USA Trough Initiative

Thermal Storage Oil-to-Salt Heat Exchanger Design and Safety Analysis

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1 BACKGROUND

Between 1984 and 1990, a total of nine Solar Electric Generating Station solar power plants were built in the southern California desert. Each plant used parabolic trough solar collectors to heat either a mineral oil or a synthetic heat transport oil. Thermal energy in the oil was used to generate steam, and the steam drove a conventional Rankine cycle power plant. The first plant employed a thermal storage system. The heat transport fluid was a mineral oil, and the combination of low vapor pressure and relatively low unit cost made a thermal storage system economic. In contrast, the collector systems in the second through ninth plants operated at higher temperatures to improve the Rankine cycle efficiency. The higher temperatures required the use of a synthetic oil, and the combination of high vapor pressure and high cost for the oil precluded the use of a thermal storage system.

The identification of an economic thermal storage system would broaden the market potential for future parabolic trough power plants. One concept under consideration was use of nitrate salt for the storage medium. The salt offered a favorable combination of very low vapor pressure, high density, reasonable specific heat, and low cost. Thermal energy from the collector field would be transferred into the storage system through an oil-to-salt heat exchanger; energy could be transferred from the system through a nitrate salt steam generator, or by reversing the flows in the oil-to-salt heat exchanger and driving an oil steam generator.

Several thermal storage concepts for use with the current generation of Rankine cycle power plants were evaluated in Task 3 of the USA Trough Initiative project (Reference 1). The preferred design consisted of the following equipment: a hot nitrate salt storage tank; a cold nitrate salt storage tank; an oil-to-salt heat exchanger; and nitrate salt circulation pumps. A schematic flow diagram of the plant is shown in Figure 1. During daytime operation, the heat transport fluid from the collector field flowed, in parallel, to the oil steam generator and to the oil-to-salt heat exchanger of the thermal storage system. Nitrate salt was pumped from the cold storage tank, through the oil-to-salt heat exchanger, and then to the hot storage tank. During cloudy weather or evening operation, nitrate salt was pumped from the hot storage tank, through the (now) salt-to-oil heat exchanger, and then the cold storage tank.

Probable candidates for the nitrate salt were one of the following:

- Binary salt, which was a mixture of 60 percent by weight sodium nitrate and 40 percent potassium nitrate

- Tertiary salt, which was a nominal mixture of 15 percent by weight sodium nitrate, 42 percent calcium nitrate, and 43 percent potassium nitrate.

Both salts were classified on Material Safety and Data Sheets as a Class 1 oxidizers. The collector field heat transport oil was either Dowtherm A from Dow Chemical Company or VP-1 from Solutia, Inc. The oil was a synthetic organic fluid mixture of 73.5 percent diphenyl oxide and 26.5 percent biphenyl.

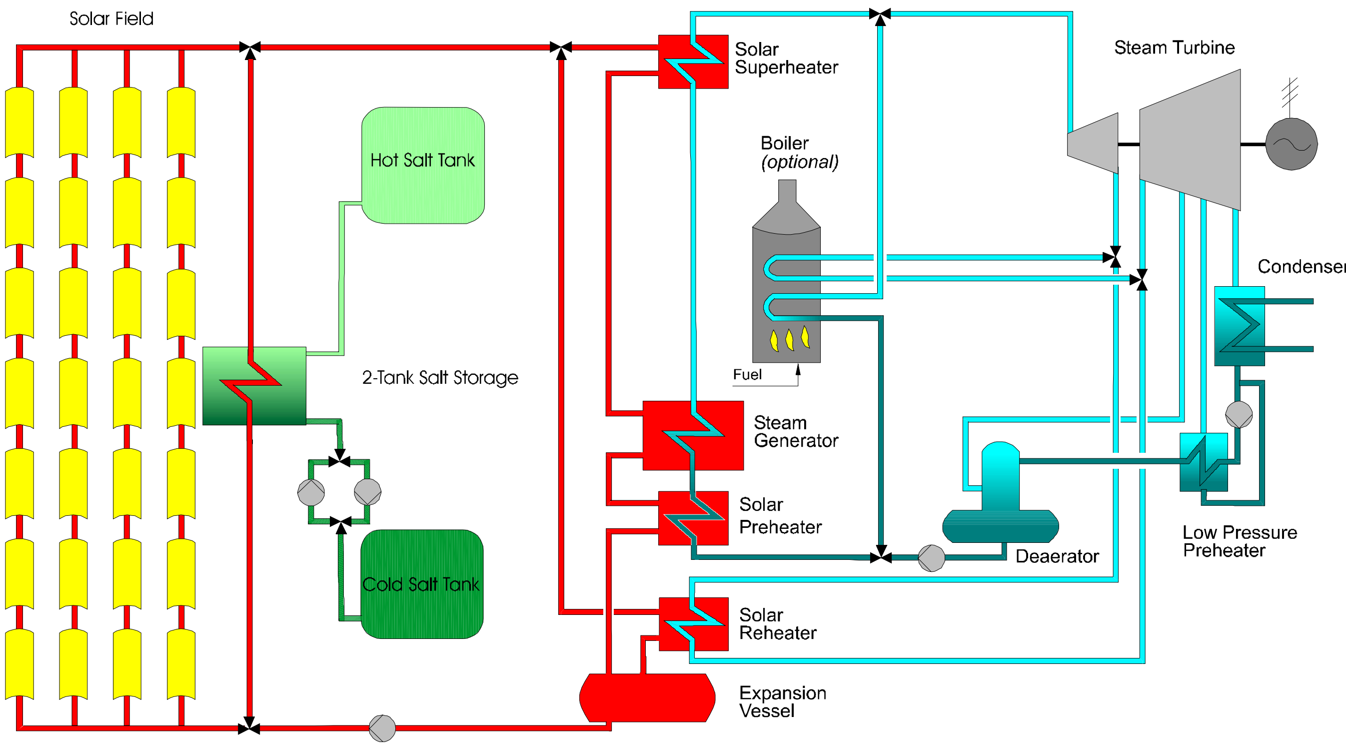


Figure 1 Parabolic Trough Power Plant with Hot and Cold Tank Thermal Storage System And Oil Steam Generator

The oil-to-salt heat exchanger operated at temperatures in the range of 400ºF to 750ºF. A leak in the heat exchanger would cause the oil and salt to mix, and a combustible mixture may form. Adequate safety provisions must be provided in the design of the heat exchanger and thermal storage system to ensure the safety of the plant personnel and to minimize equipment damage in the event of a leak.

2 STUDY OBJECTIVE AND APPROACH

2.1 Objective

The objectives of the study were as follows: 1) develop a conceptual design of an oil-to-salt heat exchanger and associated safety equipment; 2) conduct a failure modes and effects analysis of the heat exchanger and safety equipment to confirm the adequacy of the design; and 3) develop a budgetary estimate for the heat exchanger and safety equipment to support future system demonstration planning activities by the Department of Energy.

2.2 Approach

The work consisted of the following activities:

- Review data from Sandia National Laboratories and fluid suppliers on nitrate salt - hydrocarbon reactions

- Review current industrial practice for the design of heat exchangers and safety systems handling hazardous materials

- Develop a conceptual design of an oil-to-salt heat exchanger and associated safety equipment

- Conduct a failure modes and effects analysis of the heat exchanger and safety equipment

- Develop a budgetary estimate for the heat exchanger and safety equipment.

The evaluations were conducted for a Rankine cycle power plant with the following characteristics:

- Gross electric output of 88 MWe

- Gross Rankine cycle efficiency of 37.5 percent

- Steam generator thermal rating of 235 MWt

- Solar multiple of 1.15 and a thermal storage capacity of 470 MWht. The solar multiple was defined as the ratio of collector field rating to steam generator rating. The storage capacity was sufficient to operate the turbine-generator at full load for 2 hours.

With a solar multiple of 1.15, the duty of the oil-to-salt heat exchanger during thermal storage charging was 35 MWt. During thermal storage discharging, the duty of the (now) salt-to-oil heat exchanger was 235 MWt.

3 CHEMICAL REVIEW AND HEAT EXCHANGER DESIGN

The process and chemical review was based on the use of a conventional shell and tube heat exchanger to transfer energy back and forth between the heat transport fluid and the nitrate salt. Other potential heat exchanger designs included the plate and frame, and the spiral tube and shell. However, a shell and tube design was selected based on a combination of the following requirements: large heat transfer duty; fluid temperatures greater than 250 ºC (500 ºF); low potential for surface fouling; approach temperatures between the heat transport fluid and nitrate salt of less than 10 ºC (20 ºF); nominal fluid pressures greater than 10 bar (150 lb_f/in^2); demonstrated reliability in hazardous refinery applications; and competitive prices from a number of commercial fabricators.

3.1 Chemical Review

3.1.1 <u>Heat transport fluids</u>

The heat transport fluid in the collector system was either Dowtherm A or Therminol VP-1. Both fluids are synthetic organic oils, with nominal compositions of 75 percent by weight diphenyl oxide/ether and 25 percent biphenyl.

Both diphenyl oxide/ether and biphenyl have flash points over 113 ºC (235 ºF). Since the flash points are greater than 38 ºC (100 ºF), the chemicals are classified by the National Fire Protection Agency as combustible liquids. The fluids are classified in the "0 Group" on the Reactivity (Instability) Hazards Ratings of the National Fire Protection Agency. As such, the materials are not reactive with water, and are normally stable even under fire exposure conditions. While it did take "very high" temperatures to initiate the combustion of the heat transport fluid, the Material Safety Data Sheet recommends that exposure to highly oxidizing materials be avoided.

Both diphenyl oxide and biphenyl have auto ignition temperatures above 538 ºC (1,000 ºF), and the Material Safety Data Sheet for Therminol VP-1 lists an auto ignition temperature of 613 ºC (1,135 ºF). Thus, the fluid has a Flammability Hazards Rating of "1" from the National Fire Protection Agency, indicating the chemical must be preheated before ignition can occur.

3.1.2 <u>Nitrate Salts</u>

The proposed salts contained sodium nitrate and potassium nitrate, and in the case of the Hitec XL, calcium nitrate. Calcium nitrate is the least reactive of the three salts, but potassium nitrate and sodium nitrate are oxidizing agents. When the latter two salts are in contact with organic materials at temperatures above the ignition temperature, reactions may proceed quickly enough to cause ignition, violent combustion, or explosion. If combustion is established, oxides of nitrogen may also be formed. In addition, there are numerous documents in the chemical literature which describe explosive mixtures formed between nitrate salts and inorganic chemicals, such as antimony trisulfide, arsenic disulfide, sodium acetate, and sodium hypophosphite.

