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INVESTIGATION OF THERMAL STORAGE AND STEAM GENERATOR ISSUES

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ABSTRACT

A review and evaluation of steam generator and thermal storage tank designs for commercial nitrate salt technology showed that the potential exists to procure both on a competitive basis from a number of qualified vendors. The report outlines the criteria for review and the results of the review, which was intended only to assess the feasibility of each design, not to make a comparison or select the best concept.

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BACKGROUND

In 1986, two utilities, Arizona Public Service Company (APS) and Pacific Gas and Electric Company (PG&E), entered into a cooperative agreement with the Department of Energy to define the first generation of commercial central receiver power plants. The study, entitled "Solar Central Receiver Technology Advancement for Electric Utility Applications" (and called the "Utility Studies" for convenience), had two phases.

The goal of Phase I was to develop a consensus on the near term commercial plant design, with an emphasis on the selection of the preferred receiver (sodium or nitrate salt; cavity or external). The study concluded that a 100 MWe plant with a cylindrical nitrate salt receiver and a surround heliostat field was the preferred commercial design (Ref. 1-1). To simplify the study, the designs of various equipment items were fixed at the beginning so as not to divert attention from the receiver optimization studies. In the thermal storage system, this included a stainless steel hot salt thermal storage tank with external insulation. In the steam generation system, a forced recirculation design with U-tube/U-shell heat exchangers was selected.

The goal of Phase II was to establish a development plan to commercialize the technology. The study concluded that the conversion of the 10 MWe Solar One pilot plant to nitrate salt technology was the lowest cost approach to commercialize the technology (Ref. 1-2). During the study, budgetary quotes were requested from two thermal storage tank vendors and three potential steam generator vendors. Both tank vendors proposed stainless steel hot salt tanks with external insulation. However, an alternate hot salt tank design using a carbon steel shell and internal insulation was not evaluated during the study. Furthermore, two of the three heat exchanger vendors proposed kettle boiler concepts for the steam generator. Thus, there were several differences between the equipment designs for the first commercial plant and the 10 MWe plant that was to precede it.

INVESTIGATION OF THERMAL STORAGE AND STEAM GENERATOR ISSUES

Late in 1991, Southern California Edison Company organized a group of utilities and government organizations, and submitted a proposal to DOE to convert the Solar One plant to nitrate salt technology (Solar Two). The purpose of the project was to reduce the perceived risk in building the first commercial 100 MWe plant. To this end, Solar Two needed to duplicate the technical features of the first commercial plant as closely as possible.

The question arose: What should be the steam generator and thermal storage tank designs in the first commercial plant? This study sought a partial answer by reviewing potential equipment designs and identifying those which would be feasible. The approach involved the following steps:

- Subcontracts were placed with four heat exchanger vendors to examine the full range of steam generator options, as follows:

- ABB Lummus Heat Transfer: kettle evaporator with U-tube/straight shell heat exchangers
- Struthers Wells Corporation: kettle evaporator with U-tube/straight shell heat exchangers
- Foster Wheeler Development Corporation: natural circulation evaporator with straight tube/straight shell heat exchangers
- Babcock and Wilcox Company / Science Applications International Corporation (B&W / SAIC): forced recirculation evaporator with U-tube/U-shell heat exchangers

The statement of work for the steam generator vendors is presented in Appendix A

- Subcontracts were placed with three thermal storage tank vendors to examine the alternate hot salt tank designs, as follows:

  - Chicago Bridge and Iron Technical Services Company (CBI): stainless steel tank with external insulation
  - Pitt-Des Moines, Inc. (PDM): stainless steel tank with external insulation
  - S. N. Technigaz: carbon steel tank with internal refractorv insulation

  Note that only hot tank designs were evaluated, because it is generally agreed that the cold tank would be fabricated from carbon steel and use external insulation. The statement of work for the tank vendors is presented in Appendix B

- The vendors developed conceptual designs and cost estimates for the equipment required in the first 100 MWe commercial plant. Summaries of the heat exchanger and tank vendor designs and cost estimates are shown in Tables 1-1 and 1-2, respectively

- Bechtel reviewed the vendor information, assessed the technical feasibility of each design, and determined whether the equipment would be suitable for the commercial plant.

It should be noted that the purpose of this study was limited to a basic assessment of the feasibility of each design. The assessment addressed the following:

- Can a steam generator using a kettle evaporator be fabricated for a 100 MWe plant?

- Will a large, flat bottom, stainless steel tank be suitable in solar power plant service?

- Are the vendors prepared to offer warranties and budgetary cost estimates?

The study was not intended as a detailed comparison of alternate designs or a selection of the best concept. For example, each steam generator will have different overnight temperature control and morning startup requirements. The influence of these requirements on annual plant performance and revenue requirements, and the selection of the preferred concept, is discussed below under FUTURE ACTIVITIES.

Table 1-1
STEAM GENERATOR DESIGNS AND COST ESTIMATES

	Preheater	Evaporator	Superheater	Reheater	Comments[1]
ABB Lummus Heat Transfer					
- Type	UT/SS[2]	Kettle[3]	UT/SS	UT/SS	2 superheater shells in series; $4,150,000
- Shell fluid	Salt	Water/steam	Salt	Steam	
- Tube fluid	Water	Salt	Steam	Salt	
Struthers Wells Corporation					
- Type	UT/SS	Kettle	UT/SS	UT/SS	High pressure water or steam on shell side; $5,240,000
- Shell fluid	Water	Water/steam	Steam	Steam	
- Tube fluid	Salt	Salt	Salt	Salt	
Foster Wheeler Development Corporation					
- Type	ST/SS[4]	ST/SS[5]	ST/SS	ST/SS	Only straight tube design; $6,290,000
- Shell fluid	Salt	Salt	Salt	Salt	
- Tube fluid	Water	Water/steam	Steam	Steam	
Babcock and Wilcox Company / Science Applications International Corporation					
- Type	UT/US[6]	UT/US	UT/US	UT/US	Only U-shell design; $4,300,000
- Shell fluid	Salt	Salt	Salt	Salt	
- Tube fluid	Water	Water/steam	Steam	Steam	

Notes:
1. Heat exchanger costs only
2. U-tube / straight shell
3. U-tube / straight shell with integral steam drum
4. Straight tube / straight shell
5. Steam drum integral with heat exchanger
6. U-tube/U-shell

Table 1-2
THERMAL STORAGE HOT SALT TANK DESIGNS AND COST ESTIMATES

Company	Pressure Boundary	Internal Liner	Internal Insulation	External Insulation	Capital Cost[1]
Chicago Bridge and Iron	Stainless steel	None	None	Mineral wool	$3,700,000
Pitt-Des Moines	Stainless steel	None	None	Mineral wool	$5,010,000
Technigaz	Carbon steel	Incoloy 800	Refractory bricks	Mineral wool	$10,370,000

Note 1. Installed cost, with insulation and foundation

CONCLUSIONS

Based on this study, the following conclusions can be drawn regarding the steam generator designs:

- The only steam generator concept which has demonstrated nitrate salt service at 566 C (1,050 F) is the 3 MWt U-tube/U-shell design developed by Babcock and Wilcox for the Molten Salt Electric Experiment at Sandia National Laboratories in Albuquerque, New Mexico (Ref. 1-3). However, ABB Lummus has fabricated kettle boiler steam generators for the 80 MWe Luz Solar Electric Generating Stations (SEGS) which approach the size and main steam pressure required in this study. In addition, each design is judged to be technically feasible as reflected by the conceptual design, cost estimate, and offer of a warranty on workmanship and materials provided by each vendor

- The designs proposed by B&W / SAIC and Foster Wheeler place the high pressure water or steam on the tube side of the heat exchangers. This minimizes the shell thicknesses, and in theory, should minimize the thermal inertia and the morning startup times. However, the morning startup time estimated by ABB Lummus for the kettle evaporator is the same as that estimated by Foster Wheeler and B&W / SAIC. In addition, the estimated startup times for 3 of the 4 steam generators are no longer, and may be shorter, than typical startup times for 100 MWe and larger reheat turbines

- Discussions with heat exchanger specialists at Bechtel, and a review of specifications for heat exchangers purchased by Bechtel during the past 7 years, indicate that the vendors have selected fluid paths (shell or tube side) and temperature changes such that the heat exchangers operate under typical commercial conditions. In particular, the maximum temperature difference between the inlet and outlet portions of the tubesheets in all of the designs does not exceed 110 C (200 F), and these conditions can be accommodated in commercial heat exchanger designs

- There is good agreement among the vendors regarding the costs of the heat exchangers; the divergence in the estimates occurs in the auxiliary equipment, engineering, and installation required for a complete system

- The steam generators evaluated for this study, including the U-tube/U-shell design, are considerably less expensive than the design developed for Phase I of the Utility Studies. This may be attributed to the successful application of relatively lower cost kettle boilers in the Luz SEGS plants, and renewed vendor interest in commercial central receiver projects following the start of the Solar Two Project

- It appears that a steam generator for a 100 MWe commercial project can be fabricated and installed for approximately $8 million.

All of the steam generator designs evaluated in this study should be suitable for a commercial central receiver project, and the potential exists for procurement on a competitive basis from a number of qualified vendors.

