NREL National Renewable Energy Laboratory
Innovation for Our Energy Future
A national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy

Thermal Storage Commercial Plant Design Study for a 2-Tank Indirect Molten Salt System

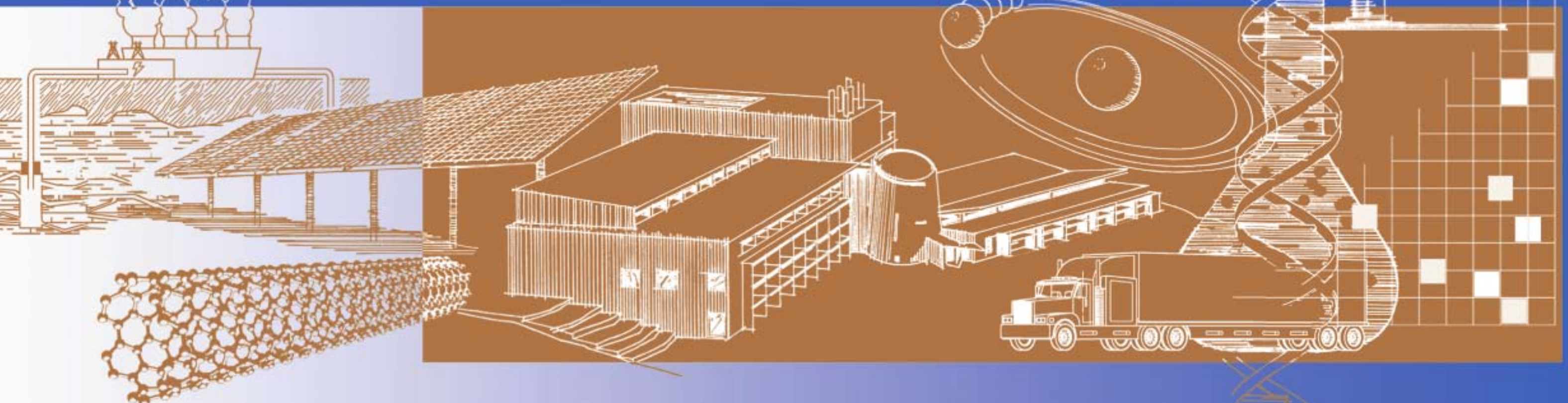
Final Report
May 13, 2002 — December 31, 2004

B. Kelly
Nexant, Inc.
San Francisco, California

D. Kearney
Kearney & Associates
Vashon, Washington

Subcontract Report
NREL/SR-550-40166
July 2006

NREL is operated by Midwest Research Institute • Battelle Contract No. DE-AC36-99-GO10337



Thermal Storage Commercial Plant Design Study for a 2-Tank Indirect Molten Salt System

Subcontract Report
NREL/SR-550-40166
July 2006

Final Report
May 13, 2002 — December 31, 2004

B. Kelly
Nexant, Inc.
San Francisco, California

D. Kearney
Kearney & Associates
Vashon, Washington

NREL Technical Monitor: H. Price
Prepared under Subcontract No. AAA-2-32432-01

National Renewable Energy Laboratory
1617 Cole Boulevard, Golden, Colorado 80401-3393
303-275-3000 • www.nrel.gov

Operated for the U.S. Department of Energy
Office of Energy Efficiency and Renewable Energy
by Midwest Research Institute • Battelle

Contract No. DE-AC36-99-GO10337

Available electronically at http://www.osti.gov/bridge

Available for a processing fee to U.S. Department of Energy
and its contractors, in paper, from:
U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831-0062
phone: 865.576.8401
fax: 865.576.5728
email: mailto:reports@adonis.osti.gov

Available for sale to the public, in paper, from:
U.S. Department of Commerce
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
phone: 800.553.6847
fax: 703.605.6900
email: orders@ntis.fedworld.gov
online ordering: http://www.ntis.gov/ordering.htm

Printed on paper containing at least 50% wastepaper, including 20% post-consumer waste

Thermal Storage Commercial Plant Design Study for a 2-Tank Indirect Molten Salt System

Final Report

Table of Contents

Contributors

Bruce Kelly, Nexant
Henry Price, NREL
David Kearney, K&A

[This page intentionally blank.]

A. Introduction and Background

The development of a near-term thermal energy storage (TES) option is a major breakthrough for parabolic trough solar power plant technology. The 2-tank indirect TES concept proposed by Nexant and SunLab is being implemented in the AndaSol 50 MWe (net) trough plant currently under development in Spain[i]. The project will have between 6-12 hours of TES depending on the economic optimization. In the U.S., Duke Solar is seriously considering this option where operational and economic factors favor its use. The economic optimum depends on the cost and performance of the TES system. The cost and performance of the indirect TES system are not independent because the size (cost) of the heat exchanger affects the physical size of the storage system and the performance of the turbine.

The objectives of this Task were to develop a set of conceptual designs for 2-tank molten salt thermal energy storage (TES) systems of varying thermal capacities for trough plants, and to estimate levelized electricity costs for optimized plant configurations with storage. The solar plant assumed in the analysis is the Solar Electric Generating System (SEGS)-type plant using VP-1 for the solar field heat transport fluid.

The availability of low cost and efficient thermal storage is one of the attributes of large-scale solar thermal technologies. Thermal storage allows electric energy to be dispatched at the times when it is needed most, and allows parabolic trough projects to achieve favorable capacity factors in excess of 50 percent. During the past two years, a significant effort has been undertaken to identify thermal storage technologies for parabolic trough plants[ii,iii]. The initial goal was to develop a thermal storage option that would be perceived as low risk, and could be used with confidence in near-term projects. An indirect 2-tank thermal storage system has been proposed which meets these requirements[iv v]. The system uses the following equipment: a cold storage tank, operating a nominal temperature of 290 °C; a hot storage tank, operating at a nominal temperature of 385 °C; a storage inventory of a mixture of binary nitrate salts; an oil-to-salt heat exchanger; and nitrate salt circulation pumps. During storage charging, heat is transferred from the collector field heat transport fluid to the nitrate salt through an oil-to-salt heat exchanger. During storage discharging, the fluid flows are reversed, and heat is transferred from the nitrate salt to the oil through the same heat exchanger.

The overall goal of this effort was to help transfer this TES technology from a prototype development status to a commercial status. The approach taken was to develop an industry/lab team to evaluate the potential TES designs for a near-term commercial plant. The team included a parabolic trough developer (Duke Solar), TES system designer (Nexant), and SunLab. Duke Solar defined the overall system requirements.

B. Reference Conditions

The reference solar plant assumed for the optimization includes:

- 55 MWe gross (or 50 MWe net) steam Rankine cycle power plant (100 and 66 bar)
- LS-2+ solar field using VP-1 HTF @ 391C
- Thermal storage of 2-12 hours equivalent full load capacity

The storage system consists of an oil-to-salt heat exchanger, a cold storage tank, a hot storage tank, and two circulation pumps. The storage medium is a mixture of 60 percent by weight sodium nitrate and 40 percent by weight potassium nitrate. The salt mixture, which was used successfully at the 10 MWe Solar Two central receiver project in southern California, offers a favorable combination of high density, low vapor pressure, moderate specific heat, low chemical reactivity, and low cost.

During a thermal storage charge cycle, a portion of the oil from the collector field is directed to the oil-to-salt heat exchanger, where the oil cools from a nominal inlet temperature of 391 °C to an outlet temperature of about 298 °C. Nitrate salt from the cold storage tank flows in a countercurrent arrangement through the heat exchanger. The salt is heated from an inlet temperature of 291 °C to an outlet temperature of 384 °C, and then stored in the hot storage tank. During a discharge cycle, the oil and salt flow paths are reversed in the oil-to-salt heat exchanger. Heat is then transferred from the salt to the oil to provide the thermal energy for the steam generator.

An optimization of the thermal storage system involves the assessment of numerous parameters, including the following:

- The inverse relationship between 1) the surface area, and cost, of the oil-to-salt heat exchanger, and 2) the quantity and cost of the storage inventory
- The inverse relationship between 1) the surface area of the oil-to-salt heat exchanger, and 2) the part load performance penalty of the Rankine cycle when operating from thermal storage.

C. Parametric Studies

A series of parametric studies was undertaken to determine the preferred design parameters for a thermal storage system with several thermal storage capacities, varying from 2 to 12 hours. Nexant conducted a series of parametric thermodynamic and capital cost studies on the Rankine cycle, the storage system, and the steam generator for the representative 50 MWe project[vi]. System designs and capital cost estimates were developed for 140 combinations of the following parameters:

- Rankine cycle live steam pressures of 101 bar and 66 bar
- Thermal storage capacities of 2, 4, 6, 9, and 12 hours of turbine operation at full load
- Oil-to-salt heat exchanger log mean temperature differences of 2 °C to 15 °C.

Based on the Nexant designs, NREL used the Excelergy model to calculate annual performance of each combination[vii]. For each design a parametric analysis was conducted to determine the optimum solar field size required to minimize the cost of energy. For this analysis, each of the 140 designs, plus the two no-storage cases, were modeled with 7 different solar field sizes. Care was taken to assure that the optimum solar field size was in the range of sizes selected. The optimum log-mean temperature difference (or heat exchanger size) that produced the lowest electric energy cost was found for each storage size for both power cycles.