3.1.2 Reaction Chemistry

One of the common methods in the chemical industry for assessing chemical hazards is the use of an adiabatic reaction calorimeter. To the best of our knowledge, no public report or scientific study exists on the mixing of nitrate salt with Dowtherm A or Therminol VP-1 at elevated temperatures in a calorimeter. Contacts were made with both of the heat transport fluid manufacturers seeking reaction data, but neither vendor had conducted the reaction tests. If the reaction data are not available from the manufacturer, the owner or user of the process is usually the one to conduct the reaction experiments. However, the consensus from both companies was to avoid the exposure of the heat transport fluid to materials which are "highly oxidizing".

The most relevant reaction data are probably from a molten salt safety study conducted by Sandia National Laboratories in 1980 (Reference 2). Liquid gasoline was introduced into an inventory of nitrate salt at a temperature of 600 ºC (1,110 ºF). The hydrocarbons vaporized when exposed to the nitrate salt, and burned at the surface of the inventory when exposed to ambient air. However, the hydrocarbons did not react with the nitrate salt prior to exposure to the atmosphere. In other words, a temperature of 600 ºC (1,110 ºF) was not high enough to initiate a theoretical reduction reaction in which an oxygen atom was removed from a nitrate molecule.

In the event of a tube or weld rupture in the oil-to-salt heat exchanger, the heat transport fluid would mix with nitrate salt and a portion of the heat transport fluid would vaporize. However, a combustion reaction was believed to very unlikely, for the following reasons:

- Therminol VP-1 and Dowtherm A both have flammability ratings of "1", while gasoline has a rating of "3". As a result, it was highly unlikely that either of the heat transport fluids would have a more energetic reaction with nitrate salt than gasoline.

- The highest temperature in the oil-to-salt heat exchanger was 390 ºC (735 ºF), which was 210 ºC (365 ºF) below the exposure tests conducted by Sandia, and 220 ºC (400 ºF) below the auto ignition temperature of the heat transport fluid.

- Oxygen was not present in the heat exchanger.

Although the potential for a combustion reaction following a heat exchanger leak was judged to be low, fabrication and examination techniques for the heat exchanger should be selected which minimize the potential for a leak. The thermal storage system should also be designed to prevent the accumulation of combustible mixtures. For example, Therminol or Dowtherm vapors can accumulate in the ullage space of the nitrate salt storage tanks following a tube leak. Safety issues associated with the design of the heat exchanger and the storage tanks are discussed in the following section.

3.2 Heat Exchanger Design

The vapor pressure of the heat transport fluid at the peak operating temperature of 390 ºC (735 ºF) is approximately 3 bar (45 lb_f/in^2). To ensure single phase flow throughout the collector field of a solar power plant, the nominal operating pressure of the heat transport fluid is about 15 bar (250 lb_f/in^2). Thus, the oil side of the oil-to-salt heat exchanger would also operate at a nominal pressure of 15 bar.

In contrast, the vapor pressure of the nitrate salt is on the order of 10 Pa (0.025 lb_f/in^2). Consequently, the thermal storage tanks would operate at atmospheric pressure, and the fluid pressure in the salt side of the heat exchanger needs to be only that required to overcome the pressure losses in the heat exchanger and the nitrate salt piping.

In terms of the capital cost of the heat exchanger, it is generally preferable to place the high pressure fluid on the tube side rather than the shell side; constraining pressure in a small diameter tube is usually less expensive than constraining pressure in a large diameter shell. From a process safety standpoint, it is also preferable to place the fluid with the highest chemical reactivity on the tube side; a failure of both the tube and the shell must occur before the fluid is exposed to the atmosphere. Thus, for reasons of both economy and safety, the heat transport fluid was selected as the tube side fluid and the nitrate salt was selected as the shell side fluid.

Once the tube and shell fluids had been determined, a heat exchanger configuration must be selected. The following Tubular Equipment Manufacturers Association designs, each suitable for daily cycling service, were evaluated:

- Type “U”, with a fixed tubesheet, a longitudinal shell baffle, and U-tubes. Differential thermal expansion between the shell and tubes was accommodated by 1) an absence of mechanical connections between the shell and the tube bundle, and 2) bending in the tubes.

- Type “S” or “T”, with a fixed tubesheet, straight tubes, and a floating tubesheet. Differential thermal expansion was accommodated by movement of the floating tubesheet.

- Type “E”, with a fixed tubesheet, straight tubes, and an expansion joint in the shell. Differential thermal expansion was accommodated by bending in the convolutions of the expansion joint.

The Type “S” and “T” designs were eliminated from further consideration because a seal material compatible with both the heat transport fluid and nitrate salt has yet to be identified. The Type “U” design was selected in preference to the Type “E” because the cost of fabricating the U-bends in the tubes was believed to be less expensive than fabricating a bellows for the shell.

To help provide a reliable and trouble free heat exchanger, the following features were incorporated in the design:

- Seamless, single piece tubes were specified. A seamless tube avoids the longitudinal joint in a welded tube, and a single piece tube avoids the butt weld joining two shorter tubes.

- Failure of the tube-to-tubesheet joint can be common in cyclic service. To minimize the potential for leaks after several years of operation, the tube-to-tubesheet joint consisted of rolling the tube into the tubesheet, and then seal welding the end of the tube to the face of the tubesheet.

- The joint between the shell and the tubesheet was welded, rather than flanged.

- Ruptures at seam areas due to defective welds or corrosion are common failures. In addition to the standard shell and channel hydraulic pressure test at 150 percent of the maximum allowable working pressures, all of the shell and channel welds were radiographed, and all of the tube-to-tubesheet joints received a dye penetrant examination.

- Corrosion allowances of 3 mm (1/8 in.) were provided on both the tube and shell sides of the heat exchanger.

Alternate designs, which could provide a small increase in reliability and safety, were also considered. In the first alternate, the heat exchanger was fabricated with two adjacent tubesheets, which were separated by a small gap. The tubes would be rolled and seal welded to both tubesheets. Thus, a failure of an outer tube-to-tubesheet joint would cause the tube side fluid to leak into the inter-tubesheet space. Similarly, a failure of the inner tube-to-tubesheet joint would cause the shell side fluid to leak into the space. Pressure measurements in the gap would signal a heat exchanger leak, and repairs could be initiated prior to a chemical reaction. In the second alternate, the heat exchanger was fabricated with concentric tubes and two tubesheets. The inner tube would be rolled and seal welded to the outer tubesheet, and the outer tubes would be rolled and seal welded to the inner tubesheet. Thus, a failure of either a tube or a tube-to-tubesheet joint would be signaled by an increase in pressure between the two tubesheets. However, the single tubesheet, single tube design was selected in preference to the alternate designs for two reasons. First, the reliability of the design has been demonstrated in numerous chemical and power plant applications, and second, the effects of a tube leak were not particularly severe. In particular, the additional features of the double tubesheet, double tube design did not justify an increase in heat transfer area of at least two and an increase in price of at least four.

3.3 Process Safety Features

Should a tube leak occur in the heat exchanger, several process safety features were available for identifying the failure and accommodating the effects, as follows:

<u>Heat Exchanger Flow, Temperature, and Pressure Monitoring</u> Continuous monitoring of the heat transport fluid and nitrate salt flow rates, temperatures, and pressures at the inlets and outlets of the heat exchangers could signal a tube failure. For example, Therminol or Dowtherm vaporizing in the nitrate salt inventory might produce one or more of the following effects: 1) a rise in the shell pressure and a decrease in the channel pressure; 2) a decrease in the inlet, and an increase in the outlet, salt flow rates; 3) an increase in the inlet, and a decrease in the outlet, heat transport fluid flow rates; and 4) a decrease in the shell temperature. For these approaches to be effective, the instruments must be well calibrated and demonstrate consistent accuracy.

<u>Heat Exchanger Isolation</u> If abnormal flow, temperature, or pressure measurements were detected, ‘stop’ signals could be sent to both the heat transport fluid pumps and the nitrate salt pumps. In addition, ‘close’ signals could be sent to automatic isolation valves at the heat exchanger inlets and outlets. Each of these actions would limit the quantity of fluids which could mix, and would limit the quantity of heat transport

fluid vapor which might accumulate in the ullage space of the thermal storage tanks.

Shell Overpressure Protection   The design pressure for the shell side of the heat exchanger was less than the design pressure for the tube side. To protect the shell from excessive pressures due to a tube leak, a pressure relief valve or rupture disc must be provided. The discharge from the relief valve or rupture disc could be directed to a stack to ensure that combustible vapors did not accumulate near plant equipment.

Nitrate Salt Storage Tank Ullage Gas Selection   To ensure that combustible mixtures could not form in the thermal storage tanks, an inert ullage gas could be used. The ullage gas of choice would be nitrogen, for the following reasons: bulk cryogenic nitrogen is available from a number of competitive suppliers; the gas is inexpensive; and nitrogen does not react with either nitrate salts or the heat transport fluids.

The choice of process safety features depends on the results of the failure modes analysis, discussed below in Section 5.

4 CONCEPTUAL DESIGN

The conceptual design for the oil-to-salt heat exchanger and thermal storage system included the following activities: identification of the preferred fluid temperatures; selection of the heat exchanger flow arrangement; calculation of the thermal storage tank dimensions; and calculation of the nitrate salt pump capacities.

4.1 Preferred System Temperatures

The optimum nitrate salt and oil temperatures in the storage system were a function of the following competing effects:

- Live and reheat steam temperatures should be as high as practical to provide a high Rankine cycle efficiency

- The approach temperatures in the oil-to-salt heat exchanger and steam generator should be as small as possible to increase the temperature difference between the cold and hot storage tanks, and thereby minimize the cost of the storage system

- The heat exchanger approach temperatures should be as large as practical to increase the temperature differences between the working fluids, and thereby minimize the cost of the heat exchangers.

The optimum temperatures were selected by calculating heat exchanger surface areas and thermal storage inventory quantities for a range of approach temperatures in the oil-to-salt heat exchanger and the steam generator, and then selecting the system with the lowest capital cost. An Excel spreadsheet model analyzed the system performance through the following steps:

- Live and reheat steam temperatures were set to 371 ºC (700 ºF), and the final feedwater temperature was set to 218 ºC (425 ºF).

- Trial approach temperatures were selected for both the hot end and cold end of the oil-to-salt heat exchanger. The solar multiple of the plant under consideration was less than 2.0; thus, the duty of the oil-to-salt heat exchanger during thermal storage discharge was the design case.