The following conclusions can be drawn regarding the hot salt storage tank designs:

- The only tank concept which has demonstrated nitrate salt service at 566 C (1,050 F) is the internally insulated design developed by Technigaz and Martin Marietta Corporation for the Subsystem Research Experiment at Sandia National Laboratories in Albuquerque, New Mexico (Ref. 1-3). However, CBI and PDM have fabricated tanks which approach the size and temperature required in this study, and each vendor is confident that a reliable design can be developed for a 100 MWe commercial project

- The internally insulated tank isolates the shell-to-floor joint from the temperature of the nitrate salt inventory, and therefore, the tank design should be highly tolerant of rapid temperature changes. However, the importance of this feature is mitigated by the established transient performance of conventional tank designs. A transient thermal model developed by Sandia National Laboratories predicts that an empty hot salt tank will cool overnight at a rate of 1 C (2 F) per hour. The following morning, as salt from the receiver is introduced into the tank at an average temperature of 454 C (850 F), the tank will initially cool at a rate of 55 C (100 F) per hour. During the next 30 minutes, the temperature of the salt from the receiver will increase to the normal outlet value of 566 C (1,050 F). Once this temperature is reached, the tank will heat at a rate of approximately 22 C (40 F) per hour. Discussions with CBI and PDM indicate that large tanks can routinely tolerate temperature ramp rates up to 56 C (100 F) per hour without suffering excessive creep or fatigue damage. Representative experience with large, externally insulated tanks which tolerate temperature transients at least as severe than those anticipated for a commercial solar project can also be found. For example, the thermal storage tanks for the SEGS I parabolic trough power plant are 21 m (70 ft) in diameter and routinely accommodate temperature change rates of 40 to 55 C (75 to 100 F) per hour. In addition, a nitrate salt tank 14 m (45 ft) in diameter fabricated by CBI for a proprietary chemical process plant in Texas normally operates at 260 C (500 F), but is periodically filled very quickly with salt at 450 C (842 F)

- During an extended shutdown, the hot tank will cool to 266 C (550 F), at which time electric energy is used to maintain the temperature of the inventory. Following the restart of the receiver, the tank may be subject to a rapid change in the temperature of the inventory. Depending on the results of a detailed thermal analysis, the tank and inventory may need to be preheated prior to the restart of the receiver to avoid excessive thermal stresses. If so, the electric energy for preheating should be included in the comparisons of the tank designs. However, the steady state thermal loss from the internally insulated tank is greater than the loss from an externally insulated design. Therefore, some annual quantity of heat tracing for the externally insulated tank can be used before the annual performance of the two designs is equal. A first order thermal analysis shows that the steady state thermal loss from the internally insulated tank is approximately 2.5 times the average of the thermal losses from the CBI and PDM designs. Assuming a Rankine cycle efficiency of 40 percent, the electric heat tracing on the externally insulated tanks could, in theory, be operated continuously and still offer the same annual thermal efficiency as the internally insulated design. Clearly, tank designs requiring such an operating strategy would not be proposed. However, it is apparent that the periodic use of trace heating on externally insulated tanks, should it be needed to limit transient thermal stresses, can be justified

- The Technigaz liner has demonstrated reliable service in numerous liquified natural gas tank installations. However, if a leak should develop in the liner of a nitrate salt tank, it is estimated that the repair procedure would be more lengthy than for an externally insulated tank. The time required to cool the inside of the tank, and in particular the thermal mass of the 512,000 refractory bricks, before repair personnel could enter would be considerably longer. In addition, the extent to which the refractory was contaminated with salt would need to be determined, and those bricks which had absorbed salt would need to be replaced. The estimated time to repair a leak in an externally insulated tank is 5 to 9 days, while the time for an internally insulated tank is estimated to be 15 to 30 days. Thus, the frequency of leaks in an internally insulated tank can be only one-half to one-third of that in an externally insulated design without suffering a disadvantage in annual availability

- There is good agreement on the cost estimates from the two vendors offering externally insulated designs and who are potential suppliers to the Solar Two and early commercial projects

- It appears that an externally insulated hot salt tank for a 100 MWe commercial project can be fabricated and installed for approximately $5 million. An internally insulated design is projected to be approximately twice as expensive.

Both the internally and externally insulated designs are judged to be acceptable for commercial service, and the potential exists for procurement on a competitive basis from a number of qualified vendors.

FUTURE ACTIVITIES

This study leaves unresolved the selection of the preferred hot salt tank and steam generator designs for the first commercial project. In particular, a definitive selection cannot be made without firm cost estimates, and it is believed that these estimates can only be obtained as part of the procurement process prior to plant construction.

A possible approach to the selection of an optimum storage tank during project procurement is outlined below. A final set of procedures will be developed as part of the Solar Two Project, and these may also form the basis for procurement activities in the first commercial project. The first steps would involve calculations by the plant engineer of the following:

- Temperature and flow rate of the salt from the receiver over the course of a year

- Minimum salt temperature to the hot tank during morning startup and following cloud transients

- Inventory required to operate the auxiliary steam generator during the daily turbine startup.

From these calculations, the temperature to, and the flow rate to and from, the hot tank over the course of a year can be determined. This information would be included in the bid package to the tank vendors.

The vendors would conduct analyses of transient thermal stresses and fatigue damage, and then develop the tank designs, operating requirements, thermal losses, leak repair times, and bid prices. The vendors will be free to select their optimum combination of features. For example, an inexpensive shell-to-floor joint with a thick salt heel may be a lower cost solution to transient stresses than a more sophisticated curved joint with a thin heel. Similarly, the vendor would specify any constraints on tank operation. For example, if the inventory temperature must be maintained at 480 C (900 F) during an extended shutdown to prevent excessive thermal stresses following the plant startup, this information would be included in the bids to the project.

The engineer would evaluate all of the bids received, and develop total annual capital and operating costs (including possible repairs) for each vendor. From this, definitive comparisons with competing designs could be made and the optimum design selected.

A similar set of procedures would be required to assess the competing steam generator designs. In particular, overnight thermal conditioning requirements would be a principal consideration in the analysis. However, more detailed operating procedures, including limits placed on morning startup rates by the turbine, would need to be developed before formal requests for proposals could be prepared.

Section 2
Background and Introduction

In 1986, two utilities, Arizona Public Service Company (APS) and Pacific Gas and Electric Company (PG&E), entered into a cooperative agreement with the Department of Energy to define the first generation of commercial central receiver power plants. The study, entitled "Solar Central Receiver Technology Advancement for Electric Utility Applications" (and called the "Utility Studies" for convenience), had two phases. The goal of Phase I was to develop a consensus on the preferred near term commercial plant design, and the goal of Phase II was to establish a development plan to commercialize the technology. The study selected a 100 MWe plant with a cylindrical nitrate salt receiver and a surround heliostat field as the preferred commercial design, and recommended the conversion of the 10 MWe Solar One pilot plant to nitrate salt technology as the lowest cost approach to commercialize the technology.

UTILITY STUDIES PHASE I STEAM GENERATOR AND THERMAL STORAGE TANK DESIGNS

The principal objective of Phase I during the Utility Studies was to select the preferred receiver coolant. To this end, the design of various equipment items was fixed early in the study so as not to divert attention from the receiver optimization studies These items included the following:

Steam Generator The design and cost of the steam generator were based on a study for a 100 MWe plant prepared by Babcock and Wilcox for Sandia National Laboratories in 1982 (Ref. 2-1), and a 3 MWt unit fabricated by Babcock and Wilcox for Sandia National Laboratories and tested at the Molten Salt Electric Experiment in 1985 (Ref. 1-3). The design used U-tube/U-shell heat exchangers and a forced recirculation evaporator. The high pressure fluid (water/steam) was placed on the tube side in each heat exchanger, and the low pressure fluid (nitrate salt) on the shell side. The installed cost, in third quarter 1987 dollars, was estimated to be $11.1 million.

The U-shell concept allowed the inlet channel tubesheet to be separated from the outlet tubesheet, and thereby avoided the exposure of a single tubesheet to the large temperature gradients inherent in the superheater and reheater. Similar reasoning in a second steam generator study for Sandia by Foster Wheeler Solar Development Corporation in 1982 led to the selection of straight tube/straight shell heat exchangers (Ref. 2-2). Differential thermal expansion between the tubes and shell was accommodated by a bellows surrounding the inlet water/steam piping.

Thermal Storage Tanks The design and cost estimate were developed by Chicago Bridge and Iron Technical Services Company (CBI) and assumed a "conventional" approach using vertical, atmospheric pressure tanks with external calcium silicate insulation. The cold and hot tanks were fabricated from carbon steel and stainless steel, respectively, and cooling air passages were located in the foundation to prevent native soil temperatures from exceeding 100 C (212 F). The installed costs of the cold and hot tank, in third quarter 1987 dollars, were estimated to be $1.0 million and $3.0 million, respectively. A brief parallel study by Pitt-Des Moines, Inc. (PDM) resulted in tank designs similar to the CBI concept.