The studies involved the following steps:

1) A representative computer model of the Rankine cycle was developed using the GateCycle program[viii].
2) The duties, overall heat transfer coefficients, surface areas, and pressure losses for the superheater, reheater, evaporator, and preheater of the steam generator were estimated using an in-house program
3) The duty, overall heat transfer coefficient, surface area, and pressure losses for the oil-to-salt heat exchanger were calculated for log mean temperature differences of 2 °C to 15 °C. The solar multiple of the collector field was specified by NREL. For a solar multiple > 1, the design condition for the oil-to-salt heat exchanger was during thermal storage charging, with a duty equal to the solar multiple value times the design duty of the steam generator.
4) Part load models were developed for the steam generator, the oil-to-salt heat exchanger, and the Rankine cycle to estimate the Therminol, nitrate salt, water, and steam temperature distributions during thermal storage discharging for heat exchanger approach temperatures of 2 °C to 15 °C.
5) Parametric cost estimates for the thermal storage tanks, the nitrate salt inventory, the oil-to-salt heat exchangers, and the nitrate salt pumps were developed for each of the heat exchanger approach temperatures.
6) The oil-to-salt heat exchanger approach temperature which provided the lowest unit storage cost, in $/kWht, was identified.
7) For each heat exchanger approach temperature, estimates of the plant capital cost, annual energy production, and levelized energy cost were developed to determine the preferred thermal storage design parameters.
8) Utilizing the NREL performance/cost model Excelergy, carry out performance analyses and electricity cost estimates for each heat exchanger approach temperature case to identify the optimum parameters.

D. Plant Design Conditions

The procedures for calculating the design point performance of the Rankine cycle, and estimating the surface areas and pressure losses in the steam generator and the oil-to-salt heat exchanger, are discussed in the following sections.

Rankine Cycle

The Rankine cycle design is a conventional, single reheat design with 5 closed and 1 open extraction feedwater heaters. The live steam pressure and temperature are 100 bar and 373 °C, respectively, and the reheat steam temperature is 373 °C. The principal design parameters are shown in Table 1, and the GateCycle flow diagram is shown in Figure 1.

Cold and hot reheat steam pressures, feedwater heater extraction pressures, feedwater heater terminal temperature differences, feedwater heater drain cooler approach temperatures, condenser pressure, circulating water temperature range, and condenser approach temperature are typical values.

Turbine expansion efficiencies, and the required live and reheat steam flow rates to achieve a gross output of 55.0 MWe, were calculated by GateCycle. Simultaneously, the low pressure turbine exhaust loss was adjusted manually to yield the desired gross cycle efficiency of 0.375.

Steam Generator

The surface areas and pressure losses of the preheater, evaporator, superheater, and reheater for the Therminol VP-1 steam generator were calculated using the following procedure in an Excel spreadsheet:

- Curve fits, as functions of temperature, were developed for the thermodynamic properties of the Therminol HTF, including density, viscosity, thermal conductivity, and specific heat.

- A Visual Basic version of the ASME Steam Tables was attached to the spreadsheet.

- Duties were calculated for the water/steam side of the heat exchangers from the pressure, temperature, and flow rate information provided by GateCycle. The required Therminol flow rate and inter-heat exchanger temperatures were calculated, by iteration, using the following constraints: a) the live and reheat steam temperatures were 20 ºC less than the collector field outlet temperature; b) the evaporator pinch point was set to 7.2 °C to achieve the specified preheater oil outlet temperature of 301.7 °C; c) the temperature of the feedwater at the outlet from the preheater was set to 0.5 °C less than the saturation temperature to improve the accuracy of the log mean temperature difference calculations for the evaporator; and d) the reheater oil flow rate was set to achieve the specified oil outlet temperature of 225.2 °C.

Table 1 Rankine Cycle Design Parameters

Gross generator output, MWe	55.04
High pressure turbine	
- Live steam pressure, bar	100.01
- Live steam temperature, C	373
- Live steam flow rate, kg/sec	60.231
- Expansion efficiency	0.852
Intermediate pressure turbine	
- Reheat steam pressure, bar	16.50
- Reheat steam temperature, C	373
- Reheat steam flow rate, kg/sec	54.044
- Expansion efficiency	0.900
Low pressure turbine	
- Expansion efficiency	0.850
- Exhaust loss, kJ/kg	22.9
Final feedwater temperature, C	235
Feedwater heater 1	
- Extraction line inlet pressure, bar	33.5
- Extraction line pressure loss, percent	0.0
- Terminal temperature difference, C	5.0
- Drain cooler approach, C	5.0
Feedwater heater 2	
- Extraction line inlet pressure, bar	14.0
- Extraction line pressure loss, percent	0.0
- Terminal temperature difference, C	5.0
- Drain cooler approach, C	5.0
Deaerator	
- Extraction line inlet pressure, bar	6.18
- Extraction line pressure loss, percent	0.0
- Feedwater outlet temperature, C	160.0
Feedwater heater 4	
- Extraction line inlet pressure, bar	3.04
- Extraction line pressure loss, percent	0.0
- Terminal temperature difference, C	4.0
- Drain cooler approach, C	5.0
Feedwater heater 5	
- Extraction line inlet pressure, bar	1.17
- Extraction line pressure loss, percent	0.0
- Terminal temperature difference, C	4.0
- Drain cooler approach, C	5.0
Feedwater heater 6	
- Extraction line inlet pressure, bar	0.37
- Extraction line pressure loss, percent	0.0
- Terminal temperature difference, C	4.0
- Drain cooler approach, C	5.0
Condenser	
- Inlet steam pressure, bar	0.080
- Surface area, m^2	5,659
Cooling towers	
- Dry bulb temperature, C	40
- Relative humidity, percent	20
- Approach, C	6.8
- Range, C	9.7

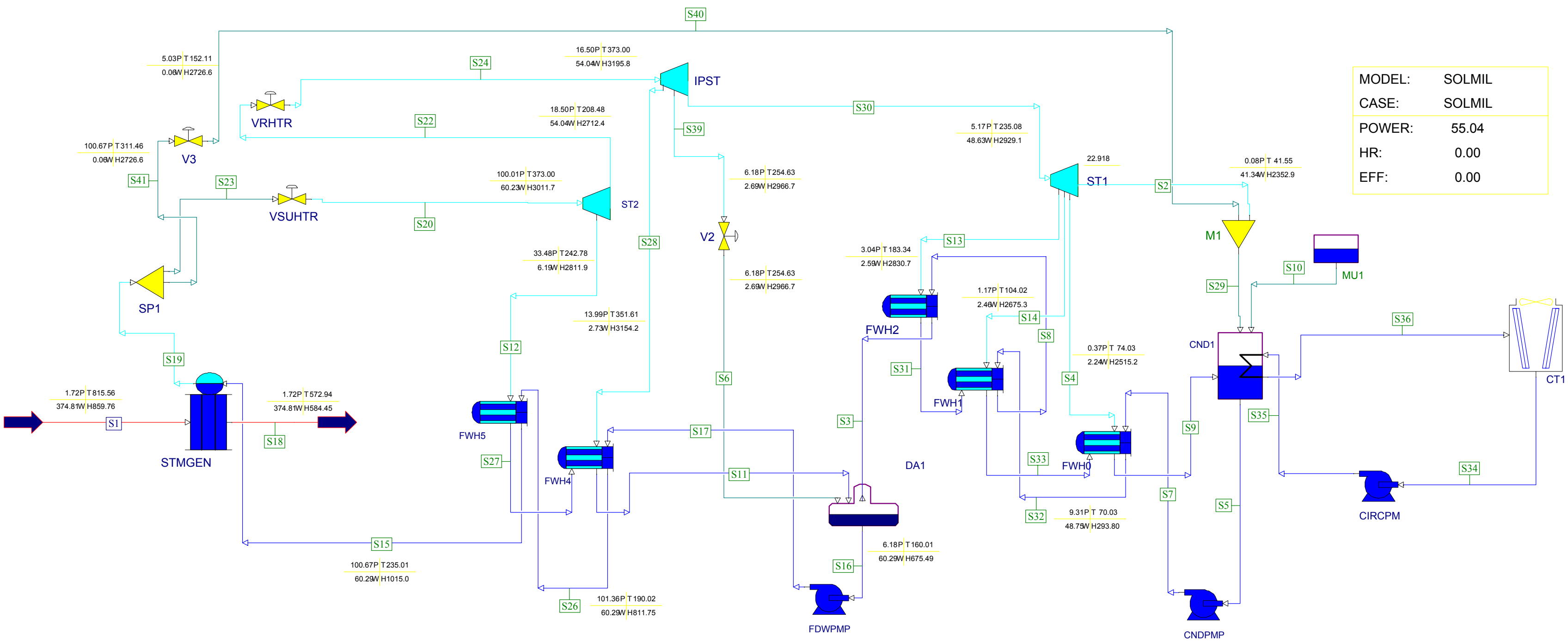


Figure 1 GateCycle diagram for 50 MWe Rankine Cycle Model

- Conventional shell and tube heat exchangers were selected for the preheater, the superheater, and the reheater. The high pressure water or steam was placed on the tube side, and the low pressure Therminol was placed on the shell side.

- A kettle evaporator, rather than a forced recirculation evaporator, was used. In general, kettle evaporators are normally less expensive than recirculation designs. Also, placing the heat transport fluid on the tube side posed only a minor potential for damage due to freeze-thaw cycles because the fluid has a favorable freezing point of only 11 °C.

- The Therminol flow through the reheater was assumed to be parallel to, and independent of, the Therminol flow through the superheater - evaporator - preheater combination. The Therminol flow from the reheater combined with the Therminol flow from the superheater - evaporator - preheater combination prior to returning to the collector field.

- Representative tube inside diameters, tube wall thicknesses, and water or steam velocities were selected for each heat exchanger, from which tube side heat transfer coefficients and pressure losses were calculated from published formulas.

- A tube baffle plate spacing was selected for each heat exchanger, from which the shell side heat transfer coefficients and pressure losses were calculated from published formulas.

- Representative tube side and shell side fouling coefficients were selected, and then combined with the convection coefficients and thermal resistance of the tube wall, to calculate overall heat transfer coefficients for each heat exchanger.

- Counterflow heat exchanger layouts, with two tube passes and two shell passes, were selected for the preheater, the superheater, and the reheater, and the surface areas were calculated as follows:

$$Area,\ m^2 = \frac{Q,\ kJ/sec}{(U_{overall},\ kJ/m^2 - {}^oC)(Log\ Mean\ Temperature\ Difference,\ {}^oC)}$$

- The overall heat transfer coefficient and surface area for the evaporator were calculated from published formulas for a kettle design.

