar field control computer.

The collector is designed for normal operation in winds up to 40 km/h and somewhat reduced accuracy in winds up to 56 km/h. The assemblies are designed to withstand a maximum of 113 km/h winds in their stow position, with the collectors aimed 30º below the horizon.

11.6 Designs and Rates of Temperature Change

11.6.1 Oil-to-Salt Heat Exchanger

In most commercial projects, there is a direct connection between the hot header returning from the collector field and the hot end of the oil-to-salt heat exchanger. As such, the rates of temperature charge seen in the hot oil header, during both morning startup and cloud transients, are transmitted directly to the oil-to-salt heat exchanger. To a first order, the oil-to-salt heat exchanger must 1) accommodate a rate of temperature change of 10 °C/min, and preferably 12 °C/min, and 2) provide a low cycle fatigue life of at least 30,000 cycles.

Essentially all commercial projects use a shell-and-tube design for the oil-to-salt heat exchanger, including the following geometries:

- U-tube bundle / straight shell, with a single tubesheet. Differential thermal expansion between the tube bundle and the shell is accommodated by bending of the tubes in the U-section of the bundle

- Straight tube / straight shell, with a fixed tubesheet and a floating tubesheet. Differential thermal expansion between the tube bundle and the shell is accommodated by lateral movement of the floating tubesheet.

The most common approach to attaching the tubes to the tubesheet is 1) strength welding the ends of the tubes to the face of the tubesheet, and 2) plastically expanding the tubes into the tubesheet. The principal sealing mechanism between the high pressure fluid on the tube side (Therminol) and the low pressure fluid on the shell side (salt) is the friction connection between the outside of the tubes and the bores in the tubesheet. The strength weld is the backup fluid boundary in the event that the friction connection relaxes.

To provide a log mean temperature difference of 5 to 7 °C at the design point, large heat transfer areas are required. Even using heat exchangers with the largest shell diameters typically available (2.75 m), 4 to 6 heat exchangers in series are typically required. One alternate approach to multiple shells is to use a flat plate heat exchanger. The corrugated plates are arranged in a stack, with alternating layers of oil flowing in one direction and nitrate salt flowing in the other. The edges of the plates are welded to form a pressure boundary, and an external frame supports the hydraulic forces. The advantages of a plate heat exchanger are very large heat transfer areas in a single unit, and low pressure drops for both fluids. The principal liability of the design are marked changes in metal thickness where the thin plates meet the edge welds, and where the edge welds meet the external frame. To control the transient thermal stresses in these regions, the allowable rate of temperature change is limited to 1.5 °C/min. It is very difficult to comply with this rate, particularly if the hot header from the collector field is connected directly to the hot end of the oil-to-salt heat exchanger. At one commercial project, at least two heat exchangers experienced significant cracking after perhaps 6 months of commercial service.

Nonetheless, this is not to imply that a plate heat exchanger cannot be suitable for commercial use. Specifically, the heat exchanger needs to be operated in a manner similar to the steam generator in a central receiver project. This involves the following steps:

- Prior to daily startup, the heat exchanger is maintained at an uniform temperature of 295 °C by establishing a flow of cold salt at the cold end of the heat exchanger. Salt leaving the hot end of the heat exchanger is returned to the cold salt tank.

- Shortly after sunrise, a flow of Therminol from the hot header of the collector field is sent to the hot end of the heat exchanger. However, if the rate of temperature change in the oil in the hot header exceeds 1.5 C/min, then the flow is attemperated by blending cold oil from the discharge of the field circulation pumps with the oil from the hot header at a mixing station upstream of the hot end of the heat exchanger.

- The flow rate of cold salt to the heat exchanger is adjusted such that the temperature of the salt at the hot end of the heat exchanger increases at the allowable rate of 1.5 °C/min.

- If the salt temperature at the hot end of the heat exchanger is below a threshold value, then the salt is returned to the cold tank to prevent a decay in the inventory temperature in the hot tank. Once the threshold temperature is exceeded, salt is directed to the hot tank.

- Cloud transients over the collector field will cause a decay in the temperature of the hot Therminol header returning from the field. The temperature decay will be transmitted to the hot end of the heat exchanger. The flow rate of salt will be adjusted to match the duty on the salt side with the duty on the Therminol side. If the salt temperature at the hot end of the heat exchanger falls below the threshold value, then the salt is returned to the cold tank.