At the time of the study, CBI and PDM had designed and fabricated tanks to requirements which were similar, but not identical, to those for nitrate salt at 566 C (1,050 F). The only design which has been proven for this service is one developed by S. N. Technigaz (a French company) and Martin Marietta Corporation. A 7 MWht thermal storage system was installed at the Central Receiver Test Facility at Sandia National Laboratories in Albuquerque, New Mexico (Ref. 2-3). The cold salt tank used a carbon steel shell with external insulation, while the hot tank a carbon steel shell with internal and external insulation. To limit the shell temperature on the hot tank to acceptable values, a layer of refractory brick was installed inside the shell. A thin, corrugated Incoloy sheet lined the inside of the bricks to protect the refractory from the corrosive effects of the nitrate salt. The design was more complex than a stainless steel tank, but it offered the advantages of a low cost pressure boundary and the ability to accept rapid temperature changes. Cost analyses by Sandia National Laboratories using information developed by Martin Marietta Corporation showed the tank to be competitive with the designs with external insulation (Ref. 2-4).

UTILITY STUDIES PHASE II STEAM GENERATOR AND THERMAL STORAGE TANK DESIGNS

During Phase II, a conceptual design and cost estimate were developed for the conversion of Solar One to nitrate salt technology. Potential heat exchanger and tank vendors were contacted for conceptual designs and budgetary estimates of a 35 MWt nitrate salt steam generator and an 80 MWht thermal storage system, respectively.

Two of the three heat exchanger vendors recommended a kettle boiler concept, in which saturated steam is generated in a pool on the shell side of the evaporator. This approach was selected based in part on the successful operation of similar equipment at the Luz Solar Electric Generating Station parabolic trough solar power plants, and on the potential for a lower capital cost. The third vendor, Babcock and Wilcox, recommended the U-tube/U-shell design. Both of the tank vendors recommended externally insulated tanks, with the cold salt tank fabricated from carbon steel and the hot tank from stainless steel. However, an alternate hot salt tank design using a carbon steel shell and internal insulation was not evaluated. Thus, there were several differences between the equipment designs for the first commercial plant and the 10 MWe plant that was to precede it, and the conceptual nature of the Phase II study could not resolve these issues.

INVESTIGATION OF THERMAL STORAGE AND STEAM GENERATOR ISSUES

Late in 1991, Southern California Edison Company organized a group of utilities and government organizations, and submitted a proposal to DOE to convert the Solar One plant to nitrate salt technology (Solar Two). The purpose of the project was to reduce the perceived risk in building the first commercial 100 MWe plant. To this end, Solar Two needed to duplicate the technical features of the first commercial plant as closely as possible.

The question arose: What should be the steam generator and thermal storage tank designs in the first commercial plant? This study sought a partial answer by reviewing potential equipment designs and identifying those which would be suitable. The approach involved the following steps:

- Subcontracts were placed with four heat exchanger vendors and three tank vendors to examine the full range of options

- The vendors developed conceptual designs and cost estimates for the equipment required in the first 100 MWe commercial plant

- Bechtel reviewed the vendor information, assessed the technical feasibility of each design, and determined whether the equipment would be suitable for the commercial plant.

It should be noted that the purpose of the study was limited to an assessment of the feasibility of each design, and not a selection of the best equipment concept.

Section 3 of this report reviews the steam generator designs, Section 4 reviews the thermal storage tank designs, and Section 5 lists the references. Appendices A and B are statements of work for the steam generator and thermal storage tank vendors, respectively.

Section 3
Steam Generator Designs and Cost Estimates

Four conceptual steam generator designs and cost estimates were developed during this study. Two of the designs, one developed by ABB Lummus Heat Transfer (ABB Lummus) and a second by Struthers Wells Corporation (Struthers Wells), employed U-tube/straight shell heat exchangers with a kettle steam generator. The third design, prepared by Foster Wheeler Development Corporation (Foster Wheeler), employed straight tube/straight shell heat exchangers with bellows to accommodate differential thermal expansion between the shell and tubes. The fourth design, presented by Science Applications International Corporation (SAIC) teamed with the Babcock and Wilcox Company (B&W), used U-tube/U-shell heat exchangers and a separate steam drum.

The discussion which follows reviews the steam generator specification, design features, warranty provisions, and cost estimate for each of the concepts.

SPECIFICATION

The principal specification used in the design of all of the steam generators is presented in Table 3-1. The steam generator is intended for a nominal 100 MWe commercial plant using a reheat turbine cycle. It is sized to produce 92.77 kg/sec (736,300 lb/hr) of main steam at a pressure of 13.03 MPa (1,890 psia) and a temperature of 540 C (1,004 F) from a feedwater flow of 93.71 kg/sec (743,700 lb/hr) at a temperature of 236 C (456 F). The steam generator must also reheat 79.92 kg/sec (634,300 lb/hr) of intermediate pressure steam from the turbine, raising its temperature from 347 C (656 F) to 538 C (1,000 F).

A value of \$2,300/m (\$700/ft) of pressure drop on the salt side of the heat exchangers was assigned to assist the vendors in selecting the optimum heat exchange area and tube configuration.

DESIGN FEATURES

The principal features of the four heat exchanger designs are summarized in Table 3-2. Shown are the tube and shell configurations, materials, duties, fluid temperatures, log mean temperature differences, overall heat transfer coefficients, and net heat exchange areas.

Although each vendor worked to the same specification, there are many differences in the heat exchanger details. These can be attributed to the following:

- Preferred approach to accommodating thermal expansion. For example, Foster Wheeler uses straight tube/straight shell heat exchangers with floating tubesheets, while ABB Lummus, B&W / SAIC, and Struthers Wells each use U-tubes with fixed tubesheets.

- Different approaches to circulation in the evaporator. Foster Wheeler and B&W / SAIC use natural and forced recirculation, respectively, while ABB Lummus and Struthers Wells use kettle boilers

Table 3-1
STEAM GENERATOR PERFORMANCE SPECIFICATION

Nominal Ratings	110 MWe gross plant output 260 MWt steam generator duty
Final Feedwater	236 C (456 F) (As required) MPa (psia) 93.71 kg/sec (743,700 lb/hr); 1% blowdown assumed
Main Steam	540 C (1,004 F) 13.03 MPa (1,890 psia) 92.77 kg/sec (736,300 lb/hr)
Cold Reheat Steam	347 C (656 F) 3.08 MPa (446 psia) 79.92 kg/sec (634,300 lb/hr)
Hot Reheat Steam	538 C (1,000 F) 2.77 MPa (402 psia) 79.92 kg/sec (634,300 lb/hr)
Nitrate Salt	566 C (1,050 F) inlet temperature (As required) MPa (psia) inlet pressure 454 C (850 F) maximum evaporator tube temperature consistent with acceptable corrosion rates for chrome-moly tubes 288 C (550 F) outlet temperature 138 kPa (20 psia) outlet pressure Specific heat $0.345 + (2.28 \times 10^{-5})(\text{Temp, F})$, $\text{Btu/lb}_m\text{-F}$ Density $131.2 - (2.221 \times 10^{-2})(\text{Temp, F})$, lb_m/ft^3 Thermal conductivity $0.25308 + (6.26984 \times 10^{-5})(\text{Temp, F})$, Btu/hr-ft-F Viscosity $60.2844 - (0.17236)(\text{Temp, F}) + (1.76176 \times 10^{-4})(\text{Temp, F})^2 - (6.11408 \times 10^{-8})(\text{Temp, F})^3$, $\text{lb}_m/\text{ft-hr}$

Table 3-2
COMPARISON OF STEAM GENERATOR TECHNICAL CHARACTERISTICS - PREHEATER

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
- Number of shells	1	1	1	1
- Type	U tube, straight shell	U tube, straight shell	Straight tube, straight shell	U tube, U shell
- Passes				
Shell	2	2	1	1
Tube	2	2	1	1
- Fluids				
Shell	Nitrate salt	Water	Nitrate salt	Nitrate salt
Tube	Water	Nitrate salt	Water	Water
- Materials				
Shell	Carbon steel	Carbon steel	Carbon steel	Carbon steel
Channel	"	"	"	"
Tubesheet	"	"	"	"
Tube	"	"	"	"
- Duty				
MWt	48.21	45.65	49.60	48.00
million Btu/hr	164.55	155.8	169.28	163.82
- Inlet temperatures, F				
Nitrate salt	658	640	646	642
Water	456	456	456	480
- Outlet temperatures, F				
Nitrate salt	550	550	550	550
Water	629	620	631	630
Log mean temperature difference, F	54.6	47.8	43.0	32.9
Fouling factor, hr-ft2-F/Btu				
Shell	0.0010	0.0005	0.0005	Not specified
Tube	0.0005	0.0005	0.0015	"
Overall heat transfer coefficient, Btu/hr-ft2-F	206.8	231.0	Not provided	338
Effective surface area, ft2	14,458	14,203	21,090 (1)	22,060 (2)

1) Represents an 18.5 percent margin on heat transfer area, including a 3 percent allowance for tube plugging
2) Includes 25 percent margin on heat transfer area

Table 3-2 (Continued)