The procedure was also subject to the following constraints:

- The maximum length of tube which could be fabricated was assumed to be 25 meters. If the calculated tube length exceeded 25 meters, two heat exchangers were used in series.

- The desired pressure loss on the shell side of the preheater, the superheater, and the reheater was in the range of 1.4 to 1.7 bar. If the pressure loss was less than or more than the desired value, the tube baffle spacing was decreased or increased, respectively. Once the desired pressure loss was obtained, the final heat exchange areas were calculated.

The steam generator configuration, showing fluid flow paths, fluid flow rates and temperatures, duties, heat exchanger areas, and pressure losses is illustrated in Figure 2 for a typical case.

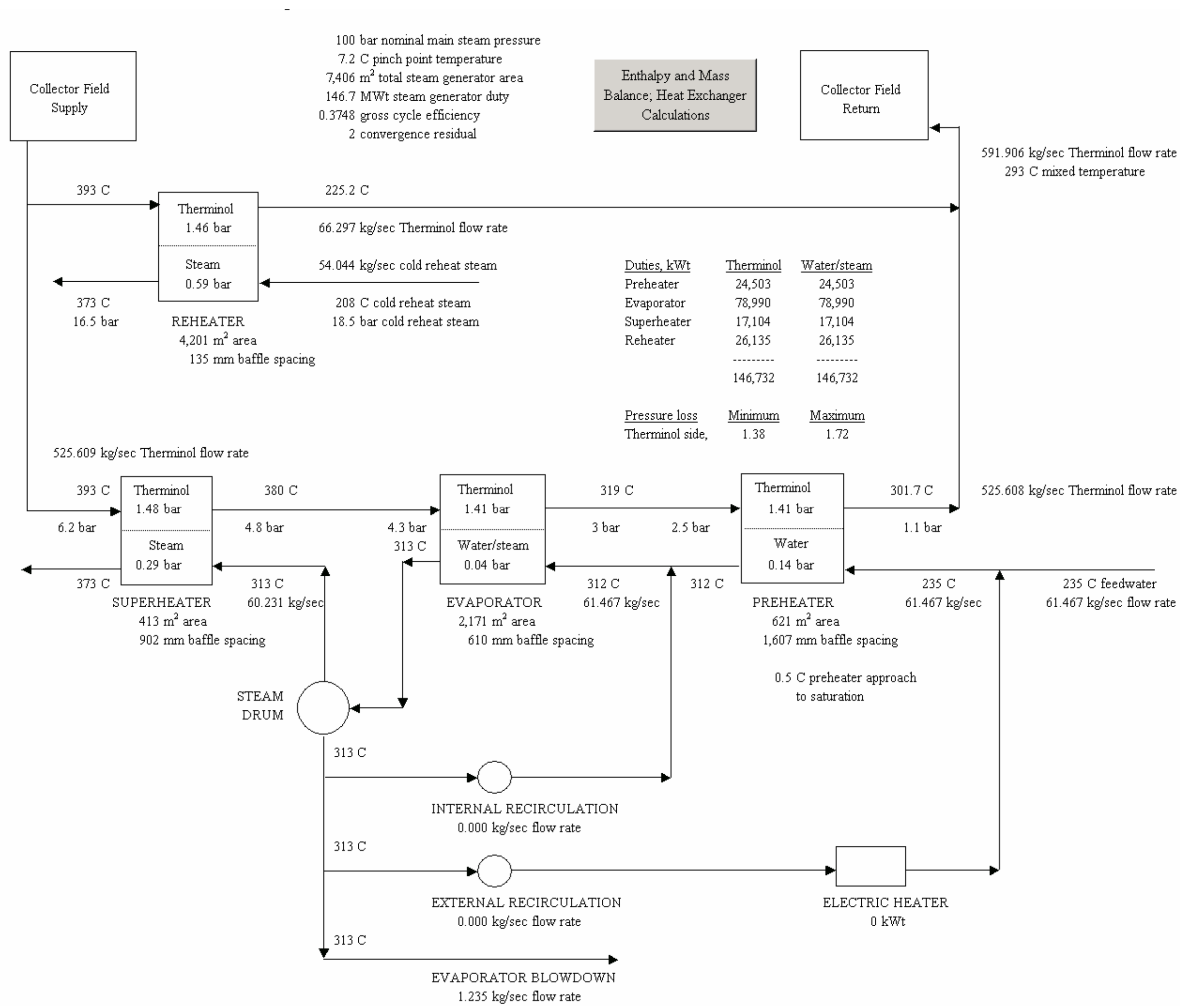


Figure 2 Typical 50 MWe Plant Steam Generator Configuration

Oil-to-Salt Heat Exchanger Design

The preferred oil-to-salt heat exchanger is a conventional shell and tube design[ix]. The surface area and pressure loss of the heat exchanger were calculated using the following procedure in an Excel spreadsheet:

- Curve fits, as functions of temperature, were developed for the thermodynamic properties of the Therminol and the nitrate salt, including density, viscosity, thermal conductivity, and specific heat.
- A duty was calculated from the heat balance information provided by GateCycle. The required fluid flow rates were calculated using the following constraints: a) the approach temperature at the hot end of the heat exchanger was the same as the approach temperature at the cold end; and b) the approach temperature was an independent variable in the optimization studies.
- The high pressure Therminol was placed on the tube side, and the low pressure nitrate salt was placed on the shell side.
- Representative values for the tube inside diameter, the tube wall thickness, and the Therminol velocity were selected, from which a tube side heat transfer coefficient and a pressure loss were calculated from published formulas.
- A tube baffle plate spacing was selected, from which the shell side salt velocity, heat transfer coefficient, and pressure loss were calculated from published formulas.
- Representative tube side and shell side fouling coefficients were selected, and then combined with the convection coefficients and thermal resistance of the tube wall, to calculate the overall heat transfer coefficient.
- A counterflow heat exchanger layout, with two tube passes and two shell passes, was assumed, and the surface area was calculated as follows:

$$Area, m^2 = \frac{Q, kJ/sec}{(U_{overall}, kJ/m^2 - {}^oC)(Log\ Mean\ Temperature\ Difference, {}^oC)}$$

The procedure was also subject to the following constraints:

- The maximum length of tube which could be fabricated was assumed to be 25 meters. If the calculated tube length exceeded 25 meters, multiple heat exchangers in series were used.
- For one heat exchanger, or for multiple heat exchangers in series, the desired total pressure loss on the shell side was in the range of 4.5 to 5.0 bar. If the pressure loss was less than or more than the desired value, the tube baffle spacing was decreased or increased, respectively. Once the desired pressure loss was obtained, the final heat exchange area was calculated.

E. Part Load Analysis

Steam Generator, Rankine Cycle, and Oil-to-Salt Heat Exchanger

The oil-to-salt heat exchanger imposes a temperature drop when energy is transferred into storage, and a temperature drop when energy is transferred out of storage. Thus, when the steam generator is operating from thermal storage, the temperature of the Therminol is less than the design value of 393 °C. The extent of the temperature decays is dependent on the log mean temperature difference selected for the heat exchanger. As expected, the decay in the steam generator source temperature leads to decays in both the live and reheat steam temperatures. Furthermore, if the flow rate of Therminol through the steam generator is fixed at the design flow rate, the thermal input to the steam generator is less than the design value. As a result, the flow rates of the live and reheat steam are lower than the design values, and the Rankine cycle operates under part load conditions.

To estimate the effects of decays in the Therminol source temperature on the performance of the Rankine cycle and the steam generator, the following iterative process was adopted:

- A trial Therminol, water, and steam temperature distribution through the superheater - evaporator - preheater combination was selected, from which the log mean temperature difference for each shell was calculated.

- A trial Therminol outlet temperature from the reheater was selected.

- Thermal duties on the water and steam sides of the four heat exchangers were calculated.

- Therminol flow rates through the reheater and the superheater - evaporator - preheater combination were calculated based on the trial temperature distribution and the thermal duties.

- The Therminol flow rate through the reheater was assumed to be fixed at the design flow rate.

- From the water, steam, and Therminol flow rates, tube side and shell side heat transfer coefficients were calculated. From these, overall heat transfer coefficients were calculated.

- The required log mean temperature difference for each shell was calculated, using the thermal duties on the water and steam sides and the overall heat transfer coefficients, as follows:

$$\textit{Required log mean temperature difference, } {}^{o}C = \frac{Q,\ kJ/sec}{(U_{overall},\ kJ/m^{2}\text{-}{}^{o}C)(Area, m^{2})}$$

- The calculated log mean temperature difference was developed using the standard formula:

$$\text{Calculated log mean temperature difference, } ^{o}C = \frac{\text{Greater temperature difference, } ^{o}C \text{ - Lesser temperature difference, } ^{o}C}{ln\left[\dfrac{\text{Greater temperature difference, } ^{o}C}{\text{Lesser temperature difference, } ^{o}C}\right]}$$

- The required and calculated values were then compared. If the values differed, a new trial Therminol temperature distribution was selected, and the process was repeated until the log mean temperature differences converged.

- The resulting values for the live steam temperature, the live steam flow rate, the reheat steam temperature, and the reheat steam flow rate were exported from the part load steam generator model to the GateCycle program.

- In GateCycle, the turbine was assumed to operate without throttling; i.e., the live steam pressure was determined by the steam flow rate through the turbine. The GateCycle program calculated the live steam pressure, the condenser pressure, the final feedwater temperature, and the gross generator output, and then exported these values back to the part load steam generator model.

- The process was repeated until constant values were obtained for the live steam pressure, live steam temperature, reheat steam pressure, reheat steam temperature, and final feedwater temperature.

The part load steam generator model was coupled with the oil-to-salt heat exchanger and the thermal storage system, as illustrated in Figure 3. The fluid temperature distribution in the oil-to-salt heat exchanger and the thermal storage system was calculated as follows:

- The hot salt tank temperature was set equal to the collector field outlet temperature minus the approach temperature of the oil-to-salt heat exchanger under full load conditions.