- The hot salt tank temperature was set equal to the live steam temperature plus the approach temperature at the hot end of the oil-to-salt heat exchanger. An enthalpy and mass balance was performed on the steam generator, which calculated an oil temperature at the exit from the preheater consistent with a trial pinch point temperature for the evaporator. From the oil and nitrate salt flow rates, tube side and shell side heat transfer coefficients were calculated for the reheater, superheater, evaporator, and preheater. Surface areas for each heat exchanger were calculated using the following expression:

$$Area, m^2 = \frac{Q, W}{(U_{overall}, W/m^2 - C)(Log\ mean\ temperature\ difference, C)}$$

The heat transfer area in the reheater was selected such that the oil exit temperature was equal to the preheater oil exit temperature.

- The cold salt tank temperature was set equal to the steam generator oil exit temperature plus the approach temperature at the cold end of the oil-to-salt heat exchanger. An enthalpy and mass balance was performed on the oil-to-salt heat exchanger, from which were calculated the following: oil and nitrate salt flow rates; tube side and shell side heat transfer coefficients; and a heat transfer area.

- The part load performance of the oil-to-salt heat exchanger during thermal storage charging was estimated by calculating the following: oil and nitrate salt flow rates; tube and shell side heat transfer coefficients; and hot and cold end approach temperatures. The hot end approach temperature, together with the nitrate salt exit temperature, defined the oil inlet temperature during storage charging. If the oil inlet temperature did not equal the desired collector field outlet temperature of 391 ºC (735 ºF), the trial hot salt tank temperature was revised up or down as necessary, and the heat exchanger calculations were repeated.

An example of the Excel model output, showing the temperature distributions within the thermal storage and steam generator systems, is shown in Figure 4. The optimum hot end and cold end approach temperatures for the oil-to-salt heat exchanger were approximately 8.5 ºC (15 ºF), and the optimum pinch point temperature for the steam generator was about 4.5 ºC (8 ºF).

4.1 Oil-to-Salt Heat Exchanger

A preliminary design for the oil-to-salt heat exchanger were prepared by an exchanger consultant. A heat exchanger data sheet for the selected configuration is shown in Figure 5 and an elevation view is shown in Figure 6.

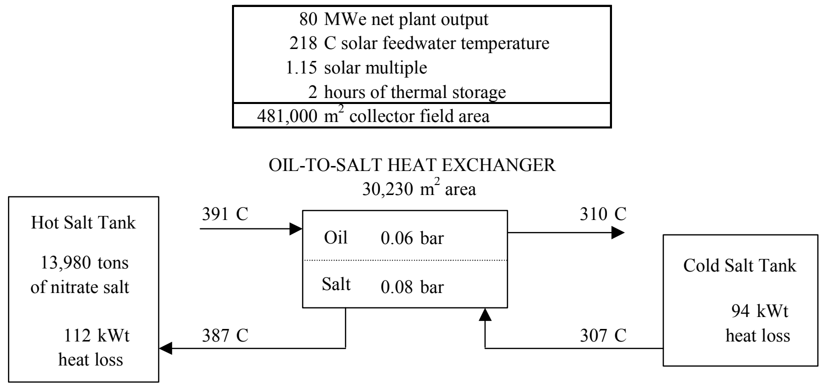
4.1.1 Series and Parallel Arrangement

Various combinations of series and parallel heat exchangers were examined to determine the optimum combination of heat transfer area and pressure loss. Constraints to the selection included a maximum shell diameter of 3 m (120 in.), a total shell side pressure loss of 5 bar (75 lb_f/in^2), and a total tube side pressure loss of 3.5 bar (50 lb_f/in^2).

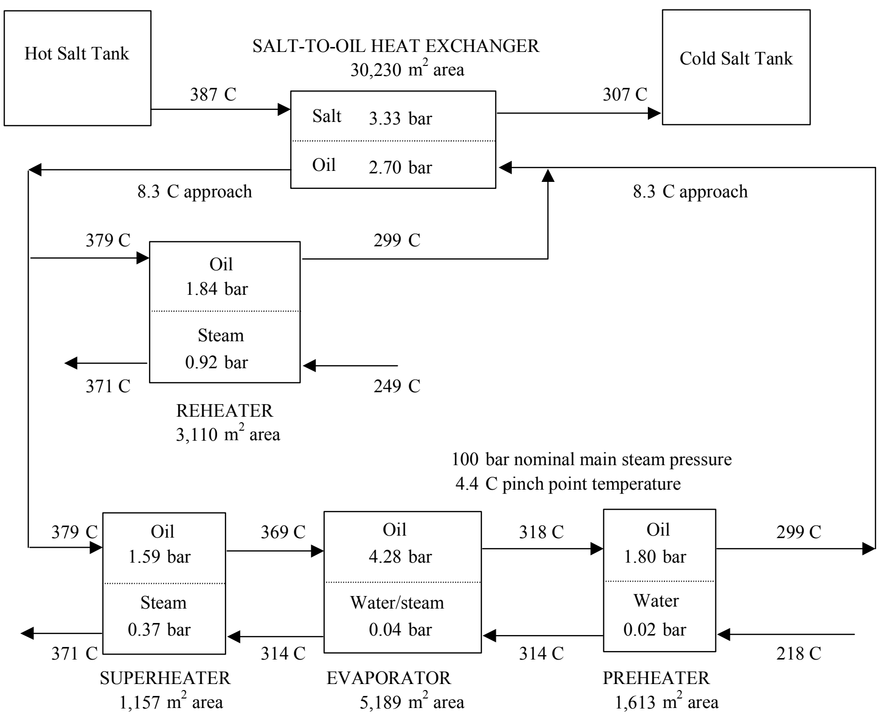
The preferred arrangement consisted of 1 parallel unit with 4 heat exchangers in series. The principal design details included the following:

- A tube diameter and wall thickness of 15.9 mm and 1.25 mm (0.625 in. and 0.049 in.), respectively. The tubes were arranged on a square pitch.

- An overall heat transfer coefficient of 932 W/m^2-C (164 Btu/hr-ft^2-F), including a fouling factor of 0.000088 m^2-C/W (0.0005 hr-ft^2-F/Btu) on both the inside and the outside of the tubes



System Characteristics During Charging



System Characteristics During Discharging

Figure 4 Thermal Storage and Steam Generation System Design Parameters

- An effective surface area of 7,553 m^2 (81,301 ft^2) in each heat exchanger, for a total surface area of 30,212 m^2 (325,203 ft^2)

- Inlet and outlet nitrate nozzle sizes, for both the nitrate salt and oil, of 815 mm (32 in.).

Heat Exchanger Specification sheet					
				Job No.	24118-109
Customer	NREL/BNI			Ref No.	
Address				Proposal No.	
Plant Location	Southern California			Date	Rev.
Service of Unit	Oil-to-Salt Heat Exchanger			Item No	
Size 102x 480	Type	BFU - HORZ	Connected in	1 Parallel	4 Series
Surf/Unit (Eff)	325203 ft²	Shells/Unit	4	Surface/Shell (Effective)	81301 ft²

PERFORMANCE OF ONE UNIT

Fluid Allocation		**Shellside**	**Tubeside**
Fluid Name		Molten Salt	Therminol VP-1
Total Fluid Entering	lb/hr	15,639,324	9,716,949
Vapor		0	0
Liquid		15,639,324	9,716,949
Steam			
Noncondensable			
Fluid Vaporized or Condensed		0	0
Liquid Density (In/Out)	lb/ft³	115.000/118.000	51.050/45.500
Liquid Viscosity	cP	2.383	0.187
Liquid Specific Heat	Btu/lb-F	0.360	0.580
Liquid Thermal Conductivity	Btu/hr-ft-F	0.295	0.051
Vapor Mol. Weight (In/Out)		0.0/0.0	0.0/0.0
Vapor Viscosity	cP	0.0000	0.0000
Vapor Specific Heat	Btu/lb-F	0.000	0.000
Vapor Thermal Conductivity	Btu/hr-ft-F	0.000	0.000
Temperature (In/Out)	°F	727.2/585.0	570.0/712.2
Operating Pressure	psi(Abs)	150.000	200.000
Velocity	ft/sec	2.354	7.136
Pressure Drop (Allow/Calc)	psi	75.000/32.303	50.000/48.201
Fouling resistance	hr-ft²-F/Btu	0.000500	0.000500

Heat Exchanged	800,682,688 Btu/hr	mtd (corr)	15.000 °F
Transfer Rate, Service	164.1	Clean	210.0 Btu/hr-ft²-F

CONSTRUCTION OF ONE SHELL

		Shellside	**Tubeside**	**Sketch**
Design/Test Pres.	psi	200/	300/	
Design Temp.	°F	750	750	
No. Passes per Shell		2	2	
Corrosion Allow.	in	0.1250	0.1250	
Connections	In	1-33.0 W.E.	33.0 300#RF	
Size &	Out	1-33.0 W.E.	33.0 300#RF	
Rating	Intermediate	0	0	

Tube No	5882 U	OD 0.625 in	Thk 16min	Length 40.00 ft	Pitch 0.81250 / 90.0°
Tube Type	PLAIN		Material	SA-249/TYP 304SS	
Shell	CS	I.D 102.00 OD in	Shell Cover	CS INT	
Channel or Bonnet	CS		Channel Cover		
Tubesheet-Stationary	CS		Tubesheet-Floating		
Floating Head Cover			Impingement Protection	NO	
Baffles Cross	CS	Type GRID- BAFFLE	%Cut 79.3 (Area)	Spacing-cc	9.0
Baffles-Long	CS		Seal Type	Lamiflex	
Supports-Tube		U-Bend		Type	
Bypass Seal Arrangement			Tube-Tubesheet Joint		
Expansion Joint			Type		
Rho-V2 Inlet Nozzle	4,652	Bundle Entrance		Bundle Exit	
Gasket-Shellside		Tubeside		Floating Head	
Code Requirement	ASME Section 8, Divsion 1			TEMA Class	R
Weight/Shell		Filled with Water		Bundle	
Remarks:	1) Provide vapor belt with 44 ft2 of shell open area per shell nozzle. 2) All welded construction on the shellside.				