COMPARISON OF STEAM GENERATOR TECHNICAL CHARACTERISTICS - EVAPORATOR

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
- Number of shells	1	1	1	1
- Type	U tube, straight shell	U tube, straight shell	Straight tube, straight shell	U tube, U shell
- Passes				
Shell	1	1	1	1
Tube	2	2	1	1
- Fluids				
Shell	Water/steam	Water/steam	Nitrate salt	Nitrate salt
Tube	Nitrate salt	Nitrate salt	Water/steam	Water/steam
- Materials				
Shell	Carbon steel	Carbon steel	1 1/4 Cr - 1/2 Mo	2 1/4 Cr - 1 Mo
Channel	Carbon steel (1)	1 1/4 Cr - 1/2 Mo	Cr-Mo and CS	"
Tubesheet	1 Cr - 1/2 Mo (1)	"	"	"
Tube	1 Cr - 1/2 Mo (2)	"	1 1/4 Cr - 1/2 Mo	"
- Duty				
MWt	104.7	107.5	103.5	102.4
million Btu/hr	357.4	367.0	353.2	349.5
- Inlet temperatures, F				
Nitrate salt	890	848	845	836
Saturated water	629	620	631	633
- Outlet temperatures, F				
Nitrate salt	658	640	646	642
Saturated steam	629	629	631	638
Log mean temperature difference, F	105.6	69.0	74.9	60.2
Fouling factor, hr-ft2-F/Btu				
Shell	0.0005	0.0005	0.0005	Not specified
Tube	0.0010	0.00067	0.0030	"
Overall heat transfer coefficient, Btu/hr-ft2-F	200.6	215.0	Not provided	283
Effective surface area, ft2	16,680	24,630	19,950 (3)	23,160 (4)

1) Stainless steel cladding on inlet pass
2) Stainless steel inserts in inlet tube pass
3) Represents a 21.2 percent margin on heat transfer area, including a 3 percent allowance for tube plugging
4) Includes 25 percent margin on heat transfer area

Table 3-2 (Continued)
COMPARISON OF STEAM GENERATOR TECHNICAL CHARACTERISTICS - SUPERHEATER

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
- Number of shells	2	1	1	1
- Type	U tube, straight shell	U tube, straight shell	Straight tube, straight shell	U tube, U shell
- Passes				
Shell	2	2	1	1
Tube	2	2	1	1
- Fluids				
Shell	Nitrate salt	Superheated steam	Nitrate salt	Nitrate salt
Tube	Superheated steam	Nitrate salt	Superheated steam	Superheated steam
- Materials				
Shell	1 Cr - 1/2 Mo steel (1)	316 stainless steel	304 stainless steel	304 stainless steel
Channel	1 Cr - 1/2 Mo steel	"	"	"
Tubesheet	1 Cr - 1/2 Mo steel (1)	"	"	"
Tube	316 stainless steel	Inconel Alloy 800	"	"
- Duty				
MWt	73.01	71.73	72.6	74.2
million Btu/hr	249.2	244.8	247.8	253.2
- Inlet temperatures, F				
Nitrate salt	1,050	1,050	1,050	1,040
Saturated steam	629	629	631	638
- Outlet temperatures, F				
Nitrate salt	890	848	808	836
Superheated steam	1,004	1,004	1,004	1,005
Log mean temperature difference, F	123.6	110.9	60.1	94.1
Fouling factor, hr-ft2-F/Btu				
Shell	0.0010	0.0005	0.0005	Not specified
Tube	0.0005	0.0005	0.0015	"
Overall heat transfer coefficient, Btu/hr-ft2-F	160.9	139.6	Not provided	417
Effective surface area, ft2	12,510	16,745	8,900 (2)	6,090 (3)

1) With stainless steel cladding
2) Represents a 13.8 percent margin on heat transfer area, including a 3 percent allowance for tube plugging
3) Includes 25 percent margin on heat transfer area

Table 3-2 (Continued)
COMPARISON OF STEAM GENERATOR TECHNICAL CHARACTERISTICS - REHEATER

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
- Number of shells	1	1	1	1
- Type	U tube, straight shell	U tube, straight shell	Straight tube, straight shell	U tube, U shell
- Passes				
Shell	2	2	1	1
Tube	2	2	1	1
- Fluids				
Shell	Superheated steam	Superheated steam	Nitrate salt	Nitrate salt
Tube	Nitrate salt	Nitrate salt	Superheated steam	Superheated steam
- Materials				
Shell	1 Cr - 1/2 Mo steel	316 stainless steel	304 stainless steel	304 stainless steel
Channel	1 Cr - 1/2 Mo steel (1)	"	"	"
Tubesheet	"	"	"	"
Tube	316 stainless steel	Inconel Alloy 800	"	"
- Duty				
MWt	34.72	35.86	34.5	34.7
million Btu/hr	118.5	122.4	117.7	118.4
- Inlet temperatures, F				
Nitrate salt	1,050	1,050	1,050	1,040
Saturated Steam	656	656	656	656
- Outlet temperatures, F				
Nitrate salt	850	848	898	836
Superheated Steam	1,000	1,000	1,000	1,000
Log mean temperature difference, F	101.5	105.6	121.8	93.1
Fouling factor, hr-ft2-F/Btu				
Shell	0.0005	0.0005	0.0005	Not specified
Tube	0.0010	0.0005	0.0015	"
Overall heat transfer coefficient, Btu/hr-ft2-F	107.7	72.54	Not provided	188
Effective surface area, ft2	10,830	16,880	6,200 (2)	8,480 (3)

1) Stainless steel cladding on inlet pass
2) Represents a 13.1 percent margin on heat transfer area, including a 3 percent allowance for tube plugging
3) Includes 25 percent margin on heat transfer area

- Different unit costs for materials and labor, which lead to differences in the optimum heat exchange area

- Different assumptions regarding fouling factors and design margins, which influence the required heat exchange area. For example, B&W / SAIC adds a margin of 25 percent to the calculated areas, while Foster Wheeler uses a margin of 13 to 21 percent including 3 percent for tube plugging.

A discussion of the similarities and differences among the designs is presented below.

ABB Lummus Kettle Boiler Steam Generator

The ABB Lummus design includes a U-tube kettle boiler and U-tube/straight shell heat exchangers for the preheater, superheater, and reheater. A flow schematic is presented in Figure 3-1. The approach uses conventional heat exchanger designs, and draws on the experience gained in fabricating the steam generators for the Luz Solar Electric Generating Stations.

It should be noted that while much of the Luz experience is applicable, the two steam generators are designed for different conditions. The Luz equipment used synthetic oil for the heat transport fluid and generated main steam at 10.0 MPa (1,450 psia) and 371 C (700 F); main steam conditions for the nitrate salt steam generator are 13.03 MPa (1,890 psia) and 540 C (1,005 F). In addition, the temperature range of the oil was only 100 C (180 F) while the range for the nitrate salt is 278 C (500 F). This larger range placed constraints on the selection of tube and shell fluids in the nitrate salt steam generator, and resulted in the use of two superheater shells in series and placing the steam flow on the shell side of the reheater.

<u>Superheater and Reheater Arrangements</u> Selecting a design with two superheaters in series offers two benefits. First, the steam temperature increases 208 C (375 F) as it progresses from the evaporator outlet to the superheater outlet. The use of two superheaters allows this increase to occur in two steps, and limits the temperature difference between the inlet and outlet portions of the tubesheet to 104 C (188 F). This is a moderate gradient and is routinely used in commercial heat exchangers. The limited gradient allows the high pressure steam to be placed on the tube side of the heat exchanger, which reduces the thickness and cost of the shell. Second, salt attemperation for main steam temperature control occurs at a lower temperature than if the cold salt were mixed at the inlet to the superheaters. Although this effect is minor, the thermodynamic efficiency is higher than if attemperation was done at the inlet to the superheater.

The large temperature change of the reheat steam (191 C (344 F)) places the same constraints on the reheater design. However, ABB Lummus elected to place the high pressure steam on the shell side and limit the number of reheater shells to one. The reheater duty was approximately one-half the superheater duty, and the lowest cost approach may have been to fabricate one heat exchanger with a high pressure shell rather than two small heat exchangers with low pressure shells. This is the same approach as adopted in the Struthers Wells reheater design.