- The Therminol source temperature for the steam generator was set equal to the hot salt tank temperature minus the approach temperature of the oil-to-salt heat exchanger under part load conditions, with the duty of the heat exchanger equal to the duty of the steam generator.

- The cold salt tank temperature was set equal to the steam generator outlet temperature plus the approach temperature of the oil-to-salt heat exchanger under part load conditions.

- The collector field return temperature was set equal to the cold salt tank temperature plus the approach temperature of the oil-to-salt heat exchanger under full load conditions.

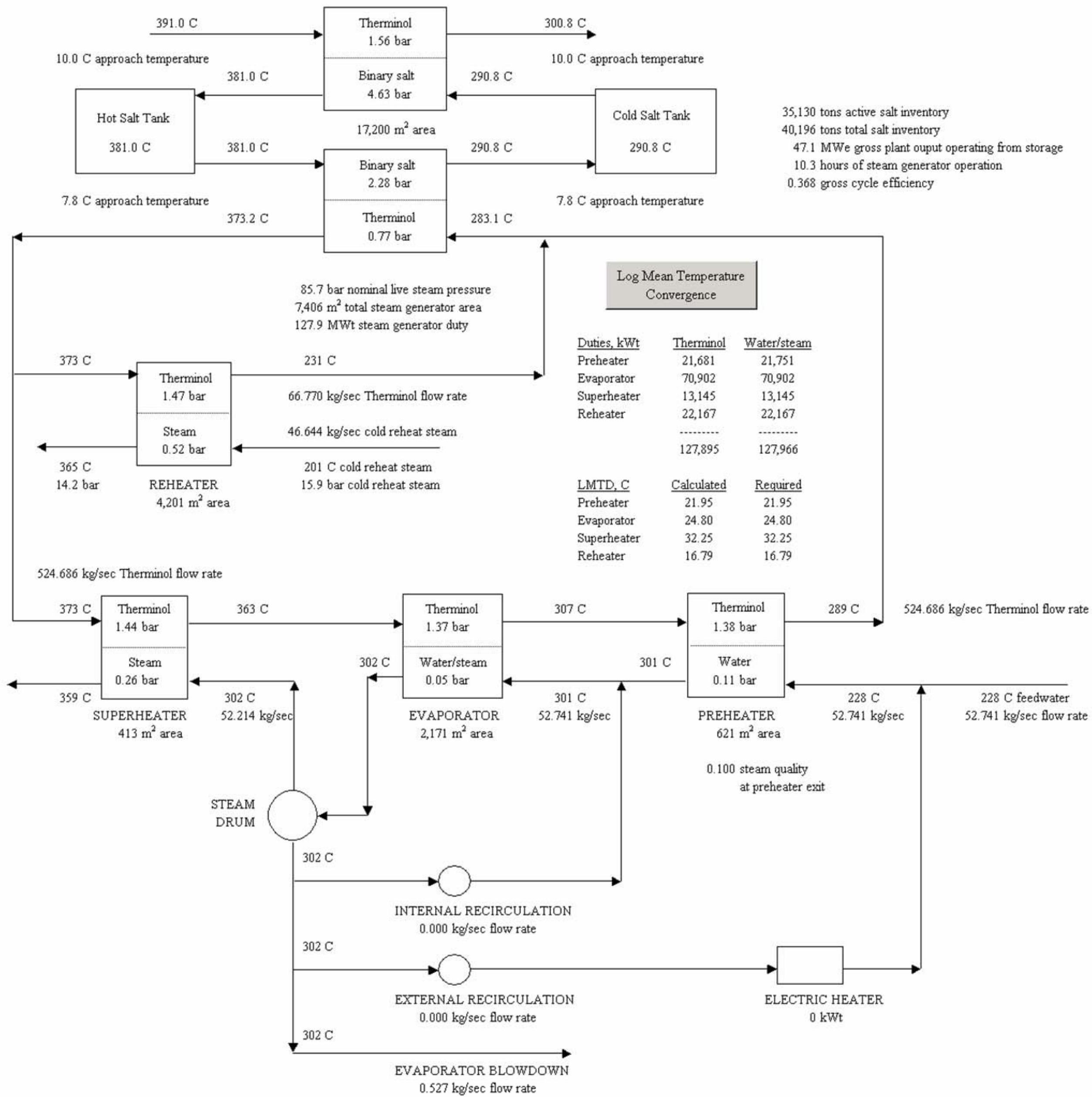


Figure 3 Part Load Steam Generator Operation
In Conjunction with the Oil-to-Salt Heat Exchanger and the Thermal Storage System

The calculation of the fluid temperature distribution within the thermal storage system was clearly an iterative process. First, a trial hot salt tank temperature was selected, from which the thermal input to the steam generator was calculated based on the following: a trial log mean temperature difference for the oil-to-salt heat exchanger; and an assumption the Therminol flow rate through the steam generator was equal to the flow rate under design point conditions. Using the procedures described above, a set of iterative calculations between the steam generator model and the GateCycle program was performed to reach convergence on the working fluid duties, temperatures, and pressures, and to calculate the Therminol exit temperature from the preheater. A trial log mean temperature difference for the oil-to-salt heat exchanger was selected, from which the cold salt tank temperature was calculated. The full load performance of the oil-to-salt heat exchanger was then analyzed to determine the 1) the required Therminol inlet temperature during thermal storage charging, and 2) the hot salt tank temperature. If hot salt tank temperature did not equal the desired value of [(391 °C) – (Log mean temperature difference for the oil-to-salt heat exchanger)], the trial hot salt temperature was revised, and the process repeated until all of the system part load requirements were satisfied.

Preheater Feedwater Exit Conditions

The fluid temperature distributions within the steam generator and the oil-to-salt heat exchanger generally converged in a smooth manner. However, for oil-to-salt heat exchanger approach temperatures greater than 4 °C, an enthalpy balance on the preheater showed the exit feedwater to be a mixture of saturated water and saturated steam.

Strictly speaking, a log mean temperature analysis of a heat exchanger is valid only if the specific heats of both the tube and the shell fluids are constant. For two phase flow, this is not the case. To improve the accuracy of the temperature distribution calculations, the part load model for the preheater was modified, as follows:

- The heat exchanger was divided into 100 sections along the length of the tubes.
- The temperature distribution on the shell side of the heat exchanger was estimated by assuming the enthalpy change of the Therminol in each section of the tube length was equal. The temperature distribution along the length of the tube was calculated by solving for the local fluid temperature which yielded the desired enthalpy. Since the specific heat of the Therminol does not change very much over the temperature range of interest, the temperature distribution between the entrance and the exit was essentially linear.
- A similar approach was taken on the water side to determine the point along the tube length at which the water reached saturation conditions. The temperature of the Therminol at this location was calculated from the distribution of Therminol temperatures. The method for estimating part load heat exchanger performance (i.e., comparing the required and the calculated log mean temperature differences) was then applied to the sensible heat transfer portion of the preheater. For the oil-to-salt heat exchanger log mean temperature differences of interest, sensible heat transfer occurred along 90 to 100 percent of the preheater tube length.
- The enthalpy of the water-steam mixture leaving the preheater was estimated from the fraction of latent heat transfer. This enthalpy, together with the saturation temperature of the feedwater, were used in the part load performance calculation for the evaporator.

A calculation of the water-steam mixture velocity at the exit from the preheater was also made to ensure the velocity was not excessive. Interestingly, the mixture velocities were typically below the design point feedwater velocity due to 1) lower feedwater flow rates under part load conditions, and 2) vapor fractions of only 1 to 9 percent.

The results of the part load steam generator and Rankine cycle analyses for oil-to-salt heat exchanger approach temperatures in the range of 2 °C to 15 °C are shown, for 100 bar and 66 bar steam conditions respectively, in Table 2 and Table 3. As expected, the hot salt tank temperature decreases uniformly, and the collector field supply temperature increases uniformly, with an increase in the oil-to-salt heat exchanger approach temperature. However, the temperature of the cold salt tank is, to a large degree, independent of the approach temperature. The effect can be traced to the live steam pressure. As the approach temperature increases, the thermal input to the steam generator and the live steam production rate both decrease. Since the turbine operates under sliding pressure, the live steam pressure also decreases. A reduction in live steam pressure is accompanied by reduction in the following: the saturation temperature in the evaporator; the Therminol temperature at the exit from the preheater; and the Therminol inlet temperature to the oil-to-salt heat exchanger during a charge cycle.

The analysis shows the impact of each heat exchanger size on power plant electric output and power cycle efficiency. The heat exchanger size is shown in the table but each size refers to a specific log-mean-temperature-difference (LMTD) between the HTF and the salt. Key to note is that the electric output and efficiency both drop when operating from storage. The magnitude of the drop depends on the size of the heat exchanger. The larger the heat exchanger, the smaller the impact on turbine performance. Note the heat exchanger with a LMTD of 2°C is 7.5 times the size of the heat exchanger with an LMTD of 15°C. The electric output only drops to 53.4 MWe and the steam cycle efficiency drops to 37.5% for TES with a LMTD of 2°C compared to 55 MWe and 37.9% when operating directly from the solar field (case 1). The electric output drops to 43.6 MWe and efficiency to 37.0% for the LMTD of 15°C case.

Also evaluated was the impact of TES on a lower pressure (66 bar) steam turbine. An earlier study had concluded that a lower pressure steam cycle would help reduce the cost of TES and result in a lower overall cost of energy. The lower pressure steam cycle allows a lower HTF return temperature to the solar field. This increases the temperature difference across the TES system, reducing the required storage volumes (and cost) for the same amount of thermal energy storage. This cycle also a 55 MWe plant (gross) when operating directly from the solar field and had a steam cycle efficiency of 35.5%. Similar reductions in efficiency and electric output occur when this cycle operated from TES, again depending on the size of the heat exchanger.