Figure 5 Heat Exchanger Data Sheet

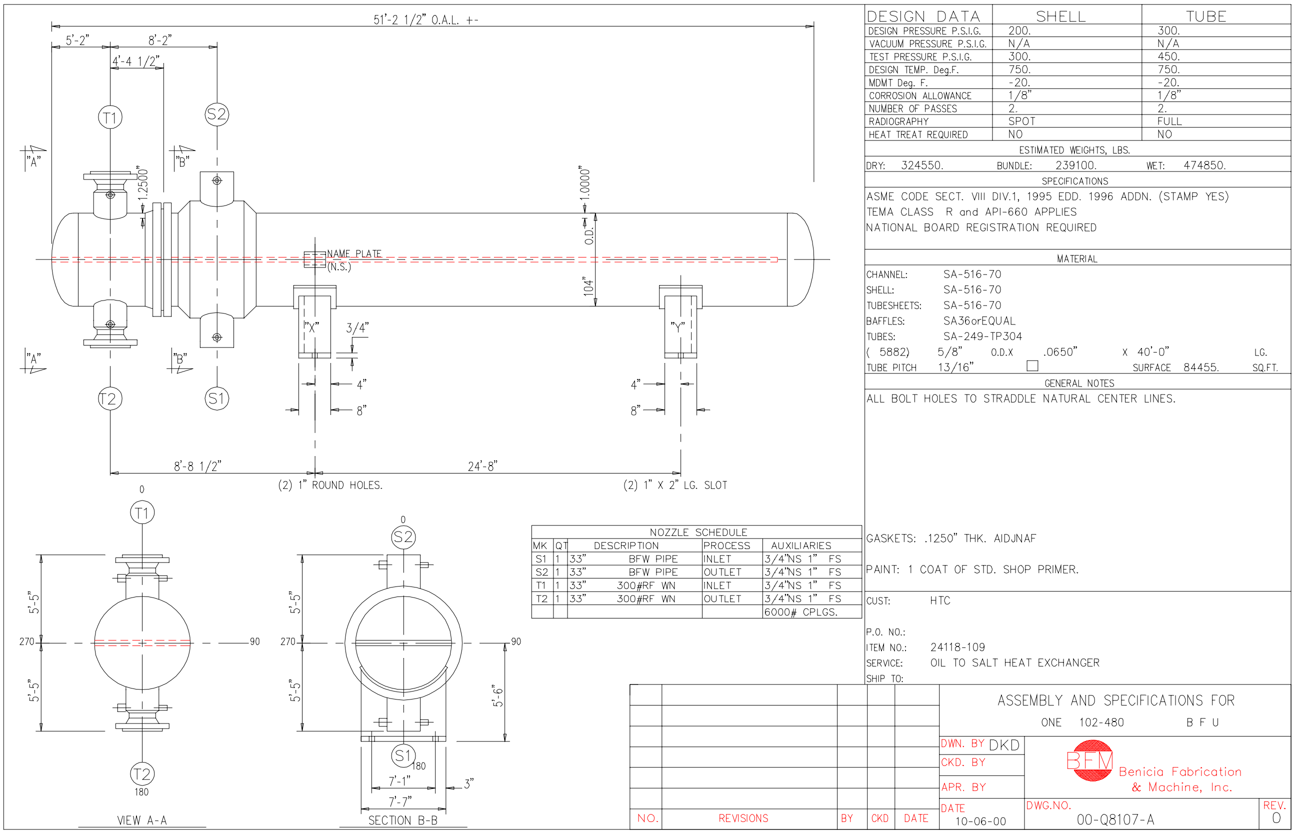


Figure 6 Oil-to-Salt Heat Exchanger Elevation Drawing

4.1.2 Baffle Design

Two baffle designs for controlling the flow on the shell side of the heat exchanger were evaluated, as follows:

- Segmented Baffle A segmented baffle is a perforated plate which spans about two-thirds of the tube bundle diameter and supports the tubes passing through the baffle. By alternating sequential baffles up and down, the nitrate salt flowed alternately along the tube bundle and then across the tube bundle. Increasing the spacing between the baffles decreased the number of times the shell fluid reversed flow, which decreased the pressure losses. However, the maximum baffle spacing was determined by tube vibration considerations.

- Grid Baffle The grid baffle consists of four alternating series of vertical and horizontal support rods. In the first group, vertical rods were placed between every other row in the bundle, and then welded to a circumferential support ring. In the second group, vertical rods would again be used, but placed between the rows not used in the first group. In the third and fourth groups, the rods were placed horizontally. The nitrate salt flowed along the length of the tube bundle in an essentially counterflow arrangement. A grid spacing of 230 mm (9 in.) effectively precluded any possibility of tube vibration.

The latter design was selected to meet the pressure loss constraints outlined above.

4.1.3 Materials

As shown in the heat exchanger data sheet, the tube material was ASTM SA-249, or Type 304 stainless steel. In theory, carbon steel tube could have been used because the operating temperatures were below 400 ºC (750 ºF), and non-corrosive fluids were used on both the shell side and the tube side. However, carbon steel tubes are not normally manufactured in the 26 m (85 ft.) lengths required for the heat exchanger. To overcome this problem, two approaches were considered. First, an alternate tube material, normally fabricated in lengths of at least 26 m, could be used. Second, two carbon steel tubes could be butt welded to provide the required length.

The latter approach was dropped in the interests of heat exchanger reliability, and the lowest cost alternate material was found to be Type 304 stainless steel.

4.2 Thermal Storage Tanks

The thermal storage system must provide a thermal capacity of 470 MWht with a hot salt temperature of 386 ºC (727 ºF) and a cold salt temperature of 307 ºC (585 ºF). The characteristics of the storage tanks are outlined in Table 1.

Table 1
Thermal Storage Tank Design Parameters

Active nitrate salt inventory	14,000,000 kg	(31,000,000 lb_m)
Storage depths		
- Active	10.7 m	(35 ft)
- Heel	0.6 m	(2 ft)
- Heel from other tank	0.6 m	(2 ft)
- Freeboard	0.3 m	(1 ft)
Tank height	12.2 m	(40 ft)
Tank diameter	29.9 m	(98 ft)
Total nitrate salt inventory	15,600,000 kg	(34,400,000 lb_m)

The required nitrate salt inventory consisted of the following: 14,000 tons for the active inventory; 800 tons for the heel in the cold salt tank; and 800 tons for the heel in the hot salt tank. A heel depth of 0.6 m (2 ft.) was assumed for each tank.

It should be noted that the heel depth for each of the thermal storage tanks at the Solar Two project was 0.9 m (3 ft.). The tank floor-to-wall weld joint was judged by the tank vendor to experience the largest thermal stresses during daily plant startup. To ensure a joint fatigue life of 30 years, a stagnant inventory heel isolated the joint from the daily thermal transients. However, it was believed that the heel for the trough storage system could be safely reduced to 0.6 m (2 ft.) for two reasons. First, the estimated rates of temperature change near the bottom of the tanks in the Solar Two project were lower than anticipated. Second, the difference in temperatures between the cold and hot tanks in the trough system was 199 ºC (358 ºF) less than the difference in temperatures in the Solar Two project; thus, the magnitude of the transient stresses should be less than the Solar Two project. Selecting a heel depth as small as possible offered the following benefits: 1) the inactive nitrate salt inventory was reduced; and 2) the diameter of the tanks was reduced, assuming the height of the tanks was constrained by the length of nitrate salt pumps available.

4.3 Nitrate Salt Pumps

During thermal storage charging, a cold salt pump, located in the cold salt tank, delivered nitrate salt to the oil-to-salt heat exchanger. During storage discharging, a separate hot salt pump, located in the hot salt tank, delivered nitrate salt to the heat exchanger. The characteristics of the two pumps are presented in Table 2, and an elevation view of a prototype pump with an extended shaft is shown in Figure 7.

Table 2
Nitrate Salt Pump Design Parameters

Cold Salt Pump		
- Design condition	Thermal storage charging	
- Oil-to-salt heat exchanger duty	35 MWt	(119,000,000 Btu/hr)
- Flow rate	291 kg/sec	(2,310,000 lb_m/hr)
- Head losses		
· Heat exchanger	0.40 m	(1.3 ft)
· Piping and valves	0.12 m	(0.4 ft)
· Static head	13.7 m	(45.0 ft)
- Motor power demand	58 kWe	(78 bhp)
Hot Salt Pump		
- Design condition	Thermal storage discharging	
- Salt-to-oil heat exchanger duty	235 MWt	(802,000,000 Btu/hr)
- Flow rate	1,941 kg/sec	(15,400,000 lb_m/hr)
- Head losses		
· Heat exchanger	18.2 m	(59.7 ft)
· Piping and valves	5.7 m	(18.8 ft)
· Static head	13.7 m	(45.0 ft)
- Motor power demand	938 kWe	(1,260 bhp)

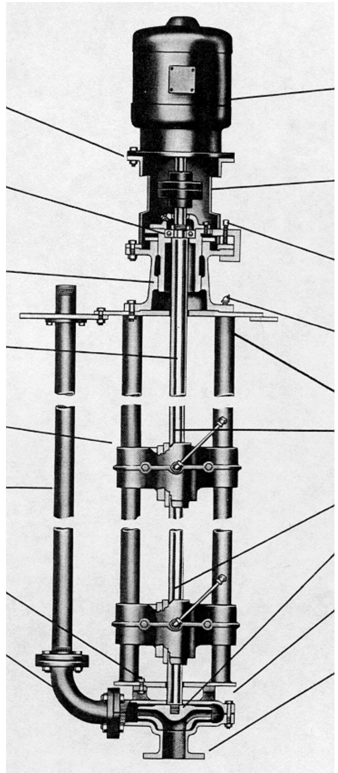


Figure 7 Cross Section of Prototype Nitrate Salt Pump

4.5 Piping and Instrument Diagram

A preliminary piping and instrument diagram for the oil-to-salt heat exchangers, thermal storage tanks, and nitrate salt pumps is shown in Figure 8. A legend for the symbols is shown on the drawing. The selection of the piping, valves, and instruments was based on the following approach:

- Temperature and pressure elements were placed at the two inlets and two outlets of each heat exchanger to aid in the detection of a tube leak.

- Nitrate salt flow rates through the heat exchangers were controlled by variable speed drives on the pump motors, rather than by control valves. The variable speed drives on the Solar Two project provided accurate flow and temperature control, and were much more reliable than the control valves.

- Nitrate salt and heat transport fluid isolation valves were placed only at the inlets to, and outlets from, the string of four heat exchangers. Should a tube leak occur, the entire system would be probably be shut down, and there would be little need to isolate an individual heat exchanger.

- Pressure/vacuum relief valves were located on each storage tank to control ullage pressures should a failure occur in the nitrogen gas system.