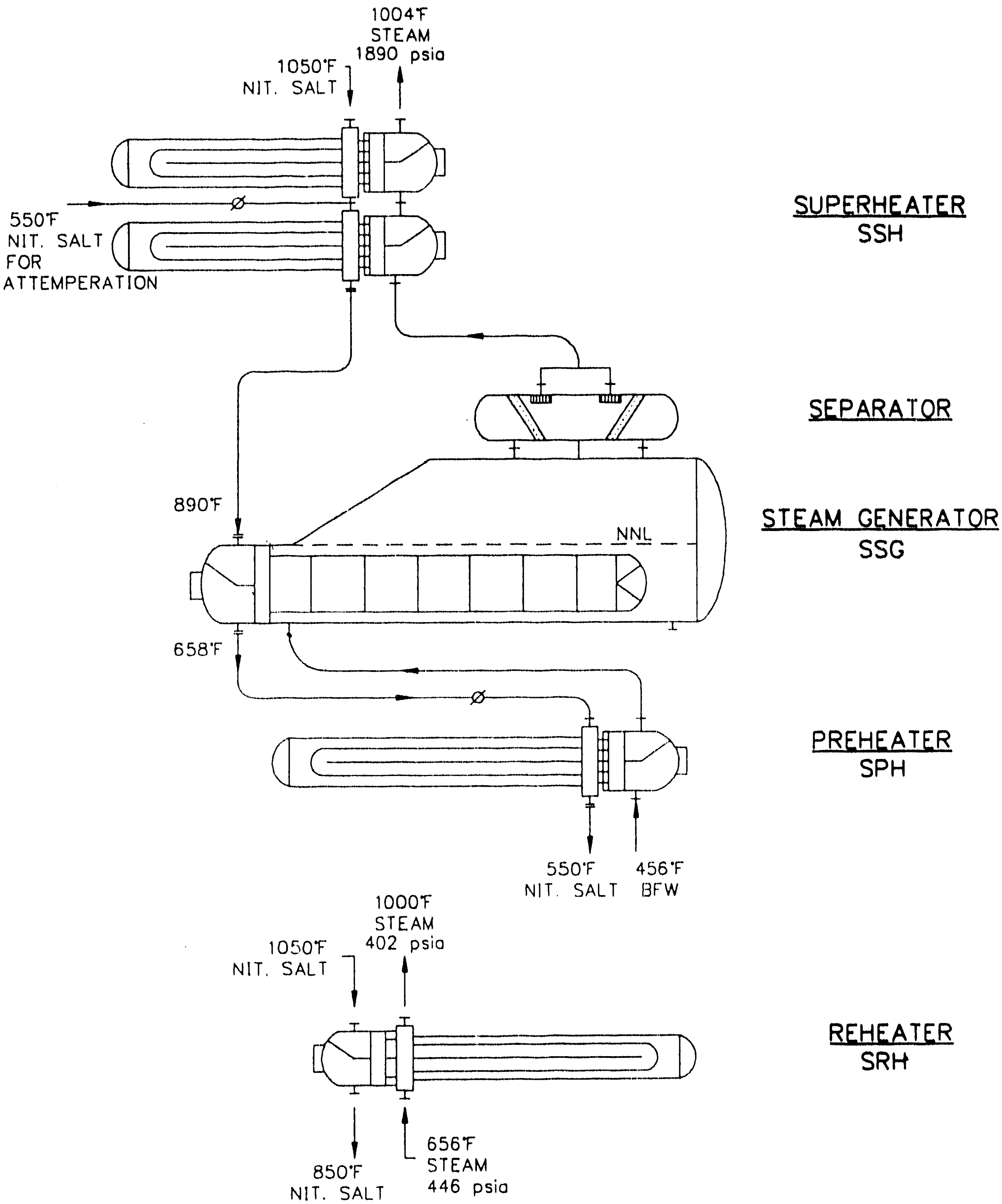

Figure 3-1 ABB Lummus Steam Generator Flow Schematic

Evaporator Inlet Flow One unusual feature of the ABB Lummus design centers on the salt flow to the evaporator. Note in the flow schematic that the salt flows through the two superheaters and into the evaporator, but that the flow from the reheater returns to the storage system at a temperature of 454 C (850 F). Apparently, the engineers at ABB Lummus were not aware of the "typical" steam generator configuration, in which the flows through the superheater and reheater are combined and directed to the evaporator. However, the "typical" arrangement was not made clear in the specification prepared by Bechtel, and ABB Lummus should not be criticized for selecting a design that is not directly comparable to those from the other vendors.

The misunderstanding is not without benefit. By separating the superheater and reheater outlet flows, the salt inlet temperature to the evaporator is raised 22 C (40 F) above the allowable value of 454 C (850 F). To prevent excessive corrosion of the 1 Cr - ½ Mo channel and tube materials, ABB Lummus proposed that the inlet channel and the high temperature portion of the tubes be clad with stainless steel. Sandia National Laboratories is currently conducting a survey of ferritic material corrosion rates, and is considering disassembly and examination of the 2¼ Cr - 1 Mo tube and shell materials used in the evaporator of the Molten Salt Electric Experiment steam generator. If it is determined that ferritic materials with chromium contents of 1 to 2¼ percent are not compatible with nitrate salt at temperatures up to 454 C (850 F), stainless steel cladding of the high temperature portions of the evaporator could be considered as an option to ferritic materials with a high chromium content, such as 9 Cr - 1 Mo.

Struthers Wells Kettle Boiler Steam Generator

The Struthers Wells design is very similar to the ABB Lummus concept. It includes a U-tube kettle boiler and U-tube/straight shell heat exchangers for the preheater, superheater, and reheater. A flow schematic is presented in Figure 3-2.

The Struthers Wells design differs from the ABB Lummus concept in two areas. First, the water/steam separators in the evaporator are placed inside the kettle boiler rather than outside. This eliminates the need for a separate vessel, but increases the kettle diameter and wall thickness by approximately 12 percent. Second, only one superheater shell is used. This reduces the number of heat exchangers, but requires the high pressure steam to be placed on the shell side of the heat exchangers. As discussed above, a temperature change of 110 C (200 F) between the inlet and outlet channels is common in commercial heat exchangers. However, a change of 200 C (360 F) would not be typical. Since the nitrate salt and steam temperature changes in the heat exchanger are 112 C (202 F) and 208 C (375 F), respectively, the steam was placed on the shell side.

Foster Wheeler Straight Tube Steam Generator

The Foster Wheeler design includes straight tube/straight shell heat exchangers for the preheater, evaporator, superheater, and reheater. The evaporator design is unique in that it incorporates the steam drum in the steam outlet channel. Bellows surrounding the inlet water or steam piping accommodate differential thermal expansion between the tubes and shell. A flow schematic is presented in Figure 3-3.