Table 2 Rankine Cycle, Steam Generator, and Oil-to-Salt Heat Exchanger Part Load Performance Values for 100 bar steam pressure

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12	Case 13	Case 14	Case 15
Oil-to-salt heat exchanger															
- LMTD, ^{o}C	N/A	2	3	4	5	6	7	8	9	10	11	12	13	14	15
- Heat exchanger area, m^2	N/A	80,860	49,625	34,755	26,408	21,201	17,693	15,116	13,252	11,789	10,559	9,623	8,746	8,069	7,465
- Salt side pressure loss, bar	N/A	5.14	5.20	5.09	5.03	5.10	5.05	5.06	4.99	4.97	5.03	4.90	5.06	5.01	5.00
- Oil side pressure loss, bar	N/A	4.05	3.36	3.42	3.45	3.18	2.90	2.70	2.38	2.14	1.93	1.77	1.62	1.51	1.41
System temperatures, ^{o}C															
- Cold tank	N/A	292	293	293	293	293	294	294	294	294	294	295	295	295	295
- Hot tank	N/A	389	388	387	387	386	385	384	383	382	381	380	379	378	377
- Collector field supply	294	294	295	296	298	299	300	301	302	303	304	305	307	308	309
- Preheater Therminol exit	294	295	294	293	292	291	290	290	289	288	287	286	285	284	283
Rankine cycle															
- Live steam pressure, bar	101.0	97.9	96.6	95.2	93.8	92.5	91.2	89.9	88.4	87.0	85.9	84.4	83.2	81.8	80.4
- Live steam temperature, ^{o}C	374	372	370	369	368	366	365	364	362	361	359	358	356	355	353
- Reheat steam temperature, ^{o}C	374	369	368	367	366	365	364	362	361	360	359	358	356	355	354
- Gross output, MWe	55.0	53.1	52.4	51.7	50.9	50.2	49.5	48.7	47.9	47.2	46.5	45.7	45.0	44.3	43.5
- Gross efficiency	0.379	0.375	0.375	0.374	0.374	0.374	0.373	0.373	0.372	0.372	0.372	0.372	0.371	0.371	0.370
Steam generator preheater															
- Exit steam quality	0.00	0.00	0.01	0.01	0.02	0.03	0.04	0.05	0.05	0.06	0.07	0.08	0.08	0.09	0.09
- Mixture exit velocity, m/sec	1.19	1.15	1.14	1.12	1.11	1.10	1.10	1.09	1.07	1.06	1.05	1.04	1.02	1.01	0.99
Unit storage cost, $/kWht															
- 2 hours	N/A	$80.31	$61.72	$52.86	$48.06	$45.29	$43.48	$42.28	$41.44	$40.89	$40.59	$40.26	$40.28	$40.20	$40.22
- 4 hours	N/A	$51.99	$42.84	$38.52	$36.26	$35.01	$34.26	$33.81	$33.54	$33.42	$33.43	$33.42	$33.60	$33.73	$33.90
- 6 hours	N/A	$42.43	$36.42	$33.62	$32.20	$31.46	$31.06	$30.86	$30.78	$30.79	$30.91	$31.00	$31.24	$31.43	$31.66
- 9 hours	N/A	$39.61	$34.39	$31.97	$30.77	$30.14	$29.83	$29.69	$29.65	$29.70	$29.84	$29.96	$30.64	$30.84	$31.49
- 12 hours	N/A	$38.69	$33.89	$31.68	$30.59	$30.03	$29.77	$29.66	$29.66	$29.73	$29.88	$30.01	$30.27	$30.48	$30.73

Table 3 Rankine Cycle, Steam Generator, and Oil-to-Salt Heat Exchanger Part Load Performance Values for 66 bar steam pressure

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9	Case 10	Case 11	Case 12	Case 13	Case 14	Case 15
Oil-to-salt heat exchanger															
- LMTD, ^{o}C	N/A	2	3	4	5	6	7	8	9	10	11	12	13	14	15
- Heat exchanger area, m^2	N/A	94,401	56,463	39,633	30,107	23,959	19,878	16,819	14,663	13,052	11,697	10,636	9,698	8,929	8,282
- Salt side pressure loss, bar	N/A	5.00	5.09	4.94	4.95	4.96	4.99	4.95	5.04	4.98	5.02	4.89	4.99	4.94	4.88
- Oil side pressure loss, bar	N/A	3.43	3.52	3.49	3.44	3.47	3.30	3.44	3.02	2.70	2.43	2.22	2.04	1.88	1.76
System temperatures, ^{o}C															
- Cold tank	N/A	271	272	272	272	272	273	273	273	274	274	274	275	275	275
- Hot tank	N/A	389	388	387	387	386	385	384	383	382	381	380	379	378	378
- Collector field supply	272	273	274	275	277	278	279	280	282	283	284	285	286	288	289
- Preheater Therminol exit	272	271	271	270	269	268	268	267	266	265	265	264	263	262	261
Rankine cycle															
- Live steam pressure, bar	66.5	64.7	64.0	63.1	62.5	61.7	60.8	60.1	59.3	58.5	57.7	56.9	56.1	55.3	54.6
- Live steam temperature, ^{o}C	374	372	370	369	367	366	364	363	361	360	358	357	355	354	352
- Reheat steam temperature, ^{o}C	374	368	367	366	365	364	362	361	360	358	357	356	354	353	352
- Gross output, MWe	55.0	53.3	52.8	52.0	51.4	50.8	50.1	49.4	48.7	48.1	47.4	46.7	46.0	45.4	44.7
- Gross efficiency	0.355	0.352	0.351	0.351	0.351	0.350	0.350	0.349	0.349	0.349	0.348	0.348	0.348	0.347	0.347
Steam generator preheater															
- Exit steam quality	0.00	0.00	0.00	0.00	0.02	0.03	0.03	0.04	0.05	0.06	0.06	0.07	0.08	0.08	0.09
- Mixture exit velocity, m/sec	1.13	1.09	1.08	1.06	1.07	1.07	1.05	1.05	1.04	1.04	1.02	1.02	1.01	1.00	1.00
Unit storage cost, $/kWht															
- 2 hours	N/A	$79.55	$58.18	$48.68	$43.53	$40.27	$38.20	$36.69	$35.76	$35.07	$34.58	$34.15	$33.96	$33.76	$33.66
- 4 hours	N/A	$49.52	$38.93	$34.26	$31.80	$30.27	$29.33	$28.68	$28.32	$28.09	$27.95	$27.85	$27.87	$27.88	$27.97
- 6 hours	N/A	$39.40	$32.40	$29.35	$27.78	$26.82	$26.26	$25.90	$25.73	$25.65	$25.63	$25.63	$25.72	$25.81	$25.95
- 9 hours	N/A	$36.50	$30.41	$27.76	$26.41	$25.60	$25.14	$24.84	$24.72	$24.67	$24.68	$24.71	$24.81	$24.92	$25.07
- 12 hours	N/A	$35.18	$29.57	$27.15	$26.20	$25.46	$25.05	$24.80	$24.69	$24.67	$24.69	$24.74	$24.85	$24.97	$25.14

F. Capital Cost Estimates

A series of parametric capital cost estimates were developed for storage systems with approach temperatures of 2 °C to 15 °C for the oil-to-salt heat exchanger. The parametric studies identified the approach temperature which yielded the lowest unit storage capital cost, in $/kWht.

Thermal Storage Tanks

The capital cost of the thermal storage tanks was developed as follows:

1) The cost of the tank shell was estimated by calculating the weight of the tank, and then applying a unit price. The unit price for carbon steel tanks, including material, shop fabrication, shipping, and field fabrication, was estimated to be $4.40/kg. Unit costs were derived from earlier central receiver cost studies[x] and Bechtel historical cost information. The tank weight was calculated from the following:

- The thickness of the wall at the bottom of the tank was calculated using the height of the tank, the density of the inventory fluid, the allowable material stress at the tank operating temperature, and conventional formulas for stresses in a hoop. The minimum wall thickness at the top was assumed to be 6 mm, and the wall thickness was assumed to vary linearly from the bottom of the tank to the top.
- The thickness of the floor was assumed to be 8 mm.
- The roof was assumed to be a self supporting dome, with a thickness of 6 mm.

2) The walls and roof were insulated with calcium silicate block insulation, which in turn was covered with a corrugated aluminum jacket for weather protection. The thickness of the insulation was assumed to vary linearly with the tank design temperature, increasing from a minimum value of 300 mm at a temperature of 290 ºC to a maximum value of 500 mm at a temperature of 565 ºC. The unit insulation costs were assumed to vary linearly with the thickness, increasing from a minimum value of $\$160/m^2$ at a thickness of 300 mm to a maximum value of $\$235/m^2$ at a thickness of 500 mm.

3) Starting from the bottom and moving up, the tank foundation consisted of a concrete slab, an insulating concrete slab, foam glass insulation, insulating firebricks, and a steel slip plate. The perimeter of the foundation was somewhat different, consisting of a ring wall of firebricks to support the additional weight of the walls and roof. An illustration of the foundation is shown in Figure 5. Costs for each of the elements were calculated from the following:

- The thickness of the concrete slab was assumed to be 610 mm, and 73 kg of reinforcing steel were provided in each cubic meter of concrete. Unit prices for the concrete were estimated to be $\$85/m^3$ and 1.3 installation hour/m^3, and unit prices for the reinforcing steel were estimated to be $0.80/kg and 0.022 installation hour/kg.