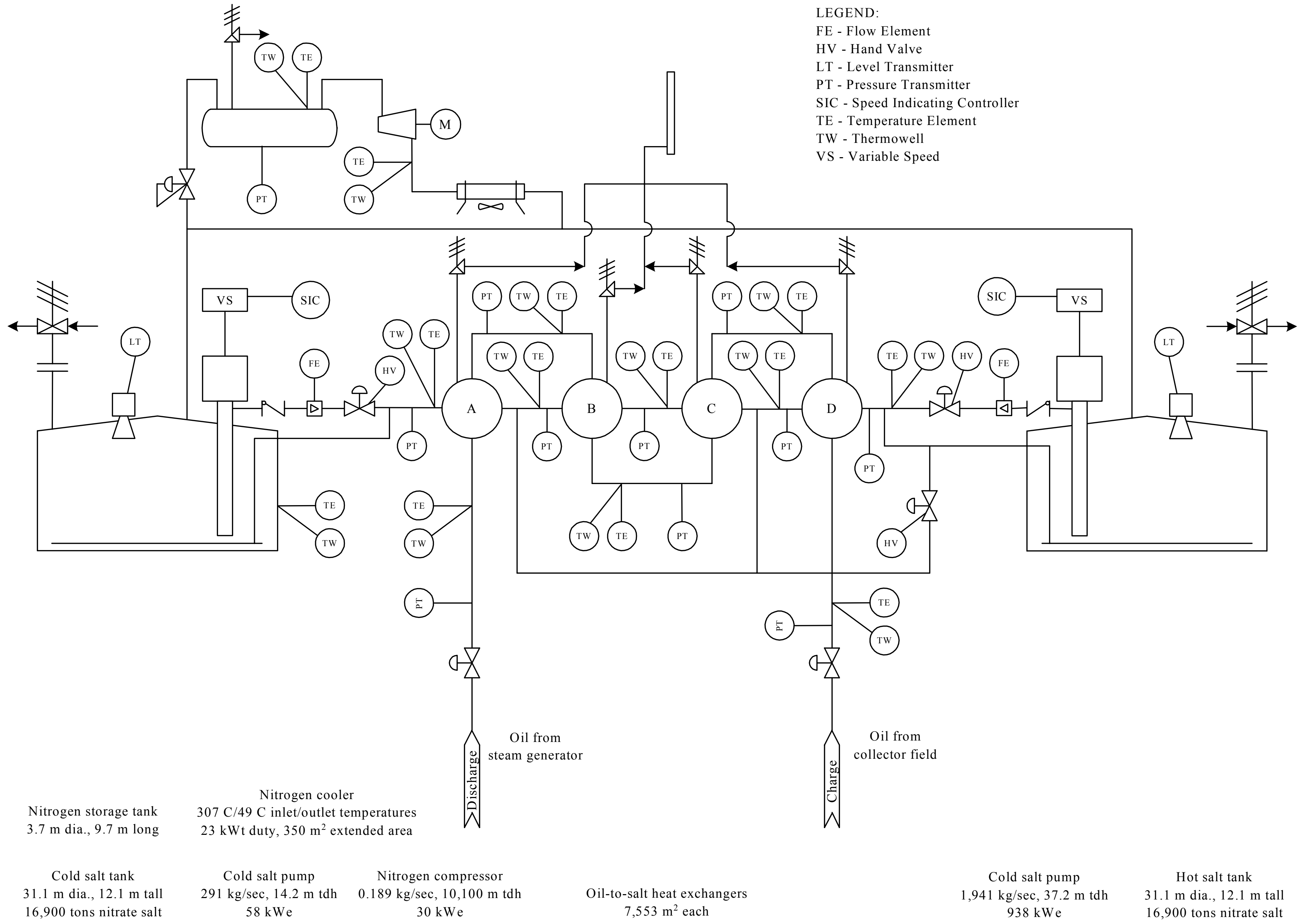

LEGEND:
FE - Flow Element
HV - Hand Valve
LT - Level Transmitter
PT - Pressure Transmitter
SIC - Speed Indicating Controller
TE - Temperature Element
TW - Thermowell
VS - Variable Speed
M
VS
SIC
A
B
C
D
Discharge
Oil from steam generator
Charge
Oil from collector field
Nitrogen storage tank
3.7 m dia., 9.7 m long
Nitrogen cooler
307 C/49 C inlet/outlet temperatures
23 kWt duty, 350 m² extended area
Cold salt tank
31.1 m dia., 12.1 m tall
16,900 tons nitrate salt
Cold salt pump
291 kg/sec, 14.2 m tdh
58 kWe
Nitrogen compressor
0.189 kg/sec, 10,100 m tdh
30 kWe
Oil-to-salt heat exchangers
7,553 m² each
Cold salt pump
1,941 kg/sec, 37.2 m tdh
938 kWe
Hot salt tank
31.1 m dia., 12.1 m tall
16,900 tons nitrate salt

Figure 8 Piping and Instrument Diagram

- Radar level detectors, rather than bubbler instruments, measured the liquid levels in the storage tanks. The radar detectors are available on a commercial basis from several vendors, and should be more reliable than the somewhat problematic bubbler instruments at the Solar Two project.

4.5 Equipment Arrangement

Possible arrangements of the oil-to-salt heat exchangers, thermal storage tanks, nitrate salt pumps, and piping are shown in Figures 9 and 10. The heat exchangers were located on a platform above the storage tanks to facilitate draining.

Salt flow into each tank was introduced through a ring header with several eductors. A similar arrangement effectively eliminated thermal stratification in the cold salt tank on the Solar Two project.

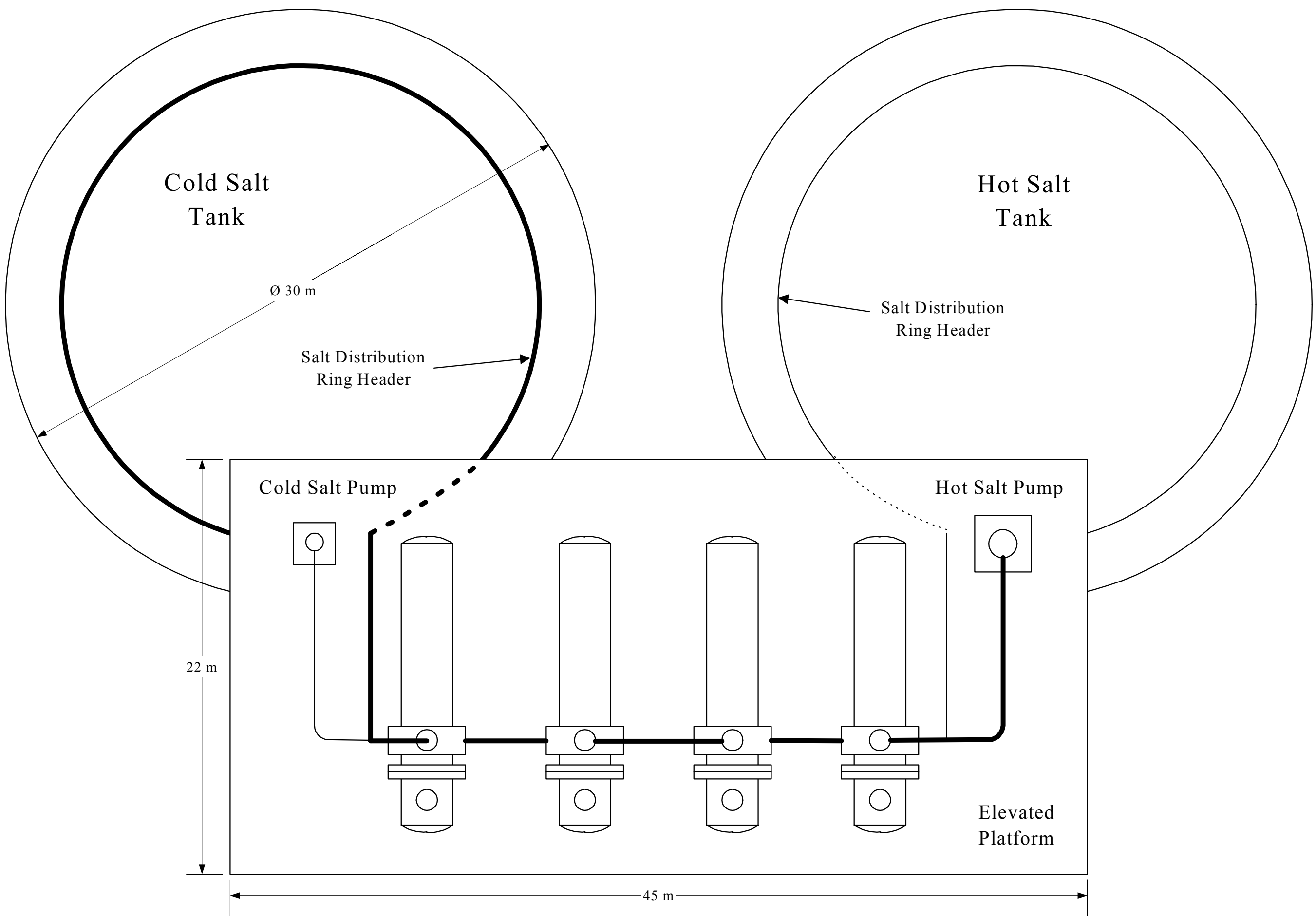


Figure 9 Equipment Arrangement - Plan View

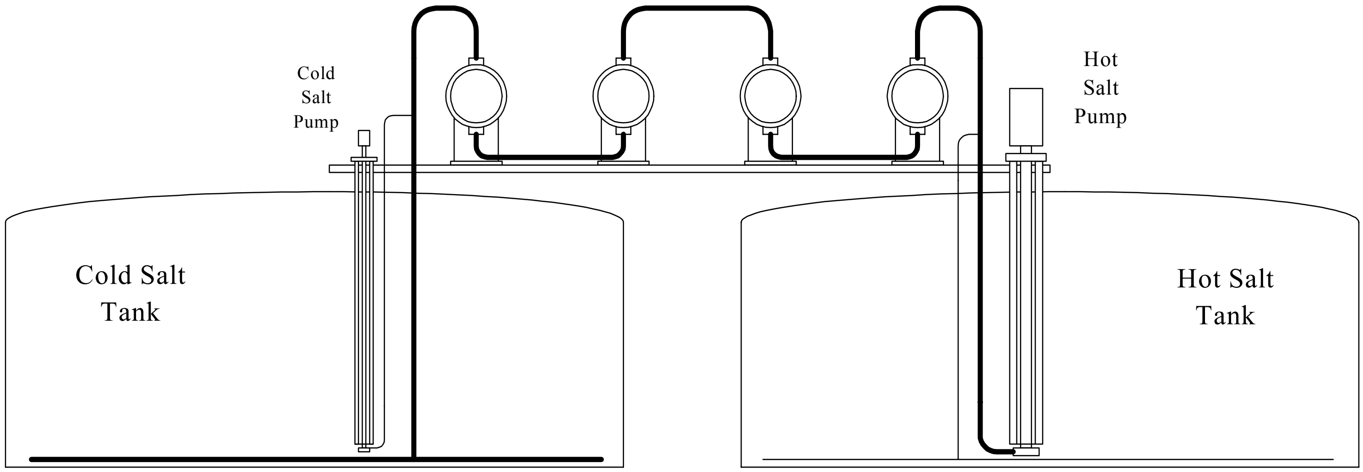


Figure 10 Equipment Arrangement - Elevation View

5 FAILURE MODES ANALYSIS

Several failures of the oil-to-salt heat exchanger and associated equipment have been postulated. The effects of the failures, and possible responses of the safety equipment and operators, are discussed below.