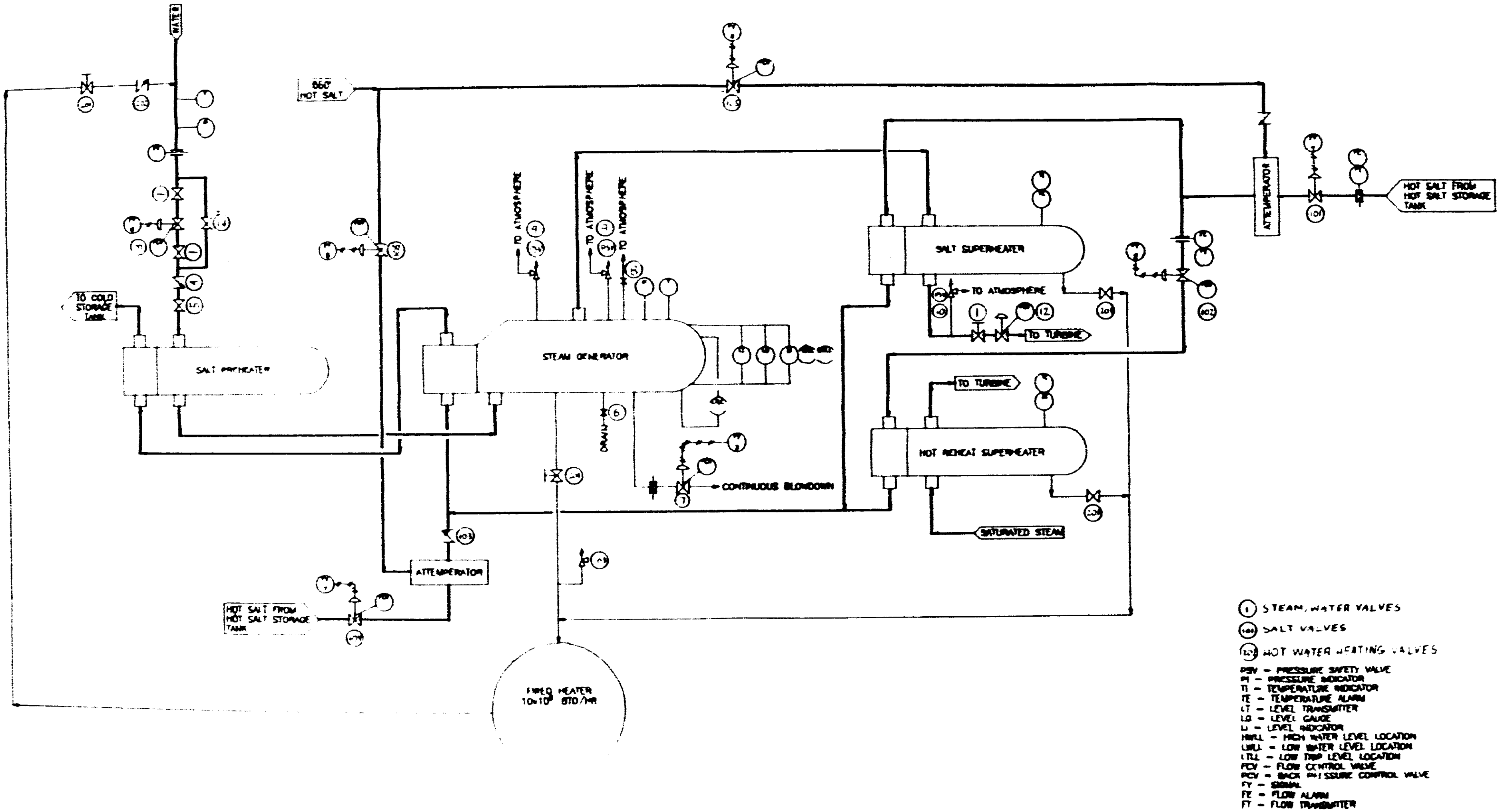

Figure 3-2 Struthers Wells Steam Generator Flow Schematic

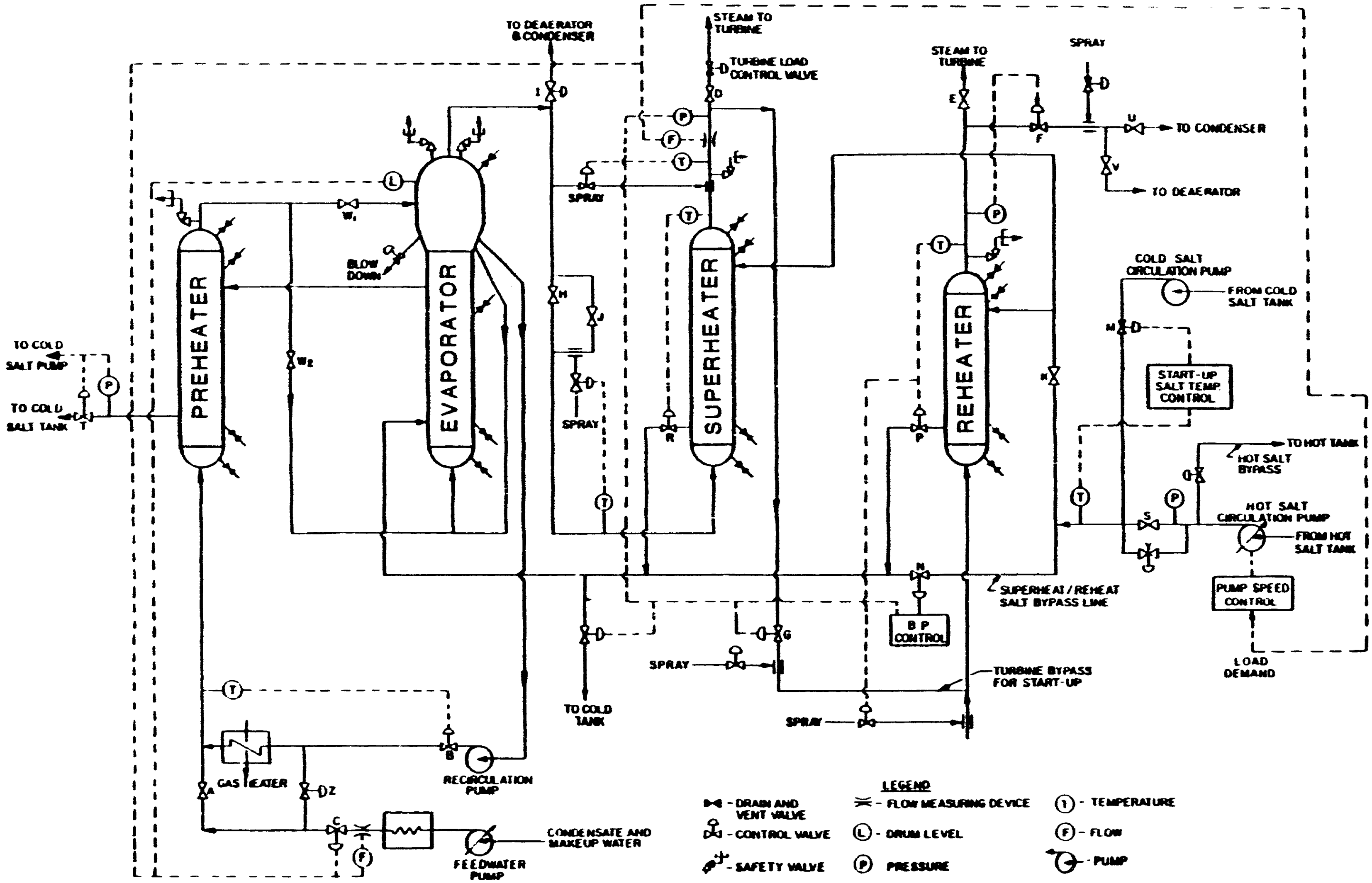

Figure 3-3 Foster Wheeler Steam Generator Flow Schematic

The design is a moderate revision to an essentially identical 100 MWe steam generator study prepared for Sandia National Laboratories in 1982 (Ref. 2-1). The 1982 report described in detail the selection rationale for the straight tube/straight shell heat exchangers, thermal analysis, hydraulic performance, mechanical design, boiling stability analysis, and operating procedures. The principal features included the following:

- To permit the use of bellows, all nitrate salt flows are on the shell side. The bellows are located outside the inlet water or steam piping, rather than in the shell, to limit the bellow sizes

- The large differential thermal expansion due to steam temperature changes in the superheater and reheater are readily accommodated by the separate inlet and outlet tubesheets

- A natural circulation evaporator was selected over forced recirculation, once-through, and Sulzer types. To promote the required circulation, the evaporator is arranged vertically, and to reduce pressure losses and costs, the steam drum is located in the outlet channel of the evaporator. The preheater, superheater, and reheater are also arranged vertically to simplify the support structure

- The preheater, superheater, and reheater use a counter flow arrangement. The evaporator uses parallel flow to improve natural circulation.

The current design was adapted from the 1982 study by adjusting heat exchanger tube lengths to account for slightly lower thermal ratings. The adjustments ranged from -3 percent for the preheater to -15 percent for the reheater.

B&W / SAIC U-Tube/U-Shell Steam Generator

The B&W / SAIC design includes U-tube/U-shell heat exchangers for the preheater, evaporator, superheater and reheater, and an elevated steam drum between the evaporator and superheater. A flow schematic is presented in Figure 3-4 (Salt Side) and Figure 3-5 (Water/Steam Side). The design, which evolved from a parallel study to that conducted by Foster Wheeler in 1982 for Sandia National Laboratories (Ref. 2-2), has several features which are different from the other vendors. These include the following:

- Separate inlet and outlet tubesheets reduce the constraints on temperature change in one heat exchanger; thus, the high pressure water/steam flows can be placed on the tube side and shell thicknesses held to a minimum

- The U-shaped tubes accommodate differential thermal expansion between the tubes and shell without the need for the floating tubesheets or bellows normally required with separate inlet and outlet tubesheets

- The heat exchangers, including the evaporator, are arranged horizontally; thus, boiling occurs in horizontal tubes

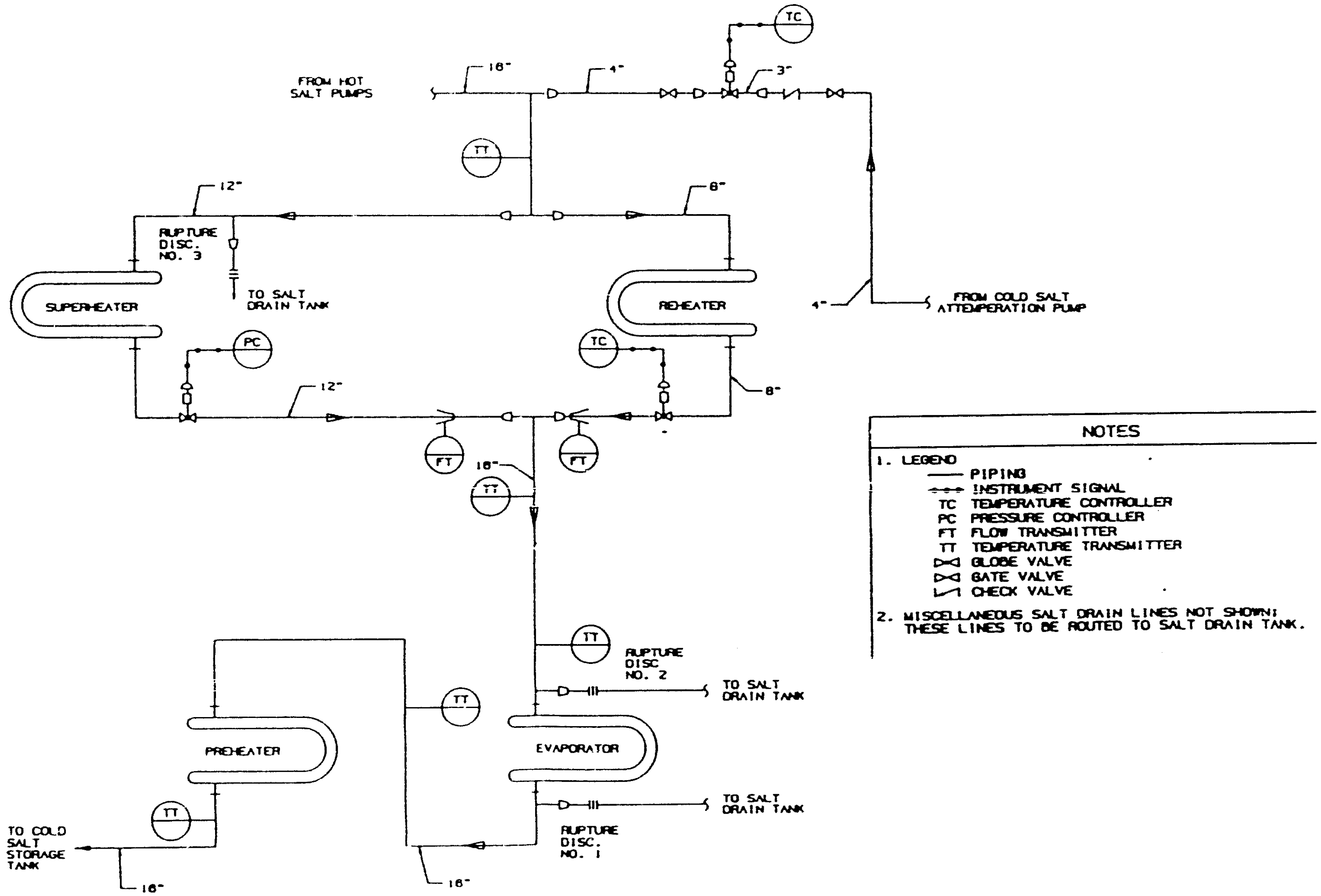

Figure 3-4 B&W / SAIC Steam Generator Flow Schematic (Salt Side)