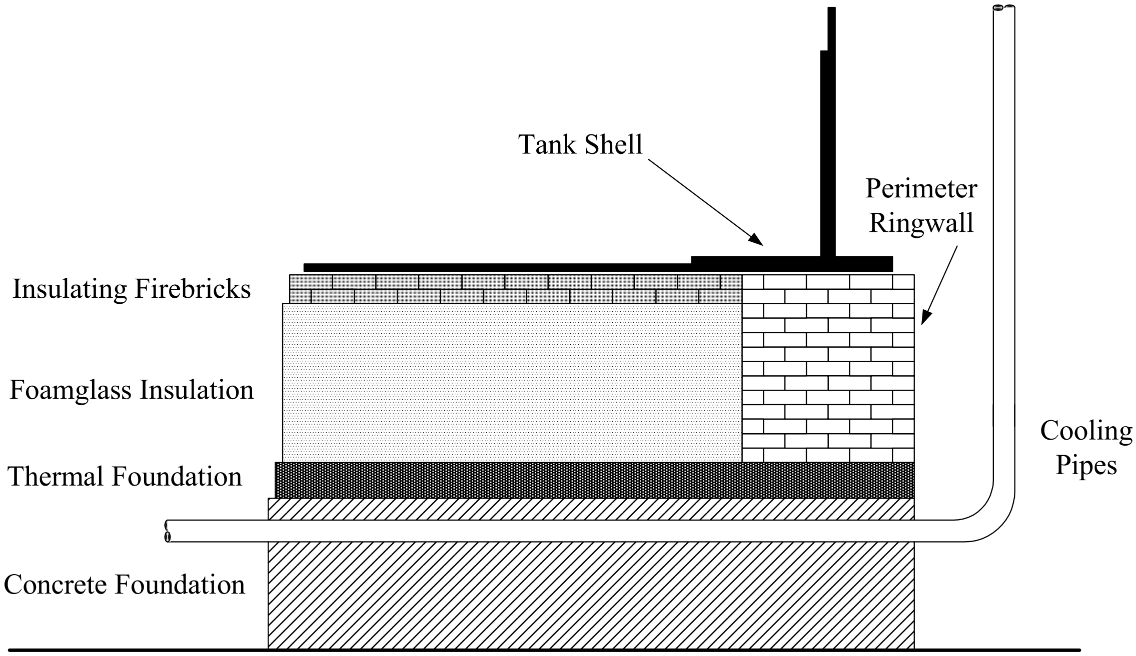


Figure 4 Thermal Storage Tank Foundation

- The thickness of the insulating concrete slab was assumed to vary linearly with the tank design temperature, increasing from a minimum value of 0 mm at a temperature of 290 ºC to a maximum value of 230 mm at a temperature of 565 ºC. Unit prices for the insulating concrete were estimated to be $100/m^3$ and 1.3 installation hour/m^3.
- The thickness of the foam glass insulation was assumed to vary inversely with the tank design temperature, decreasing from a maximum value of 400 mm at a temperature of 290 ºC to a minimum value of 300 mm at a temperature of 565 ºC. Subcontract unit prices for the insulation were estimated to be $356/m^3$.
- The thickness of the insulating firebrick was assumed to vary linearly with the tank design temperature, increasing from a minimum value of 0 mm at a temperature of 290 ºC to a maximum value of 165 mm at a temperature of 565 ºC. Unit prices for the firebricks were estimated to be $1 each. Installation costs, without mortar, were estimated to be 0.10 labor hours for each brick.
- The height of the perimeter ring wall was assumed to be the sum of the foam glass thickness and the insulating firebrick thickness. Unit prices for the firebricks were estimated to be $1 each. Installation costs, with mortar, were estimated to be 0.33 labor hours for each brick.
- The thickness of the steel slip plate was assumed to be 6 mm. Unit prices for the plate were estimated to be $1.30/kg, with 0.022 installation hour/kg.
- The foundation cooling pipes were assumed to be 200 mm, Schedule 20 carbon steel pipes located 1,200 mm on center. Unit prices for the pipe were estimated to be $2.20/kg, with 1.15 installation hour/m.

Storage Media

A delivered unit price for binary nitrate salt was estimated to be \$0.43/kg, based on budgetary information from the Solar Tres central receiver project. To the material price were added \$0.02/kg for the fuel to melt the salt, and \$0.05/kg for the labor to handle the salt once at the site, for a total installed cost of \$0.50/kg.

Oil-to-Salt Heat Exchangers

The unit price for the oil-to-salt heat exchanger was estimated to be \$146/m^2 (see [xi]). Installation labor costs were estimated to be 2.2 hours per metric ton, based on a unit weight of 200 kg/m^2.

Nitrate Salt Pumps

Budgetary prices for the nitrate salt pumps heat exchangers were obtained from the Solar Two project and the Solar Tres project. A regression analysis of the data provided the following equation for the unit cost of cold salt pumps:

$$\text{Unit cost, \$/kWe} = \$14{,}720\ (\text{Pump motor power, kWe})^{-0.4488}$$

A regression analysis of the data for the hot salt pumps provided the following:

$$\text{Unit cost, \$/kWe} = \$5{,}512\ (\text{Pump motor power, kWe})^{-0.1845}$$

The installation labor rate was estimated to be 100 hours per pump for motor ratings below 75 kWe, 300 hours for motor ratings between 75 and 750 kWe, and 500 hours for motor ratings between 750 and 1,500 kWe.

Nitrate Salt Piping

Unit costs for the nitrate salt piping costs were developed on the following basis:

- Pipe material costs were estimated to be \$2.20/kg for carbon steel, \$4.40/kg for ferritic steel, and \$6.60/kg for stainless steel lines
- Unit pipe weights were calculated from the design system pressures, maximum allowable material stresses, and conventional hoop stress formulas.
- Installation labor rates for material handling, lineup, tack welds, and production welds were developed from Bechtel standards
- Allowances for hangers, supports, inspection, and testing were developed as factors of the material and installation costs
- Unit electric heat trace power rating were developed from a differential equation which calculated temperature rise rates for the pipe material and thermal insulation. A preheat time of 2.0 hours was assumed, with initial and final pipe temperatures of 0 ºC and 175 ºC, respectively. The number of heat

trace cables was calculated from a maximum unit power rating of 165 W/m per cable, and 2 spare cables were installed with the active cables. Unit heat trace costs were estimated to be $50/m-cable, including the cables, control thermocouples, electric power supply to the cables, and installation.

- Pipe thermal insulation costs were developed from budgetary price information provided to the Solar Two project. Insulation thicknesses were a function of the pipe diameter and fluid temperature, and ranged from a minimum of 150 mm to a maximum of 200 mm.

Summary Investment Cost Plots

The total storage system unit cost results of the parametric cost analyses are summarized in Table 2 and Table 3, and illustrated in Figure 5 and Figure 6. Table 4 shows the thermal storage system cost breakdown for storage capacities of 2 to 12 hours for the 100 bar case at an LMTD of 7°C.

Table 4 Thermal Storage System Costs
101 Bar Rankine Cycle and 7°C Oil-To-Salt Heat Exchanger LMTD

	2 Hour	4 Hour	6 Hour	9 Hour	12 Hour
Cold Tank					
- Tank	$747,000	$1,427,000	$2,097,000	$3,091,000	$4,193,000
- Thermal insulation	$239,000	$376,000	$495,000	$659,000	$990,000
- Foundation	$390,000	$635,000	$858,000	$1,171,000	$1,716,000
Hot tank					
- Tank	$796,000	$1,526,000	$2,244,000	$3,311,000	$4,488,000
- Thermal insulation	$281,000	$442,000	$583,000	$776,000	$1,166,000
- Foundation	$443,000	$734,000	$1,002,000	$1,382,000	$2,004,000
Storage Media	$4,465,000	$8,931,000	$13,396,000	$20,094,000	$26,792,000
Oil-To-Salt Heat Exchangers	$2,865,000	$2,865,000	$2,865,000	$3,780,000	$4,653,000
Nitrate Salt Pumps	$1,420,000	$1,420,000	$1,420,000	$1,693,000	$1,847,000
Balance of Storage System	$1,165,000	$1,836,000	$2,496,000	$3,596,000	$4,785,000
	-------------	-------------	-------------	-------------	-------------
	$12,811,000	$20,192,000	$27,456,000	$39,553,000	$52,634,000