- At the end of the day, two scenarios can arise:

  - The storage system will immediately switch from a charge mode to a discharge mode. In this case, the flows on both the salt side and the oil side are stopped to preserve the normal temperature distribution the heat exchanger. The two fluid flow directions can then be reversed without changing the temperature profile in the heat exchanger.

  - The storage system will switch from a charge mode to a discharge mode after some delay period. The heat exchanger is returned to a uniform temperature of 295 °C by essentially reversing the startup process. At the beginning of the discharge process, the heat exchanger is restarted, but with the fluid flows in the reverse direction. This will require the use of a cold salt / hot salt mixing station upstream of the hot end of the heat exchanger.

11.6.2 Storage Tanks

Preheating with Combustion Gases

For the large storage tanks typical of a commercial project, the allowable rate of temperature change for either the cold tank or the hot tank during preheating is 4.5 °C/min. Nonetheless, the more important temperature limits, in terms of ensuring the low cycle fatigue lives of the tanks, are the following:

- 25 °C between any two points on the wall

- 33 °C between any two points on the floor.

Normal Operation

The allowable rate of temperature change for either the cold tank or the hot tank during normal operation is 1 °C/min. As above, the more important temperature limits are the following:

- 25 °C between any two points on the wall.

- 15 °C between any point on the wall and any point on the floor (or the roof)

- 5 °C between the center of the floor and the perimeter of the floor.

11.7 Efficiencies

In a commercial parabolic trough project, a representative storage tank has a diameter of 40 m and a wall height of 14 m. The active storage mass is on the order of 25,000 tons. With a cold tank temperature of 292 °C, and a hot tank temperature of 386 °C, the nominal storage capacity is 930 MWht.

Representative flux fluxes through the insulation for each tank are 75 W/m2 through the roof, 60 W/m^2 through the wall, and 60 W/m^2 into the foundation. As such, the steady state heat losses from the storage system are about 15 MWht. This results in a daily storage efficiency of approximately (930 MWht - 15 MWht) / 930 MWht = 0.98.

11.8 Materials

11.8.1 Oil-to-Salt Heat Exchanger

If the heat exchanger is a shell-and-tube design, the shell, the channel, and the tubesheet can be fabricated from carbon steel components. A nominal salt-side corrosion allowance is 3 mm (1/8 in.). In contrast, the tube wall thicknesses required to withstand the absolute and the differential fluid pressures are in the range of 1.65 to 2.1 mm. Since it is impractical to add a corrosion allowance of 3 mm to a thin wall tube, a tube material is selected which requires essentially no allowance for corrosion. The candidate materials include various low chromium ferritic steels, or an L grade of a 300-series stainless steel. The industry has often defaulted to stainless tubes as these are available in longer lengths (30 m, 100 ft.) than ferritic tubes.

As discussed in Section 11.2, 6 heat exchangers in series are typically needed to provide the required heating length. The temperature profile along the heating length during a charge cycle is nominally the same as the temperature profile along the heating length during a discharge cycle. As a cost savings measure, some commercial projects have used carbon steel tubes for the 3 heat exchangers at the cold end of the temperature profile, switching to stainless steel tubes for the 3 heat exchangers at the hot end of the temperature profile.

If the heat exchanger is a plate design, the plates are fabricated from stainless steel, and the supporting structure is fabricated from carbon steel.

11.8.2 Storage Tanks

If the project uses the technical or the refined grade of the salt, the chloride content is relatively low, and the corresponding corrosion rates are moderate. As such, carbon steel is a suitable fabrication material for both the cold tank and the hot tank. Nonetheless, it can be noted that stress corrosion cracking of carbon steel has been observed in the tank at one commercial project. The topic is discussed above in Section 5.2.1.

If the project uses the industrial grade of sodium nitrate, rather than the technical or the refined grade, then the chloride content is likely at a level that justifies the use of a low chromium ferritic steel, such as 2¼ - 1 Mo, for the hot salt tank.