5.1 Tube Rupture

5.1.1 Failure and Effects

The most severe failure would be a rupture in a tube of the oil-to-salt heat exchanger. The flow from a ruptured tube can be estimated using the basic formula for the flow through a nozzle or orifice, as follows:

$$W = 2.0265\, K\, Y\, d^2 \sqrt{\frac{\Delta P}{\rho_1}}$$

where W = Discharge flow rate, kg/hr
K = Discharge coefficient (0.7 for square edged orifice)
Y = Expansion factor (1.0 for liquid)
d = Tube inside diameter, mm
ΔP = Difference between fluid pressures in the tube and shell, kPa
ρ_1 = Fluid density in tube, kg/m^3

Assuming a heat transport fluid pressure of 1,480 kPa (215 lb_f/in^2) and a nitrate salt pressure of 172 kPa (25 lb_f/in^2), the flow rate of heat transport fluid into the shell from each end of the ruptured tube was estimated to be 284 kg/hr (626 lb_m/hr).

The saturation pressure of the heat transport fluid, at a temperature of 378 ºC (712 ºF), was approximately 310 kPa (45 lb_f/in^2). However, if the pressure on the shell side of the heat exchanger was 172 kPa (25 lb_f/in^2), a portion of the heat transport fluid would vaporize just downstream of the tube break. An enthalpy balance on the heat transport fluid showed liquid and vapor fractions of 18 and 82 percent by weight, respectively. The total vapor flow rate of 466 kg/hr (1,026 lb_m/hr) represented a volume flow rate of 163 m^3/hr (96 ft^3/min).

If the pressure of the Therminol vapor on the shell side of the heat exchanger did not rise above the pressure setting for the relief valve, the heat transport fluid vapor would be carried into the ullage space of either the cold salt tank or the hot salt tank. The vapor concentration as a function of time, and the flammability limits of biphenyl, are shown in Figure 10. If the storage tank was full, a combustible mixture was formed after about 7 minutes, and persisted for 68 minutes. If the storage tank was empty, a combustible mixture was formed in 74 minutes and lasted for 716 minutes.

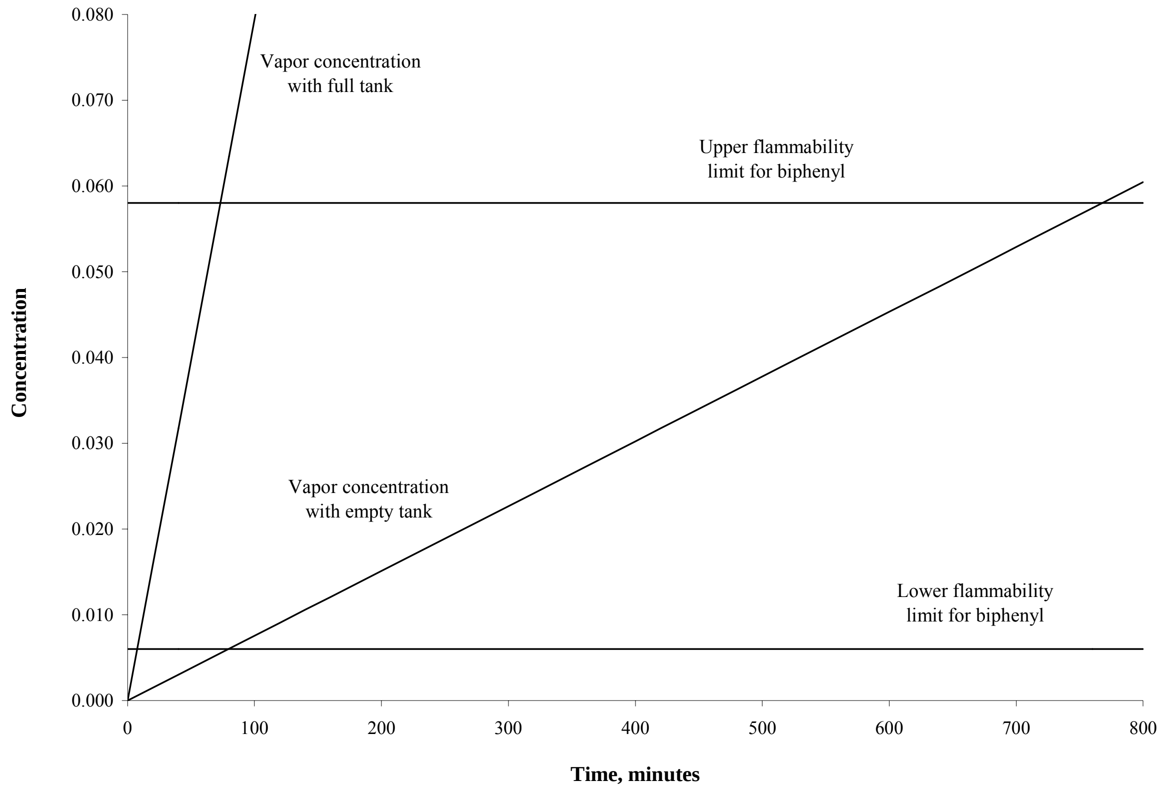


Figure 10 Biphenyl Vapor Concentration in the Thermal Storage Tank Following a Tube Rupture in the Oil-to-Salt Heat Exchanger

5.1.2 Equipment and Operator Responses

A tube rupture would most likely be discovered through a pressure rise on the shell side of the heat exchanger. In response to a high pressure alarm, the distributed control system would perform the following: issue a ‘stop’ command to the nitrate salt pump; and issue a ‘close’ command to the nitrate salt and the heat transport oil isolation valves. No responses would be required on the part of the operators.

In theory, the heat transport oil pumps could also be stopped to limit the quantity of fluids which mixed in the heat exchanger. However, issuing a trip command for the oil pumps would result in a defocus command for the collector field, and shortly thereafter, a trip command for the turbine-generator. The potential damage to the collector field and turbine-generator due a trip command was judged to be more severe than some additional Therminol vaporization in the heat exchanger, and the heat transport oil pumps would likely remain in operation.

5.1.3 Further Safety Measures

As noted in Section 3, an electric spark or a flame would be required to ignite a combustible mixture in the storage tank. Thus, the likelihood that a tube leak would lead to a fire was judged to be very remote.

However, if the probability of a fire needed to be reduced even further, a cover gas of nitrogen could be used in the storage tanks.

A schematic for the nitrogen system is shown in Figure 8. At the beginning of the day, the cold salt tank would be full and the hot salt tank would be empty. The pressure and mass of nitrogen in the gas storage tank would be 690 kPa (100 lb_f/in^2) and 818 kg (1,803 lb_m), respectively.

As the storage system was charged, nitrogen flowed from the hot salt tank to the cold salt tank, and cooled from an initial temperature of 386 ºC (727 ºF) to a final temperature of 307 ºC (585 ºF). To prevent the ullage pressure in the cold tank from falling below 101 kPa (14.7 lb_f/in^2), some of the nitrogen would be taken from the gas storage tank, throttled to atmospheric pressure, and sent to the cold salt tank. When the storage system was completely charged, the pressure and mass in the nitrogen storage tank were 358 kPa (25 lb_f/in^2) and 202 kg (445 lb_m), respectively.

As the storage system was discharged, nitrogen flowed from the cold salt tank to the hot salt tank, and heated from an initial temperature of 307 ºC (585 ºF) to a final temperature of 386 ºC (727 ºF). To prevent the ullage pressure in the hot tank from rising above 101 kPa (14.7 lb_f/in^2), nitrogen was drawn from the hot salt tank, compressed to a pressure of 690 kPa (100 lb_f/in^2), and sent to the gas storage tank.

The required gas storage tank was 3.66 m (12 ft) in diameter and 9.7 m (32 ft) long, and the compressor power requirement was 30 kWe.

The gas storage system imposed three minor demands on the plant, as follows:

- The compressor operated for 2 hours each day. Over the course of a year, the auxiliary energy demand was a modest 22 MWhe.

- Prior to compression, the nitrogen must be cooled from 307 ºC (585 ºF) to a nominal temperature of 50 ºC (120 ºF) to provide a reasonable compressor outlet temperature. Over the course of a year, the heat rejected by the nitrogen-to-air heat exchanger was a moderate 17 MWht.

- Atmospheric pressure tanks must be vented to the atmosphere through a combination pressure/vacuum relief valve. If nitrogen from the gas storage tank is delivered to the cold salt tank at a pressure slightly below the vacuum relief setting, and if nitrogen is removed from the hot salt tank at a pressure slightly below the pressure relief setting, the consumption of nitrogen can, in theory, be zero. In practice, some nitrogen will be lost due to inaccurate pressure control or diffusion. However, the losses in a well maintained system should be small, and a nitrogen ullage system should not impose a significant annual operating expense on the plant.

5.2 Heat Transport Pumps

5.2.1 Failure and Effects

Failure of the heat transport pumps during thermal storage charging would cause rapid cooling of the oil-to-salt heat exchanger, and a reduction in the fatigue life of the exchanger. Similarly, failure of the oil pumps during discharging would cause a rapid heating of the heat exchanger, and also a corresponding reduction in fatigue life.

5.2.2 Equipment and Operator Responses

Failure of the heat transport pumps during storage charging or discharging would soon result in a trip command for the turbine-generator. Thus, there would be no penalty for stopping the nitrate salt pumps, and a trip command for the salt pumps would be issued by the distributed control system immediately after a failure in the heat transport fluid pumps. No responses on the part of the operators were required.

It should be noted that the steam generator heat exchangers at the Solar Electric Generating Station plants are routinely subjected to fairly rapid temperature changes during plant startup. The heat exchangers have been operated in this manner for more than 10 years with no apparent mechanical damage. As a result, a failure of the heat transport pumps was not likely to cause significant fatigue damage to the oil-to-salt heat exchangers, even if the flow from the nitrate salt pumps was not stopped immediately.