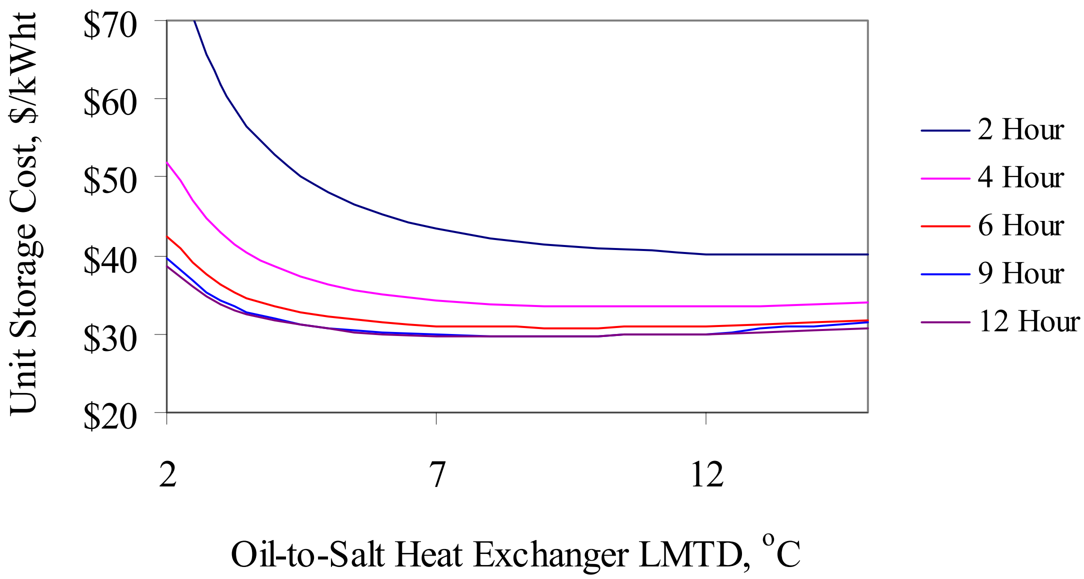


Figure 5 Effect of LMTD on unit cost for several storage capacities, at 100 bar

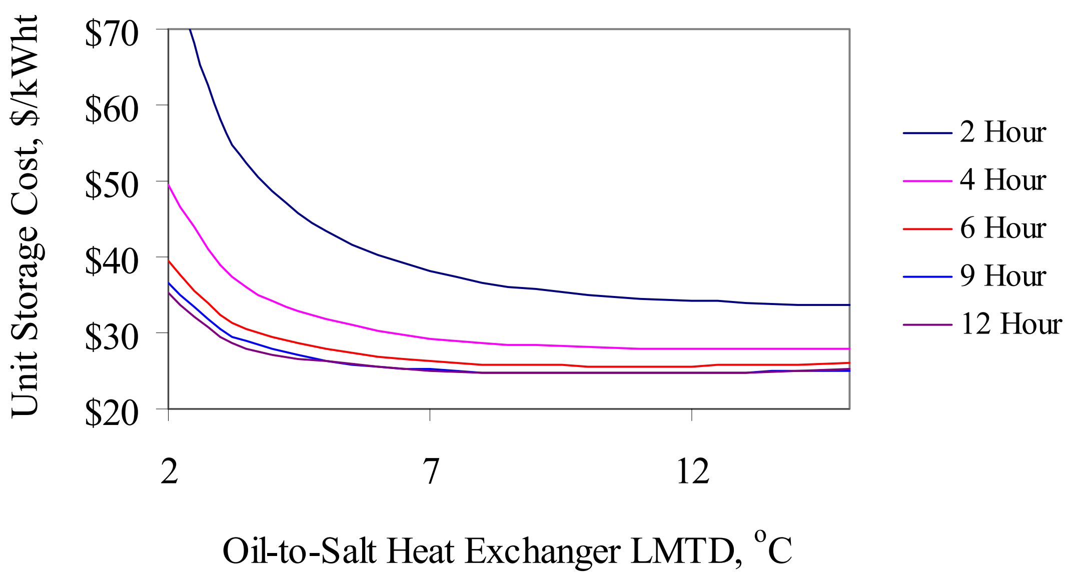


Figure 6 Effect of LMTD on unit cost for several storage capacities, at 66 bar

G. Annual Performance and Electricity Cost

NREL adapted the Excelergy model for this study to more accurately model the performance impact of operation from thermal storage. For the analysis, the levelized cost of energy was used as the figure of merit to compare each design. Figure 7 below shows the results for the 6-hour storage case where seven solar field sizes were evaluated with each of the TES heat exchanger sizes. The optimum 6 hour TES case has a solar field size of about 460,000 m^2 of solar field with an LMTD of about 7°C.

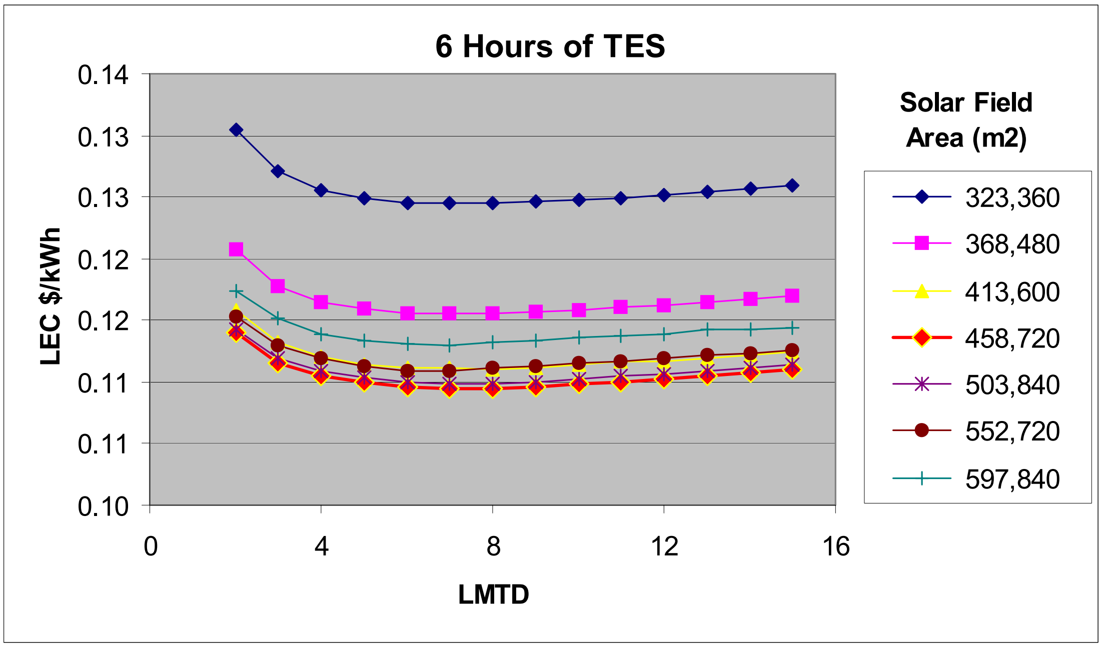


Figure 7 LMTD Optimization for 7 hours of TES, 55 MWe (gross), 101 bar steam cycle

Figure 8 shows the LEC for each storage volume and each heat exchanger size. The LMTDs that provide the minimum LEC are indicated on the graph for each TES capacity. Larger storage sizes are optimized with larger heat exchangers (lower LMTDs). Note that the curve is fairly flat and the minimum LEC occurs over a fairly wide range of LMTDs.

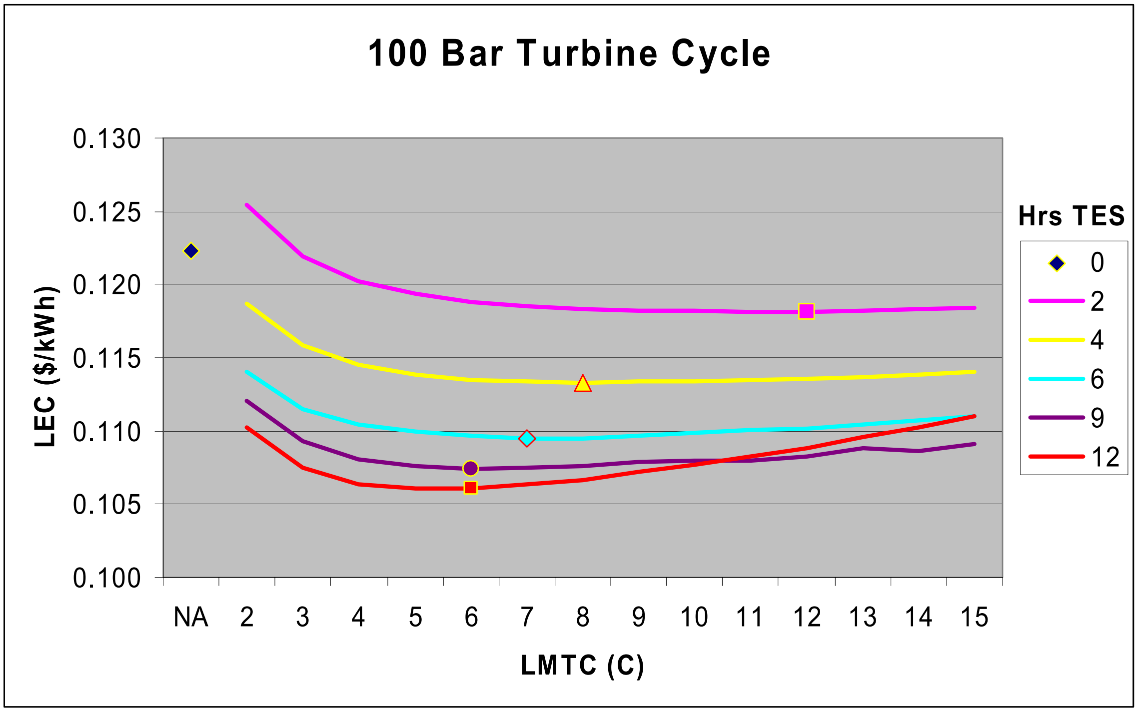


Figure 8 TES Optimization for 55 MWe Trough Plant, 101 bar steam cycle

Also important to note is that the LEC for the plant decreases with capacity. Table 5 compares the cost of energy for the plant with the minimum TES cost with the plant with the minimum LEC. The optimum LEC usually occurs with a larger TES heat exchanger (lower LMTD). However, the LEC for the optimized design is not substantially lower than the lowest cost TES design (0.1% - 0.6%).

Table 5 TES Design Comparison – Minimum TES Cost Verses Minimum LEC

Thermal Energy Storage (hours)	2	4	6	9	12
System with minimum TES cost					
Heat Exchanger LMTD (°C)	14	10	9	9	8
TES System Cost ($/kWht)	40.20	33.42	30.78	30.78	29.66
LEC ($/kWh)	0.1183	0.1134	0.1097	0.1079	0.1067
System with minimum LEC					
Heat Exchanger LMTD (°C)	12	8	7	6	6
TES System Cost ($/kWht)	40.26	33.81	31.06	30.14	30.03
LEC ($/kWh)	0.1182	0.1133	0.1095	0.1074	0.1061

Figure 9 and Table 6 show the comparison of cost of energy for 66 and 100 bar turbine steam cycles. In this study the 100 bar turbine cycle appears to result in a lower cost of energy than the 66 bar system. This is a different result than was observed in the earlier K&A study [3]. Additional comparisons with the earlier work is needed to understand this difference.