11.8.3 Salt Pumps

For both the cold salt and the hot salt pumps, representative materials of construction include the following:

- Motor stand, column, bowls, and suction bell: Carbon steel
- Pump shaft: Low chromium ferritic steel
- Impellers: Martensitic stainless steel
- Shaft bearing: Cast iron.

11.9 Instruments

Recommended approaches for the type and the installation of instruments in salt service are discussed above in Section 9.10.

11.10 Salt Valves

Recommended approaches for the types of valves in salt service are discussed above in Section 9.5.2.

12. Current State of the Art - Inorganic Heat Transfer Fluids

12.1 Introduction

As discussed in Section 10.3, the most likely candidate for the working fluid in a parabolic trough plant with inorganic heat transfer fluids is the 60/40 binary nitrate salt mixture. For the purposes of the discussion below, the working fluid is assumed to be this binary mixture.

Parabolic trough projects using the binary salt working fluid have similar arrangements to central receiver projects using the same binary salt working fluid. The collector system is supplied with cold salt from a cold salt tank. The salt is heated to the nominal hot side temperature, and delivered to a hot salt tank. Salt from the hot tank is supplied to a steam generator, and the salt leaving the steam generator is returned to the cold tank to complete the cycle. The steam generator supplies live steam and reheat steam to a conventional single-reheat Rankine cycle for electric power production.

To date, trough projects using binary salt as the working fluid have been limited to the following:

- An ENEA research facility near Rome, using one-half of a collector loop
- A 9 MWe equivalent trough field built by Enel near Priolo Gargallo, Sicily. The project has a thermal storage capacity of 8 hours, and supplies steam to an existing power plant
- A 1 MWe demonstration project near Massa Martana, Italy, developed by Archimede Energy and Chiyoda. The plant has 5 hours of thermal storage.

No commercial plants have been built to date, for the following reasons:

- A proven means for freeze recovery in the collector field has yet to be demonstrated. Various approaches have been proposed, including permanent impedance heating systems, portable impedance heating systems, and oscillating the collectors into, and out of focus, for limited periods of time. Nonetheless, each of the options have various liabilities, as follows:
    - The permanent systems are expensive, and can increase the unit cost of the collector field by as much as one-third
    - The time required for portable systems to recover from a completely frozen field ranges from days to weeks
    - The optical heating approach requires several hours of uninterrupted radiation, and if not performed correctly, has the potential for permanently damaging the receiver tubes.

- Several years ago, the gross Rankine cycle efficiency of plant using Therminol, and operating with single reheat steam conditions of 100 bar / 370 °C / 370 °C, was a nominal 0.375. Today, gross cycle efficiencies in the range of 0.39 to 0.41 are possible due to improvements in the isentropic expansion efficiencies of steam turbines. The improvement in cycle efficiency has reduced some of the motivation to switch heat transfer fluids.

In general, the increase in project risk of a binary salt heat transfer fluid relative to a Therminol heat transfer fluid, particularly in terms of plant availability, may not offset the benefits of a lower cost for the storage system and a higher efficiency for the Rankine cycle. Since there are no commercial projects using salt as the working fluid, the current state of the art discussed below is somewhat conjectural.

12.2 Process Flow Diagrams

A simplified process flow diagram of a parabolic trough plant using salt as the working fluid is shown in Figure 12-1. Unlike trough projects using Therminol as the working fluid, the storage system does not require an intermediate heat exchanger, and the working fluid in the steam generator is salt. A simplified flow diagram on the water/steam side of the steam generator is shown in Figure 12-2.

12.3 Collector System

The collector system in a trough project using salt as the working fluid is nominally the same as the collector system in a trough project using Therminol as the working fluid, as described in Section 11.5.

12.3.1 Absorber Selective Surface

If nitrate salt is the heat transfer fluid, then the design collector field outlet temperature is in the range of 500 to 540 °C. With a maximum receiver tube temperature of perhaps 560 °C, a different selective surface is chosen compared to system based on Therminol. The selective surface for use with nitrate salt sacrifices a portion of the absorptivity (0.92, rather than 0.96) in favor a lower emissivity (0.07, rather than 0.095) at 400 °C. An example of emissivity values, over a range of absorber tube temperatures, for Therminol and nitrate salt applications is shown in Figure 12-3 [22].