5.3 Nitrate Salt Pump

5.3.1 Failure and Effects

Failure of the cold salt pump during thermal storage charging would cause rapid heating of the oil-to-salt heat exchanger, and a reduction in the fatigue life of the exchanger. Similarly, failure of the hot salt pump during discharging would cause a rapid cooling of the heat exchanger, and also a corresponding reduction in fatigue life.

5.3.2 Equipment and Operator Responses

Failure of the cold salt pump during storage charging would have little influence on the operation of either the collector field or the Rankine cycle. To avoid undesirable trips of the collector field or the turbine-generator, a 'close' command would be issued to the isolation valve on the oil inlet to the heat exchangers by the distributed control system to limit the increase in temperature of the exchangers.

Failure of the hot salt pump during storage discharging would soon result in a trip command for the turbine-generator. Thus, trip commands for the heat transport fluid pumps and the steam generator would be issued immediately by the distributed control system after a failure of the hot salt pump.

No responses were required by the operators following a failure of either the cold or the hot salt pump.

5.4 Nitrogen Ullage Gas System

5.4.1 Failure and Effects

Failure of the ullage system during a charge cycle would cause air to mix with the nitrogen ullage in the cold salt tank; failure during a discharge cycle would cause nitrogen to be lost from the hot salt tank.

The nitrate salts are chemically stable in the presence of air. Thus, the only effect of a failure in the ullage gas system would be a small increase in the risk of equipment damage due to the unlikely event of a fire or explosion in the storage tank following the unlikely event of a tube leak in the oil-to-salt heat exchanger.

5.4.2 Equipment and Operator Responses

During a charge cycle, ullages pressures below atmospheric in the cold salt tank would be relieved by the vacuum relief valve. Similarly, during a discharge cycle, ullage pressures above atmospheric in the hot salt tank would be relieved by the pressure relief valve.

No responses were required by the operators following a failure of the ullage system.

6 CAPITAL COST ESTIMATE

A conceptual capital cost estimate for the thermal storage system, including the oil-to-salt heat exchangers, nitrate salt tanks and inventory, nitrate salt pumps, and auxiliary equipment, has been developed. The basis for the estimate is outlined in the following sections. A summary of the estimate is presented in Table 3, and details of the estimate are presented in Appendix A.

6.1 Mechanical Equipment

Costs for the mechanical equipment were developed as follows:

- Oil-to-salt heat exchangers. A unit price of $146/m^2 ($13.50/ft^2) was developed by the heat exchanger consultant in collaboration with a local fabrication shop.

- Nitrate salt pumps. Unit prices for the hot and cold salt pumps were estimated to be $600/kW ($450/bhp) and $1,680/kW ($1,250/bhp), respectively, using cost information from the Solar Two project.

- Nitrogen compressor and cooler. Material and installation costs were developed from Bechtel historical data on similar refinery equipment.

6.2 Tanks and Vessels

Material thicknesses for the walls and floors of the thermal storage tanks were estimated using the same material stresses as the tanks in the Solar Two project. Foundation concrete and insulation material quantities were scaled directly from the materials required for the cold salt tank in the Solar Two project. Material cost, installation costs, and thermal insulation costs were estimated using vendor data from the Solar Two project.

6.3 Nitrate Salt Inventory

In an effort to hold the storage system costs to a minimum, a binary nitrate salt has been used in the cost estimate. The unit price for salt in solid form, delivered to the site, was estimated to be $400/metric ton ($0.18/lb$_m$) based on vendor information from the Solar Two project. Labor and materials costs for handling and melting were estimated to be $50/metric ton ($0.025/lb$_m$).

If a salt with a lower melting point is required, a tertiary salt could be substituted for the binary salt. A probable salt would be Hitec XL, which is a nominal mixture of 15 percent sodium nitrate, 43 percent potassium nitrate, and 42 percent calcium nitrate. Purchased in small quantities of 25 kg, the price was approximately $715/metric ton ($0.325/lb$_m$); purchased in the very large quantities required for the thermal storage system, the price could be as low as $575/metric ton ($0.26/lb$_m$). Substituting the tertiary salt would increase the capital cost of the storage system by $2,900,000, and increase the unit cost by $6/kWht to a new value of $46/kWht.

Table 3
470 MWht Thermal Storage System Conceptual Cost Estimate

Item	Direct Field Cost, $1000	Contingency	Total Cost, $1000
Land	2	0%	2
Mechanical Equipment	5,163	10%	5,679
Tanks and Vessels	3,418	15%	3,931
Nitrate Salt Inventory	7,100	5%	7,455
Piping and Instrumentation	663	20%	796
Civil and Structural	461	20%	554
Electric Equipment and Bulk Materials	261	20%	313
Distributed Control System	50	30%	65
	---------		---------
Total	17,118		18,794
Unit Cost, $/kWht	$36.40		$40.00

6.4 Piping and Instrumentation

All of the nitrate salt, heat transport oil, and nitrogen piping and valves were fabricated from carbon steel. Unit material prices for the pipe were estimated to be $2.20/kg ($1.00/lb$_m$), and prices for the valves were developed from Solar Two cost data. Installation labor costs were developed from Bechtel historical data. Unit insulation and electric heat trace costs were developed directly from vendor prices on the Solar Two project.

6.5 Electric Equipment and Bulk Materials

Unit electric equipment and bulk material costs were estimated using Bechtel historical refinery and power plant information.

6.6 Distributed Control System

An allowance of $50,000 was included for incremental costs to the distributed control system, as follows:

- Development of additional graphic displays on the operator consoles for temperature, pressure, flow, level, pump speed, and valve positions

- Additional programmable logic controller relay boards for inputs from, and outputs to, the electric heat trace circuits, pneumatic valves, nitrate salt pump motors, nitrogen compressor, and nitrogen cooler fan.

- Additional software programming for the distributed control system and the programmable logic controllers.

6.7 Labor

The following direct labor costs were used: \$17/hour for mechanical work; \$14/hour for civil/structural work; and \$18/hour for both piping and electrical work. An allowance for distributable costs was added to each direct labor cost to cover expenses for equipment rental, consumable supplies, and site maintenance during construction. The distributable rate was estimated to be 80 percent.

6.8 System Contingencies

The conceptual design outlined in Section 4 was not a complete final design. As a result, all of the equipment, bulk materials, and labor required for construction have yet to be identified. To include costs for the unidentified items, a contingency was added to each system cost.

6.9 Approaches for Cost Reductions

Possible refinements to the design which could lead to a reduction in the cost of the storage system include the following:

- The approach temperatures in the oil-to-salt heat exchanger were smaller than what might be considered typical commercial practice. The heat exchanger was designed such that the temperatures of the live and reheat steam would be 371 ºC (700 ºF) whether the steam generator was operating from the collector field or the thermal storage system. In practice, a more detailed analysis, which considered full and part load Rankine cycle performance over the course of a year, might lead to a different set of temperatures. For example, allowing the live and reheat steam temperatures to decrease to 365 ºC (690 ºF) during the limited number of hours operation from storage would reduce the surface area in the oil-to-salt heat exchangers by about 3,500 m^2, and reduce the capital cost by \$500,000.

- If carbon steel tubes could be found with a length of 27 m (90 ft.), the unit cost of the heat exchangers could be reduced by about \$22/m^2 (\$2/ft^2) for a capital cost savings of \$700,000.

- Eliminating the elevated platform would reduce the capital cost by about \$500,000. However, a small sump and pump would need to be located below grade to drain the heat exchangers.

- The separate hot and cold salt tanks could be replaced by a single thermocline tank. If the storage component tests currently underway at Sandia National Laboratories prove successful, the unit cost of the storage system could be reduced by 20 to 30 percent.

7 REFERENCES

1) "Task 2 Report, Thermal Storage for Rankine Cycle Power Plants", Task Order Authorization Number KAF-9-29765-09, Nexant LLC, San Francisco, California, February 2000

2) "Molten Salt Safety Study", Martin Marietta Corporation (Denver, Colorado), Sandia National Laboratories Report SAND80-8179, Month, 1980

470 MWht Thermal Storage System Total Field Cost Estimate

Code	Description	Qty	Unit	Unit Cost Material	Unit Cost S/C	Labor Rate	MH Rate	Total MH	$ Material	$ Labor	$ S/C	$ Total
0.00	LAND	2	AC		950						1,900	1,900
1.00	MECHANICAL EQUIPMENT											
1.10	OIL-TO-SALT HEAT EXCHANGERS	4	EA	1,100,000		30.60	390	1,560	4,400,000	47,740		4,447,740
	U-TUBE, STRAIGHT SHELL; 7,553 M^2 EACH											
	CARBON STEEL SHELL, CHANNEL, AND TUBESHEET											
	TYPE 304 STAINLESS STEEL TUBES											
1.20	HOT SALT PUMP	1	EA	565,614		30.60	460	460	565,610	14,080		579,690
	1,941 KG/SEC, 37.2 M TOTAL DEVELOPED HEAD, 938 kWe DRIVE											
	VARIABLE SPEED DRIVE											
1.30	COLD SALT PUMP	1	EA	97,150		30.60	90	90	97,150	2,750		99,900
	291 KG/SEC, 14.2 M TOTAL DEVELOPED HEAD, 58 kWe DRIVE											
	VARIABLE SPEED DRIVE											
1.40	NITROGEN COMPRESSOR	1	EA	10,050		30.60	140	140	10,050	4,280		14,330
	0.189 KG/SEC, 10,100 M TOTAL DEVELOPED HEAD, 30 kWe DRIVE											
1.50	NITROGEN COOLER	1	EA	18,837		30.60	70	70	18,840	2,140		20,980
	FORCED DRAFT, FINNED TUBE HEAT EXCHANGER											
	23 kWt DUTY, 350 M^2 EXTENDED HEAT TRANSFER AREA											
									------------	----------	----------	------------
1.0	TOTAL MECHANICAL EQUIPMENT								5,091,650	70,990	0	5,162,640
2.00	TANKS AND VESSELS											
2.10	NITRATE SALT TANKS	2	EA	795,000		25.20	16,100	32,200	1,590,000	811,440		2,401,440
	29.9 M DIAMETER, 12.1 M TALL, CARBON STEEL											
2.20	WALL AND ROOF INSULATION	3,678	M^2		129						475,010	475,010
	300 mm CALCIUM SILICATE WITH ALUMINUM JACKET											
2.30	SALT TANK FOUNDATIONS											
	- EXCAVATION	1,805	M^3		2.5						4,510	4,510
	- BACKFILL	401	M^3		5						2,000	2,000
	- RING WALLS - CONCRETE	188	M^3	65		25.20	1.92	361	12,210	9,100		21,310
	- FORMS	38	M^2	22		25.20	6.20	233	810	5,870		6,680
	- REBAR	12	T		700						8,140	8,140
	- FOUNDATION CONCRETE	702	M^3	65		25.20	1.92	1,348	45,640	33,970		79,610
	- FIREBRICK PERIMETER INSULATION	51,625	EA	0.35		25.20	0.02	1,033	18,070	26,030		44,100
	- FOAMGLASS FOUNDATION INSULATION	642	M^3		356						228,570	228,570
	- FOUNDATION COOLING PIPES	871	M	73		32.40	1.38	1,202	63,890	38,940		102,830
	200 mm SCH 20, CARBON STEEL, 1.5 M ON CENTER, 3 M STACK HT.											