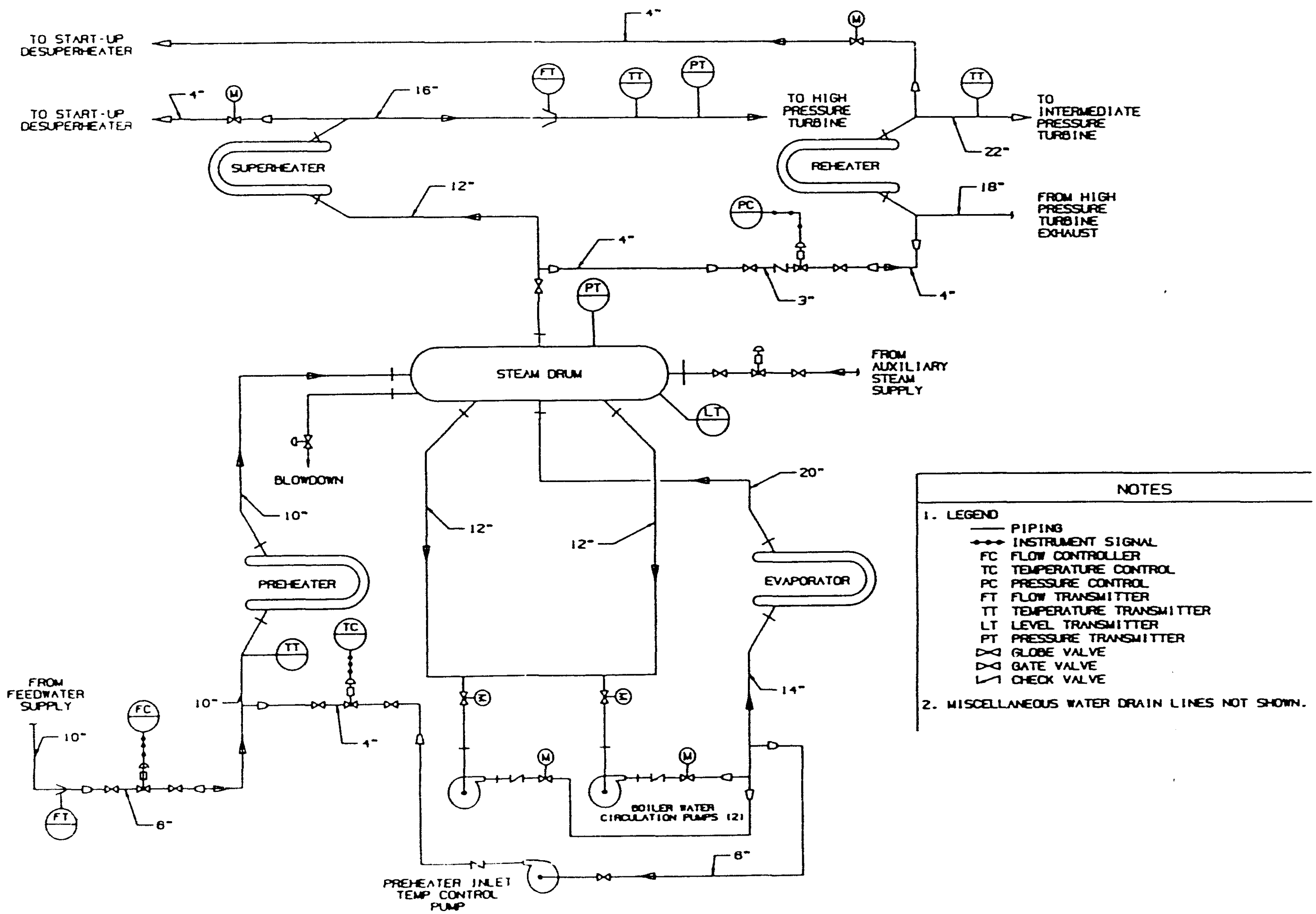

Figure 3-5 B&W / SAIC Steam Generator Flow Schematic (Water/Steam Side)

- The last 9.8 m (32 ft) of the evaporator tube use internal spiral ribs to prevent departure from nucleate boiling in the tube sections with high quality steam

- The evaporator uses recirculation pumps to maintain adequate water/steam flow rates in the tubes

- An elevated steam drum provides saturated water to the recirculation pumps at the required suction head, and dries the saturated steam flowing to the superheater.

It can be noted that the only nitrate salt steam generator built to date for solar applications was the Babcock and Wilcox 3 MWt U-tube/U-shell design installed in the Molten Salt Electric Experiment at the Central Receiver Test Facility in Albuquerque, New Mexico.

VESSEL SHELL THICKNESSES AND WEIGHTS

A comparison of the steam generator vessel shell thicknesses and weights is shown in Table 3-3.

Table 3-3
COMPARISON OF STEAM GENERATOR VESSEL SHELL THICKNESSES[1] AND WEIGHTS[2]

	ABB Lummus		Struthers Wells		Foster Wheeler[3]		B&W / SAIC	
Preheater								
- Thickness	13	(0.50)	111	(4.375)	25	(1.0)	13	(0.5)
- Weight	43,000	(95,000)	52,600	(115,900)	54,000	(119,000)	35,700	(78,800)
Evaporator								
- Thickness	156	(6.125)	194	(7.625)	25	(1.0)	13	(0.5)
- Weight	170,000	(375,000)	218,000	(480,600)	122,000	(269,000)	78,400	(172,800)
Steam drum								
- Thickness	156	(6.125)	194	(7.625)	171	(6.75)	95	(3.75)
- Weight	With evaporator		With evaporator		With evaporator		41,100	(90,500)
Superheater								
- Thickness	37	(1.4375)	140	(5.5)	19	(0.75)	10	(0.375)
- Weight	97,000[4]	(214,000)	65,300	(144,000)	28,500	(62,800)	12,100	(26,700)
Reheater								
- Thickness	60	(2.375)	38	(1.5)	19	(0.75)	16	(0.625)
- Weight	57,000	(126,000)	62,200	(137,100)	20,900	(46,000)	12,700	(28,100)
Total	464,000	(1,024,000)	398,100	(877,600)	225,400	(496,800)	180,000	(396,900)

Notes:
1) mm (in.)
2) Dry weight, without insulation; kg (lb)
3) Approximate; weights shown are those in Ref. 2-1
4) Weight for 1 of 2 superheaters

The ABB Lummus, Foster Wheeler, and B&W / SAIC preheater designs place the high pressure water on the tube side, while the Struthers Wells approach places the low pressure nitrate salt on the tube side. The theoretical weight advantage is realized in the ABB Lummus and B&W / SAIC designs, but it is not apparent in the Foster Wheeler approach.

The ABB Lummus and Struthers Wells kettle evaporators place the high pressure water-steam mixture on the shell side, while the Foster Wheeler and B&W / SAIC designs place the nitrate salt on the shell side. As expected, the kettle evaporators are considerably heavier than the designs in which boiling occurs in the tubes.

The ABB Lummus, Foster Wheeler, and B&W / SAIC superheater designs place the high pressure steam on the tube side, while the Struthers Wells approach places the low pressure nitrate salt on the tube side. The theoretical weight advantages are realized in the Foster Wheeler and B&W / SAIC designs, but the total weight of the 2 ABB Lummus heat exchangers is three times that of the Struthers Wells superheater. Some of this difference may be attributed to the large tubesheets in the ABB Lummus design; they are 2.1 m (82 in.) in diameter and 660 mm (26 in.) thick.

In a manner similar to the evaporators, the ABB Lummus and Struthers Wells reheaters place the high pressure steam on the shell side, while the Foster Wheeler and B&W / SAIC designs place the nitrate salt on the shell side. As with the evaporators, the ABB Lummus and Struthers Wells heat exchangers are considerably heavier than the other two designs.

As shown by the column totals in the table, the weight of the kettle boiler designs is approximately twice that of the Foster Wheeler and B&W / SAIC designs.

STARTUP TIMES

Startup times from cold and warm conditions for each of the steam generators are summarized in Table 3-4.