[22] Ruegamer, T., et al, (SCHOTT Solar CSP GmgH, Mitterteich, Germany), 'Molten Salt for Parabolic Trough Applications: System Simulation and Scale Effects', SolarPACES 2013, Las Vegas, Nevada

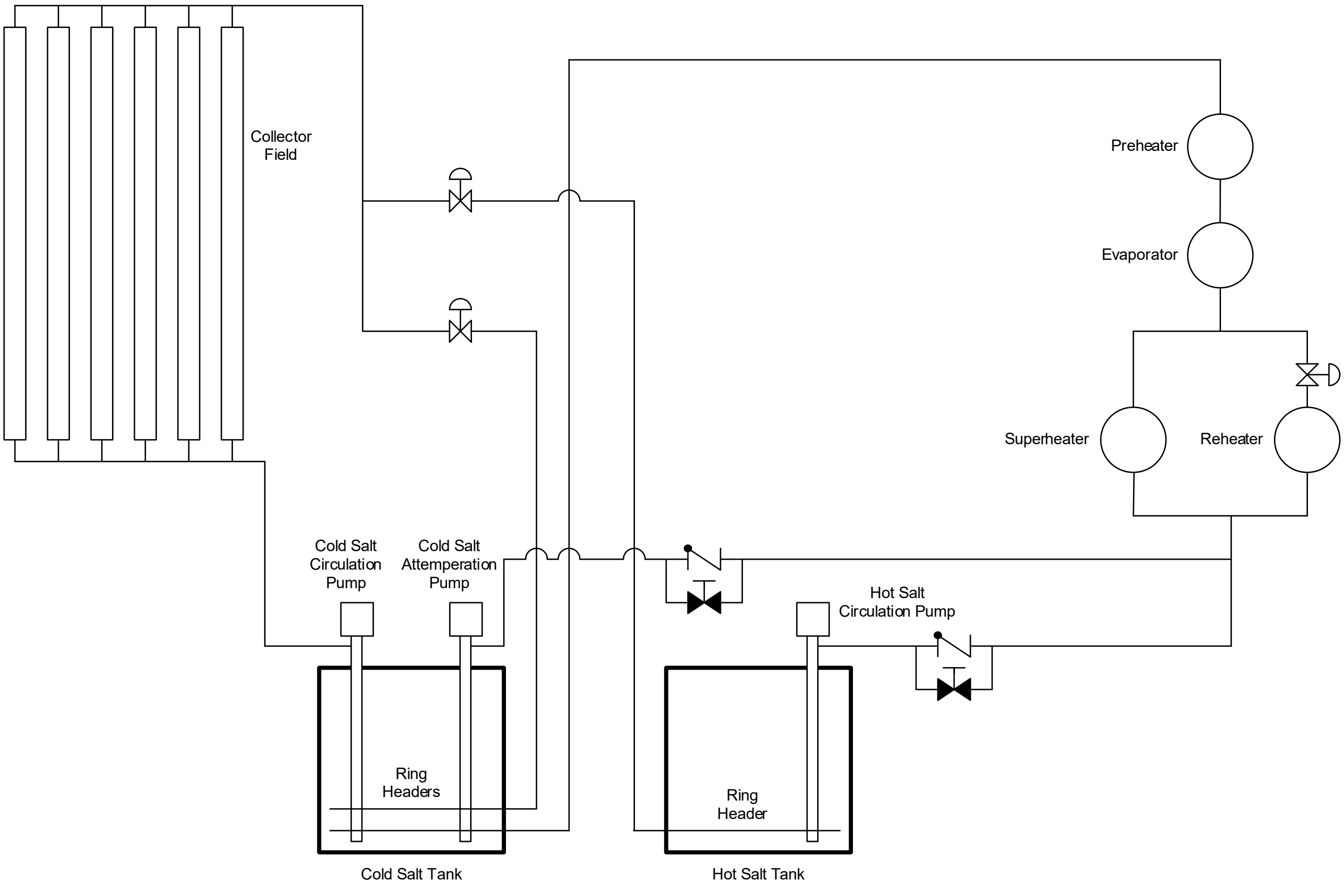

Figure 12-1 Simplified Process Flow Diagram of a Trough Project using Salt as the Working Fluid