470 MWht Thermal Storage System Total Field Cost Estimate

				Unit Cost		Labor	MH	Total	$			
Code	Description	Qty	Unit	Material	S/C	Rate	Rate	MH	Material	Labor	S/C	Total
2.40	NITROGEN STORAGE VESSEL	1	EA	42,000		30.60	60	60	42,000	1,840		43,840
	3.7 M DIAMETER, 12.7 M LONG, CARBON STEEL											
2.0	TOTAL TANKS AND VESSELS								1,772,620	927,190	718,230	3,418,040
3.00	<u>NITRATE SALT INVENTORY</u>	15,600	T	400		30.60	1.8	28,098	6,240,000	859,800		7,099,800
	BINARY SALT, $400/METRIC TON											
	$0.055/KG FOR HANDLING AND MELTING											
4.00	<u>PIPING AND INSTRUMENTATION</u>											
4.10	PIPE AND FITTINGS (THERMINOL, NITRATE SALT, AND NITROGEN)											
	- 700 mm (28 in.) PIPE - SHOP FAB, CS A106 SCH 40	171	M	363		32.40	9.02	1,540	61,960	49,900		111,860
	- 50 mm (2 in.) PIPE - SHOP FAB, CS A106 SCH 40	122	M	12		32.40	2.76	336	1,460	10,890		12,350
4.20	PIPE WELDS											
	- 700 mm (28 in.) PIPE - SHOP FAB, CS A106 SCH 40	47	EA			32.40	43.90	2,049		66,390		66,390
	- 50 mm (2 in.) PIPE - SHOP FAB, CS A106 SCH 40	27	EA			32.40	6.30	168		5,440		5,440
4.30	VALVES											
	- 700 mm (28 in.) VALVE, CS GATE, 300#, AIR OPERATED	4	EA	12,000					48,000			48,000
	- 700 mm (28 in.) VALVE, CS CHECK, 300#	2	EA	8,000					16,000			16,000
	- 50 mm (2 in.) VALVE, CS GLOBE, 300#, AIR OPERATED	1	EA	700					700			700
	- 50 mm (2 in.) VALVE, CS GATE, 300#, AIR OPERATED	2	EA	300					600			600
4.40	INSULATION											
	- 700 mm x 150 mm CALCIUM SILICATE, ALUMINUM JACKET	101	M		581						58,510	58,510
	- 700 mm (28 in.) PIPE FITTINGS	28	EA		620						17,360	17,360
	- 700 mm (28 in.) VALVES	6	EA		531						3,190	3,190
	- 50 mm x 150 mm CALCIUM SILICATE, ALUMINUM JACKET	52	M		126						6,560	6,560
	- 50 mm (2 in.) PIPE FITTINGS	27	EA		135						3,590	3,590
	- 50 mm (2 in.) VALVES	3	EA		115						350	350
4.50	PIPE SUPPORTS											
	- 700 mm (28 in.) SUPPORTS	6	EA	650					3,900			3,900
	- 50 mm (2 in.) SUPPORTS	18	EA	200					3,600			3,600
4.60	ELECTRIC HEAT TRACE											
	- 700 mm (28 in.) SALT LINE; 6 CABLES	91	M	295		32.40	10.80	979	26,780	31,720		58,500
	- 50 mm (2 in.) SALT LINE; 3 CABLES	42	M	148		32.40	5.40	226	6,190	7,320		13,510
	- VALVE BODIES	9	EA	450		32.40	24	216	4,050	7,000		11,050
	- NITRATE SALT PUMPS	2	EA	750		32.40	24	48	1,500	1,560		3,060
4.71	MISCELLANEOUS MATERIAL (PIPING)	1	LT						13,620			13,620
4.72	MISCELLANEOUS LABOR (PIPING)	1	LT			32.40	3,274	3,274		106,080		106,080

470 MWht Thermal Storage System Total Field Cost Estimate

Code	Description	Qty	Unit	Unit Cost Material	Unit Cost S/C	Labor Rate	MH Rate	Total MH	$ Material	$ Labor	$ S/C	$ Total
4.80	SALT STORAGE SYSTEM INSTRUMENTATION	1	LT	64,000		32.40	1,080	1,080	64,000	34,990		98,990
4.0	TOTAL PIPING AND INSTRUMENTATION								252,360	321,290	89,560	663,210
5.00	CIVIL AND STRUCTURAL											
5.10	ELEVATED PLATFORM											
	- STRUCTURAL STEEL	200	T	1,400		25.20	24	4,800	280,000	120,960		400,960
	- CONCRETE	139	M^3	75		25.20	1.92	267	10,420	6,730		17,150
	- FORMS	28	M^2	16		25.20	0.58	16	450	400		850
	- REBAR	14	T		700						9,720	9,720
	- EMBEDS	900	KG	2		25.20	0.10	90	1,350	2,270		3,620
	- EXCAVATION	100	M^3		3						250	250
	- BACKFILL	50	M^3		5						250	250
	- STEEL AND GRATING	13	T	1,600		25.20	24	312	20,800	7,860		28,660
5.0	TOTAL CIVIL AND STRUCTURAL								313,020	138,220	10,220	461,460
6.00	ELECTRIC EQUIPMENT AND BULK MATERIALS											
6.10	ELECTRIC EQUIPMENT											
	4.16 KV, 350 MVA SWGR - 1200A. OUTDOOR ACB WITH CUBICLE	1	EA	45,000		32.40	43	43	45,000	1,390		46,390
	LOAD CENTER XFMR, 4.16-.48KV, 20/175 KVA, AA, 55/65C RISE	1	EA	5,000		32.40	72	72	5,000	2,330		7,330
	OUTDOOR, 800A, 480V. AIR CIRCUIT BREAKER	2	EA	3,000		32.40	14	29	6,000	940		6,940
	OUTDOOR 480V. MOTOR CONTROL CENTER	1	EA	15,000		32.40	240	240	15,000	7,780		22,780
	125V. DC DISTRIBUTION PANEL	3	EA	2,000		32.40	0	1	6,000	30		6,030
	480V. POWER RECEPTACLES	6	EA	50		32.40	0	2	300	60		360
	LOCAL CONTROL STATIONS	2	EA	150		32.40	0	1	300	30		330
	480-208/120V DRY TYPE TRANSFORMERS 9 KVA 3PHASE	4	EA	600		32.40	0	1	2,400	30		2,430
	208/120V. 3PH. DISTRIBUTION PANELS	2	EA	1,500		32.40	0	1	3,000	30		3,030
	DUPLEX RECEPTACLES 120V.	6	EA	4		32.40	0	2	20	60		80
	SWITCHES 120V.	6	EA	4		32.40	0	2	20	60		80
	FIRE DETECTION SYSTEM	1	LT	3,000		32.40	384	384	3,000	12,440		15,440
	LIGHTING FIXTURES - LOW PRESSURE SODIUM	10	EA	250		32.40	2.88	29	2,500	940		3,440
	LIGHTING PANEL, 480/277 3PH., 4W 24 CIRCUIT, OUTDOOR	1	EA	1,100		32.40	29	29	1,100	940		2,040

470 MWht Thermal Storage System Total Field Cost Estimate

Code	Description	Qty	Unit	Unit Cost Material	Unit Cost S/C	Labor Rate	MH Rate	Total MH	$ Material	$ Labor	$ S/C	$ Total
6.42	ELECTRIC BULK QUANTITIES											
	CABLE TRAY, INCLUDING SUPPORTS, COVERS, FITTINGS, ETC.	100	M	33		32.40	3.46	346	3,280	11,210		14,490
	METALLIC CONDUIT	500	M	21		32.40	1.50	748	10,660	24,240		34,900
	NONMETALLIC CONDUIT	500	M	3.3		32.40	0.39	197	1,640	6,380		8,020
	CABLE - 600V. 1/C #6 AND LARGER	400	M	6.6		32.40	0.16	63	2,620	2,040		4,660
	CABLE - 600V. M/C #8 AND SMALLER	2,400	M	1.6		32.40	0.12	283	3,940	9,170		13,110
	CABLE - 1/C 5KV	200	M	20		32.40	0.24	47	3,940	1,520		5,460
	CABLE - INSTRUMENT AND SPECIAL	1,600	M	1.6		32.40	0.12	189	2,620	6,120		8,740
	CONNECTIONS - 600V AND INSTRUMENT/SPECIAL	525	EA	0.5		32.40	0.32	170	260	5,510		5,770
	CONNECTIONS - 5KV	6	EA	50		32.40	5.38	32	300	1,040		1,340
	LIGHTING CONDUIT	150	M	3.3		32.40	0.87	130	490	4,210		4,700
	COMMUNICATIONS CONDUIT	150	M	6.6		32.40	1.18	177	980	5,730		6,710
	LIGHTING WIRE - 1/C THHN	600	M	0.3		32.40	0.08	47	200	1,520		1,720
	LIGHTING WIRE - METALCLAD CABLE	600	M	3.9		32.40	0.63	378	2,360	12,250		14,610
	COMMUNICATIONS CABLE - MULTICONDUCTOR	100	M	3.3		32.40	0.16	16	330	520		850
	COMMUNICATIONS CABLE - METALCLAD	200	M	3.9		32.40	0.63	126	790	4,080		4,870
	GROUNDING WIRE, INCLUDING RODS, CADWELDS, AND PADS	200	M	6.6		32.40	0.47	94	1,310	3,050		4,360
	LIGHTNING PROTECTION - TOTAL PACKAGE	1	LT	2,000		32.40	240	240	2,000	7,780		9,780
6.40	TOTAL ELECTRIC EQUIPMENT AND BULK MATERIALS								127,360	133,430	0	260,790

Labor productivity (LP)	1.20
Direct wage rates:	
- Mechanical	17.00
- Civil/Structural	14.00
- Piping	18.00
- Electrical	18.00
Distributable costs: Fraction of direct labor cost (DIST)	0.80
Direct + indirect wage rates:	
- Mechanical (MECH)	30.60
- Civil/Structural (C/S)	25.20
- Piping (PIPE)	32.40
- Electrical (ELEC)	32.40