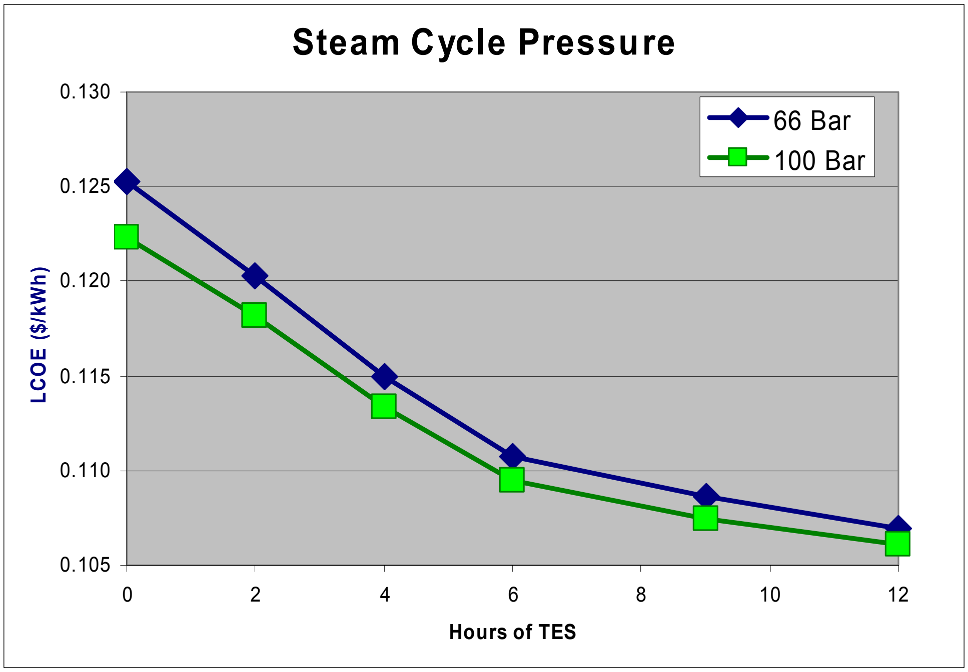


Figure 9 Comparison of LEC for 55 MWe plants with 66 and 100 bar turbine steam cycles

Table 6 TES Design Comparison – Minimum TES Cost Verses Minimum LEC

Thermal Energy Storage (hours)	2	4	6	9	12
100 Bar Turbine Cycle					
TES System Cost ($/kWht)	40.26	33.81	31.06	30.14	30.03
LEC ($/kWh)	0.1182	0.1133	0.1095	0.1074	0.1061
66 Bar Turbine Cycle					
TES System Cost ($/kWht)	40.26	33.81	31.06	30.14	30.03
LEC ($/kWh)	0.1203	0.1150	0.1108	0.1086	0.1069

The SEGS plants allow the turbine to operate up to 115% of design turbine output. This analysis also looked at the impact on LEC of allowing the turbine to operate up to 115% of design output, assuming no change in power cycle equipment cost. Figure 10 shows the impact on plant LEC as compared to the plant that is constrained to a maximum of 100% turbine output. The overall LEC is lower in part because the plant is optimized to a larger solar field sizes.

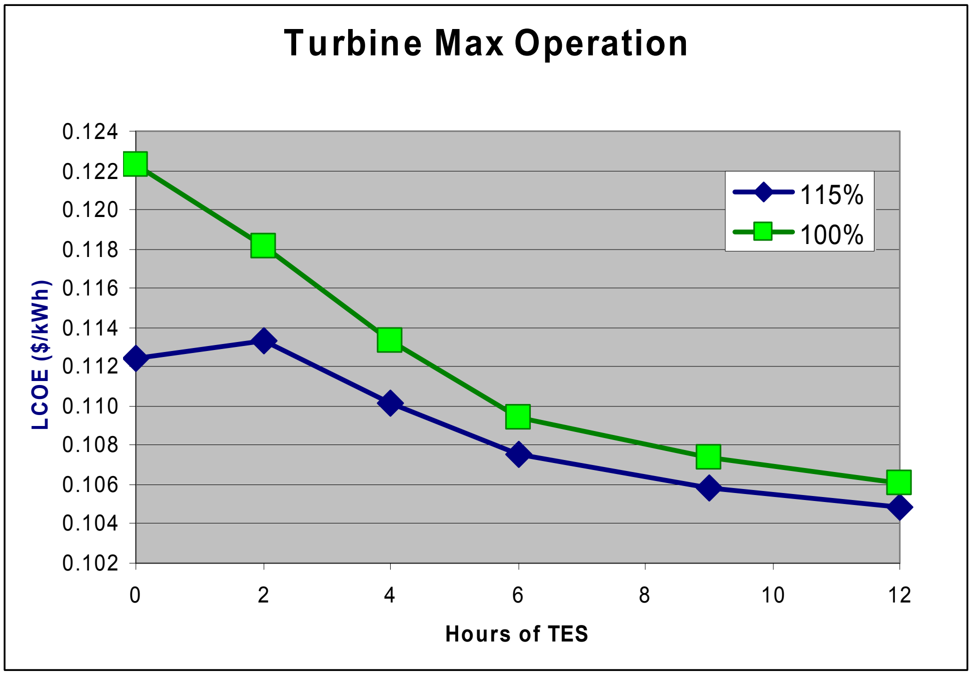


Figure 10 Impact on LEC of Turbine Over Design Operation

Conclusions

- The performance impact of the TES system must be considered in the overall cost optimization of the power plant. The lowest cost TES design does not correspond to the lowest cost of electricity.
- Although the TES system cost is lower for a 66 bar turbine power cycle, the lower efficiency of the 66 bar turbine results in a higher cost of solar energy. This is the opposite conclusion to an earlier study.
- For the 50 MWe (net) trough plant, the indirect thermal storage system can help reduce the cost of electricity from the plant.
- In order to develop an optimum solar plant design, it is important to conduct a detailed design analysis that accounts of costs, economics, and plant performance in order to determine the optimum TES configuration and solar field size. The analysis above was done to minimize the LEC, however other economic figures of merit (benefit/cost ratio, net present value or internal rate of return) would likely be used for a real project. These would likely result in different optimum TES configurations and solar field sizes.

Endnotes

i Aringhoff, R., and M., Geyer, "AndaSol- 50MW Solar Plants with 9 Hour Storage for Southern Spain," Proceeding of the 11th SolarPACES International Symposium on Concentrated Solar Power and Chemical Energy Technologies, September 4-6, 2002, Zurich, Switzerland.

ii Hermann, U. "Survey of Thermal Storage for Parabolic Trough Power Plants," NREL/SR-550-27925. NREL, Golden, CO, September 2000.

iii Engineering Evaluation of a Molten Salt HTF in a Parabolic Trough Solar Field. Task 6 Report. Final System Performance and Cost Comparisons. Prepared by Kearney and Associates for NREL, August 2001.

iv Nexant Inc., "Thermal Storage Oil-to-Salt Heat Exchanger Design and Safety Analysis", Task Order Authorization Number KAF-9-29765-09, San Francisco, California, March 22, 2001.

v Nexant Inc., "USA Trough Initiative: Thermal Storage Oil-to-Salt Heat Exchanger Design and Safety Analysis", , National Renewable Energy Laboratory Task Order Authorization Number KAF-9-29765-09, San Francisco, California, March 22, 2001.

vi Nexant Inc., "USA Trough Initiative: Rankine Cycle, Steam Generator, and Two-Tank Thermal Storage Parametric Studies", , National Renewable Energy Laboratory Task Order Authorization Number KAF-9-29765-09, San Francisco, California, September 27, 2002.

vii Price, H., "Heat exchanger Size Optimization for Indirect 2-Tank Molten-Salt Heat Thermal Storage System for Parabolic Trough Plants," National Renewable Energy Laboratory, Golden, CO, September 27, 2002.

viii GateCycle Program, Version 5.34, GE Enter Software, Inc. and the Electric Power Research Institute.

ix "Thermal Storage Oil-to-Salt Heat Exchanger Design and Safety Analysis", Task Order Authorization Number KAF-9-29765-09, Nexant Inc. (San Francisco, California), March 22, 2001.

x "Investigation of Thermal Storage and Steam Generator Issues", Bechtel Corporation (San Francisco, California), Sandia National Laboratories Report SAND93-7084, August 1993.

xi "USA Trough Initiative: Thermal Storage Oil-to-Salt Heat Exchanger Design and Safety Analysis", Nexant Inc. (San Francisco, California), National Renewable Energy Laboratory Task Order Authorization Number KAF-9-29765-09, March 22, 2001.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. REPORT DATE *(DD-MM-YYYY)*
July 2006

2. REPORT TYPE
Subcontract Report

3. DATES COVERED *(From - To)*
13 May 2002 - 31 December 2004

4. TITLE AND SUBTITLE
Thermal Storage Commercial Plant Design Study for a 2-Tank Indirect Molten Salt System: Final Report

5a. CONTRACT NUMBER
DE-AC36-99-GO10337

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
B. Kelly (Nexant, Inc.) and D. Kearney (Kearney & Associates)

5d. PROJECT NUMBER
NREL/SR-550-40166

5e. TASK NUMBER
CP06.1301

5f. WORK UNIT NUMBER

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

Nexant, Inc.
101 Second St., 11th Floor
San Francisco, CA 94105

Kearney & Associates
PO Box 2568
Vashon, WA 98070

8. PERFORMING ORGANIZATION REPORT NUMBER
AAA-2-32432-01

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Renewable Energy Laboratory
1617 Cole Blvd.
Golden, CO 80401-3393

10. SPONSOR/MONITOR'S ACRONYM(S)
NREL

11. SPONSORING/MONITORING AGENCY REPORT NUMBER
NREL/SR-550-40166

12. DISTRIBUTION AVAILABILITY STATEMENT
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

13. SUPPLEMENTARY NOTES
NREL Technical Monitor: H. Price

14. ABSTRACT *(Maximum 200 Words)*
Subcontract report by Nexant, Inc., and Kearney and Associates regarding a study of a solar parabolic trough commercial plant design with 2-tank indirect molten salt thermal storage system.

15. SUBJECT TERMS
concentrating solar power; solar parabolic trough; solar; solar thermal electricity

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
Unclassified	Unclassified	Unclassified	UL		

19b. TELEPONE NUMBER (*Include area code*)