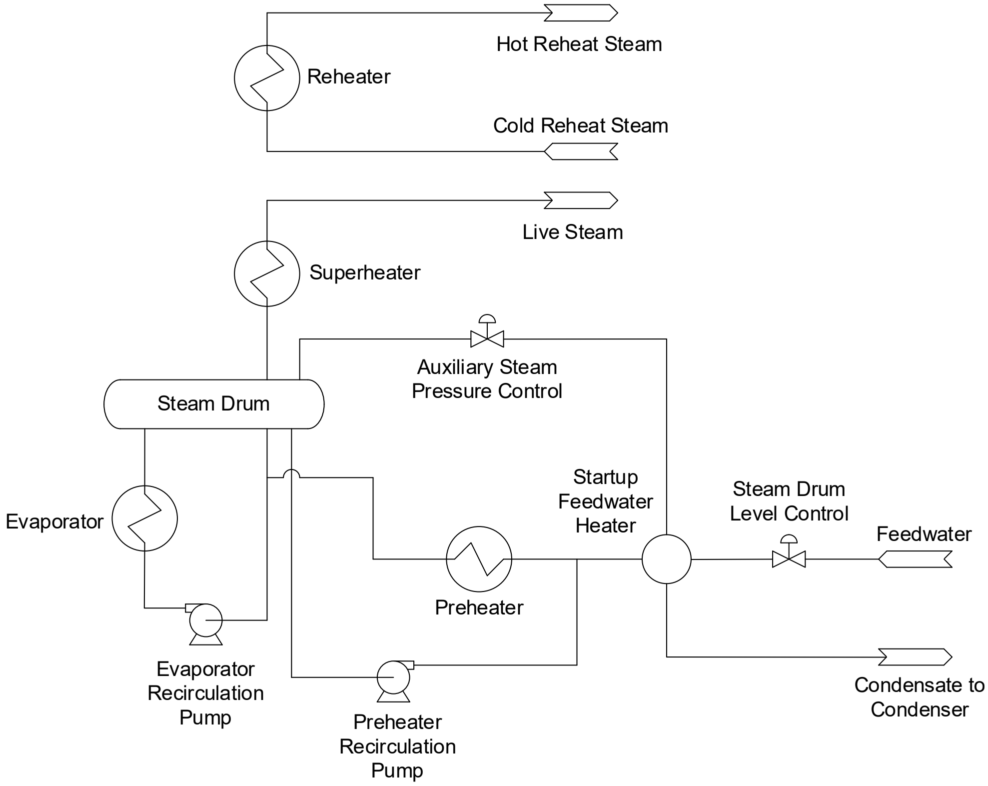

Figure 12-2 Simplified Process Flow Diagram for Water/Steam Side of Steam Generator

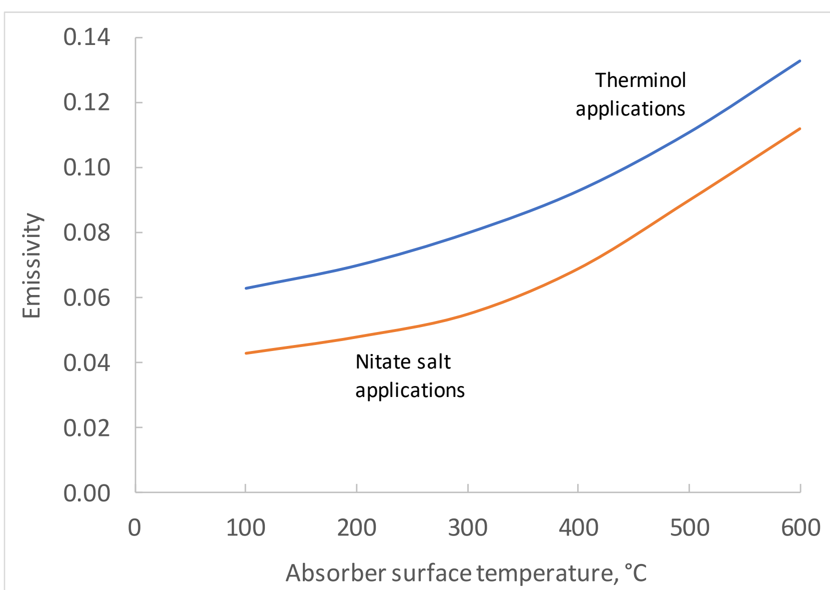

Figure 12-3 Emissivity as a Function of Absorber Surface Temperature

12.3.2 Impedance Heating of Receiver Tubes

As discussed in Section 10.9.2, the receiver tubes are preheated using portable impedance heating systems. Since the heat losses from a receiver tube are a function of the status of the receiver tube (intact vacuum, lost vacuum, or broken glass envelope), the power required by the impedance heating system is also a function of the status of the receiver tube. Using the unit heat loss estimates shown in Figure 11-8, at a wind speed of 10 miles per hour, the required unit impedance heating capacities are listed in Table 12-1.

Table 12-1 Impedance Heating Requirements as a Function of Receiver Status

Receiver Status	Impedance Duty, W/m	Impedance Current, Amp
New, with vacuum intact	170	440
Air in the annular space	525	770
Broken glass envelope	1,900	1,470

Due to the large electric currents required for an impedance system, the sizes of the transformers, the switchgear, and the power cables are also large. This, in turn, leads to expensive heating systems. The minimum required unit heating duty is on the order of 200 W/m, which is based on the receiver tubes in an as-new condition. However, once in commercial service, the thermal efficiency of the receiver tubes will start to decay, largely as the result of gases entering the annular space between the tubes and the glass envelope. The gases increase the convection heat transfer, and degrade the quality of the selective surface on the receiver tube.

Ideally, the impedance heating system would have the capacity to preheat a tube with air in the annular space; however, the associated costs may be unaffordable. The solution for preheating a receiver which has started to lose its vacuum is to install a temporary insulated cover on those tubes which have a heat loss greater than, for example, 250 to 300 W/m. This approach, in turn, requires the identification of those receiver tubes which have a heat loss greater than 250 W/m. This can be performed by a drone using an infrared camera. The camera can identify those receiver tubes which have outer glass temperatures higher than design values.

For receiver tubes with broken glass envelopes, it is impractical to overcome the heat losses using an impedance system. To preheat these tubes, a temporary insulated cover must be used.

12.3.3 Field Designs using Therminol and Nitrate Salt

In a commercial project, there are several km of receiver tubes in the collector assemblies, and several km of piping in the field supply and return headers. Further, the collector assemblies and the field piping are all at nominally the same elevation. As such, once the solar field is filled with the heat transfer fluid, it becomes impractical to drain the solar field.

The principal differences between collector fields using Therminol and nitrate salt as the working fluid include the following:

- Flex hose assemblies (Section 10.9.1) replace the ball joint assemblies to accommodate 1) the thermal expansion of the receiver tubes, and 2) the relative rotation between the collector structure and the fixed field piping.
- An impedance heating system (Section 10.9.2) is used to preheat the receiver tubes prior to filling with salt, and to recover from a situation in which the salt freezes in the receiver tubes
- The field piping uses resistance heat tracing (Section 10.9.3) to preheat the piping prior to filling with salt, for freeze protection under conditions in which the salt is not circulating, and to recover from a freeze event.

Maintenance of Therminol Loops

For maintenance of a Therminol collector loop, the following steps are taken:

- The isolation valves at the inlet and at the outlet of the loop are closed
- A maintenance truck, with a vacuum tank, is located next to the loop
- A temporary piping connection is made between the outlet of the loop and the vacuum tank
- A vent valve at the inlet of the loop is opened, and air entering the vent valve pushes Therminol into the vacuum tank.

To refill the loop, the process is as follows:

- The vacuum tank is vented to the atmosphere
- A temporary piping connection is made between the inlet of the loop and a Therminol pump located below the vacuum tank
- A temporary piping connection is made between the outlet of the loop and the top of the vacuum tank

- Therminol is circulated from the tank through the loop until all of the air has been removed from the loop.

Maintenance of Nitrate Salt Loops

Draining and refilling a collector loop using nitrate salt is performed in a manner similar to that for a collector loop using Therminol. However, with nitrate salt, the loop will be subject to freezing once the isolation valves are closed. As such, the portable impedance heating system will need to be in place prior to draining the loop. Also, the vacuum tank will need to have some form of electric heating to maintain the salt at temperatures above the freezing point.

12.3.4 Freeze Protection

During all periods in which the collector field is not in normal operation, freeze protection for the collector field (receiver tubes, flex hose assemblies, and field piping) is provided by circulating cold salt from the cold tank, through the collector field, and back to the cold tank. The inventory temperature in the cold tank is maintained above the minimum set point (275 °C) by electric salt heaters. A representative commercial 10 MWe heater[23] is shown in Figure 12-4.

Figure 12-4 10 MWe Electric Salt Heater

[23] https://www.chomalox.com/-/media/files/literature/en-us/lit-pq132-directconnect.pdf?la=en

The heater consists of a series of tubes inserted into a shell, similar to a shell-and-tube heat exchanger. Inside each tube is an electric heating element, and the space between the element and the inside of the tube is filled with a ceramic packing. Heat transfer from the element to the tube is by conduction. The tubes are welded to a tubesheet. The salt flows on the shell side of the heat exchanger; as such, the salt does not come into contact with the heating elements.

The electric heaters, which operate at 4.16 kV, are supplied by the plant medium voltage distribution system or by the standby Diesel generator.

The capacity of the electric heaters is based on a zero thermal input from the field piping heat trace system or from the impedance heating systems. A 10 percent factor is added for thermal losses associated with lost vacuum and broken glass envelopes on the receiver tubes.

12.3.5 Field Piping Heat Trace Diagnostics

Regarding the heat tracing of the field piping, a commercial project will have several kilometers of salt piping. If a heat trace cable fails, then the DCS is able to locate the failed cable, under one of the following conditions:

- If the supply current drops to zero, then the cable has developed an open circuit
- If the cable develops a short circuit, then the breaker protecting the circuit will open
- If the duty cycles of the cables in Zone 1 are consistently higher than the duty cycles of the cables in a similar Zone 2, then one of the cables in Zone 1 may have developed an open circuit.

A potentially larger problem exists if salt circulation is stopped for some period of time, and a local defect in the insulation causes the salt to freeze. Identifying the location of the frozen zone may be difficult, as the heat trace system is still working perfectly. One approach is to install a fiber optic cable within the pipe insulation. The refraction properties of the glass are a function of the optic cable temperature. Measuring the transit time of a light pulse from a source to the low temperature zone, and then back from the low temperature zone to the source, can be used to indicate the distance to the low temperature zone.

12.4 Thermal Storage System

The thermal storage system in a trough project using salt as the working fluid is essentially identical to the thermal storage system in a central receiver project using salt as the working fluid.

The design of the storage system is discussed in Section 6.10 and Section 7.3 of Volume 2 - Central Receiver Specifications.

12.5 Steam Generation System

The steam generation system in a trough project using salt as the working fluid is essentially identical to the steam generation system in a central receiver project using salt as the working fluid.

The design of the steam generation system is discussed in Section 6.11 and Section 7.4 of Volume 2 - Central Receiver Specifications.

12.6 General Salt System Equipment

The general salt system equipment includes the piping, flanges, fittings, supports, anchors, guides, and valves.

The general salt equipment in a trough project using salt as the working fluid is essentially identical to the general salt equipment in a central receiver project using salt as the working fluid.

The equipment is described in Section 6.5 of Volume 2 - Central Receiver Specifications.

12.7 Instruments

The salt instruments in a trough project using salt as the working fluid are essentially identical to the salt instruments in a central receiver project using salt as the working fluid.

The instrument concepts are discussed in Section 6.6 of Volume 2 - Central Receiver Specifications.

12.8 Heat Tracing System

The heat tracing system in a trough project using salt as the working fluid is essentially identical to the heat tracing system in a central receiver project using salt as the working fluid.

The design of the heat tracing system is discussed in Section 6.7 of Volume 2 - Central Receiver Specifications.

12.9 Salt Pumps

The salt pumps in a trough project using salt as the working fluid are essentially identical to the salt pumps in a central receiver project using salt as the working fluid.

The design of the salt pumps is discussed in Section 6.8 and Section 7.2 of Volume 2 - Central Receiver Specifications.

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 44 mole percent sodium nitrate ($NaNO_3$), and 56 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.

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

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

Isotropic compressibility ($NaNO_3$) at the melting point $2 * 10^{-10}$ 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