Table 3-4
STARTUP TIMES FROM COLD AND WARM CONDITIONS[1]

Vendor	From Cold	From Warm
ABB Lummus	Not specified	0.5 to 1
Struthers Wells	10	Less than 5[2]
Foster Wheeler	10	1
B&W / SAIC	4	Less than 1

Notes:
1. Time, in hours, to normal operating conditions; a cold startup is from ambient temperature; a warm startup follows an overnight shutdown
2. Conservative value in lieu of transient thermal analysis; shorter times are likely

Estimates of the times required to heat the steam generator from ambient temperature to normal operating conditions ranged from 4 to 10 hours. These times are of interest to the plant operators, but have a limited influence on the feasibility assessment. This is because the steam generator is allowed to cool to ambient perhaps only once or twice a year, and the startup times and energies will have little influence on annual plant performance.

Following an overnight shutdown, the estimates of startup times ranged from 1 hour to approximately 5 hours. These times are of more interest, because daily startup times and energies can have a measurable influence on the annual performance. Intuitively, the heat exchanger designs which put the low pressure salt on the shell side, and thereby minimize shell thicknesses, should offer the best transient response. However, this is not borne out by the vendor responses. A review of the table shows that one steam generator with a kettle evaporator (ABB Lummus) has a comparable startup time to the steam generators which exclusively place the low pressure salt on the shell side (Foster Wheeler and B&W / SAIC). Furthermore, the Struthers Wells startup time was based on a conservative temperature ramp rate (56 C/hr (100 F/hr)), which was known to result in acceptable thermal stresses. Struthers Wells stated that a shorter startup time is likely, but a detailed transient analysis would be required to determine the minimum.

It should also be noted that the steam generator startup times may not govern the startup time for the turbine plant. In particular, main and reheat steam temperature ramp rates in the steam generator must meet the allowable ramp rates specified by the turbine manufacturer. A survey of turbine designs on recent cogeneration and utility projects at Bechtel showed the following:

- Small (20 to 40 MWe) non-reheat turbines designed for cyclic service can be started following an overnight shutdown in approximately 0.5 hours. The turbines use separate high pressure and low pressure sections to achieve this transient performance. The size of the high pressure section is held to a minimum by operating at a high speed (10,000 rpm). A step-down gearbox connects the high speed section to the 3,600 rpm low pressure section and generator. The high pressure section also uses a vertical split case with separate inner and outer sections to minimize the thermal mass

- Large (100 to 200 MWe) reheat turbines designed for base load service generally require at least 2 hours for startup following an overnight shutdown. The principal rate limitations are imposed by the thick metal sections where the horizontally-split upper case joins the lower case. The design features noted above for small, cyclic duty turbines are not currently available in large turbines. However, the transient performance can be improved by incorporating features found in some European turbines designed for cyclic service. Specifically, electric or steam trace heating can be added to the case joint to reduce the startup times.

The startup times for ABB Lummus, Foster Wheeler, and B&W / SAIC steam generators are no longer, and may be shorter, than typical startup times for the turbine generator. This may also be true for the Struthers Wells design, depending on the results of further thermal analysis. Thus, all of the designs should be equally acceptable for commercial service.

ADVANTAGES AND DISADVANTAGES

Some of the qualitative advantages and disadvantages of each of the three heat exchanger concepts are discussed below in Table 3-5.

Table 3-5
HEAT EXCHANGER CONCEPT ADVANTAGES AND DISADVANTAGES

Item	Straight tube/straight shell	U tube/straight shell	U tube/U shell
Design	Less complex thermal analysis; more complex structural analysis (bellows)	More complex thermal analysis (tubesheet)	Less complex thermal and structural analyses
Fabrication	Least complex, if bellows is available	Average complexity Thick shell wall (if high pressure)	More complex (U bend closure)
Operation	Heat exchangers tolerant of rapid temperature changes, but limited by steam drum or bellows	Less tolerant of temperature changes, but limits may be set by turbine	Heat exchangers tolerant of rapid temperature changes, but limited by steam drum
Maintenance	More complex if bellows must be removed	Average complexity for tube plugging	Average complexity for tube plugging
Reliability	Bellows may need to be demonstrated	Significant design, fabrication, and operating experience	Good reliability shown at MSEE, but test duration was limited

It should be noted that all of the steam generator concepts are based on mature, commercial heat exchanger designs. The advantages and disadvantages noted above reflect relatively minor differences in complexity, and none of the approaches can be considered to be either clearly preferred or seriously disadvantaged.

The only area in which some reservations might be made is the requirement for bellows in the straight tube/straight shell concept. Foster Wheeler, in the 1982 study for Sandia National Laboratories, stated that sodium steam generators in European (SNR-300) and USSR (BN-600) nuclear breeder plants used expansion bellows in the shells. In addition, a Westinghouse sodium steam generator with bellows on the shell side of the heat exchanger is currently undergoing tests at the Rockwell International Energy Technology Engineering Center facility in Santa Susana, California. Nonetheless, a test program to demonstrate bellows reliability in nitrate salt service under moderate thermal cycling conditions may be required.

WARRANTY PROVISIONS

The warranty provisions outlined by each vendor are presented in Table 3-6.

Table 3-6
COMPARISON OF STEAM GENERATOR WARRANTY PROVISIONS

Item	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
First quality and free from defects	Yes	Yes	Yes	Yes
Performance guarantee	Not discussed	Not discussed	Not discussed	Could be provided
Repair after initial service date	12 months	12 months	12 months	12 months
Repair after delivery date	18 months	18 months	18 months	Not discussed
Liability for storage, operation, maintenance, erosion, corrosion, or alterations	Responsibility of project	Responsibility of project	Responsibility of project	None stated
Liability for consequential damages	Not discussed	Responsibility of project	Responsibility of project	Responsibility of project
Guarantee for fitness for a particular purpose	Not guaranteed	Not guaranteed	Not guaranteed	Not discussed

The provisions among the vendors are quite comparable, and indicate that the heat exchangers and auxiliary equipment will be commercial items supplied on a competitive basis.

CAPITAL COST ESTIMATES

The capital cost estimate for each steam generator consists of two elements: the investment cost for design, procurement, and installation, and the operating cost for the hot salt pumps to overcome the pressure drop through the heat exchangers. The later element can be converted to an equivalent capital cost to assist in the assessment of the four designs.

Design, Procurement, and Installation Cost Estimates

Design, procurement, and installation costs for the four steam generator designs are shown in Table 3-7. Several items are apparent from a review of the table. First, the Struthers Wells and B&W / SAIC estimates included not only the heat exchangers, but also the supporting items required for a complete steam generation system. These items included the inter-heat exchanger piping, insulation, trace heating, instrumentation, valves, structural steel, engineering, installation, and contingency. In contrast, the ABB Lummus and Foster Wheeler estimates included only the 4 heat exchangers, other shop costs, engineering, contingency, and fee.

Second, a comparison of the heat exchanger costs, presented in Table 3-8, shows reasonably good correlations among heat exchange areas, shell side fluids, weights, and unit costs. Specific observations include the following:

- Preheater - The unit weight costs from ABB Lummus and Struthers Wells are approximately $^1/_4$ those from Foster Wheeler and B& W / SAIC. The relatively complex fabrication of the U-shell in the B&W / SAIC design may account for its high unit costs (this is also the case for the B&W / SAIC evaporator, superheater, and reheater). However, the Foster Wheeler straight tube/straight shell should, in theory, be the least complex to fabricate, but this is not reflected in the estimates.

- Evaporator - The ABB Lummus and Struthers Wells evaporators are approximately twice as heavy as the Foster Wheeler and B&W / SAIC designs. This is a consequence of placing the high pressure water/steam on the shell side. However, there is not a cost penalty for doing so; the kettle evaporators are competitive with the other designs. The Foster Wheeler unit costs are also noticeably higher than the costs from the other vendors. This may be a consequence of integrating the steam drum with the evaporator.

- Superheater - To limit the temperature change across the tubesheet, the ABB Lummus design splits the superheater into two shells, and Struthers Wells places the high pressure steam on the shell side. The consequences are evident. The surface areas are $1^1/_2$ to 3 times the Foster Wheeler and B&W / SAIC requirements and the heat exchanger weights are 2 to 16 times as high. However, the unit area costs are competitive with the Foster Wheeler cost and only 60 percent greater than the B&W / SAIC estimate. This may be traced to the use of standard commercial heat exchanger designs by ABB Lummus and Struthers Wells.

  The unit weight cost of the ABB Lummus heat exchanger is only $^1/_3$ to $^1/_4$ that of the other designs, which can perhaps be attributed to the large tubesheets noted above in the discussion of VESSEL SHELL THICKNESSES AND WEIGHTS.

Table 3-7
COMPARISON OF STEAM GENERATOR COST ESTIMATES

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
DESIGN, PROCUREMENT, AND INSTALLATION COST				
- Preheater	$370,000	$415,000	$871,000	$690,000
- Evaporator	$1,230,000	$1,125,000	$1,626,000	$935,000
- Steam drum	Not applicable	Not applicable	Included with evaporator	$240,000
- Superheater	$1,950,000	$1,979,000	$625,000	$450,000
- Reheater	$600,000	$1,668,000	$495,000	$550,000
- Recirculation pumps	Not applicable	Not applicable	Not included	$539,000
- Other shop costs	Included in total	Included in total	$637,000	Included in total
- Salt Piping and Attemperators	Not included	$117,000	Not included	$719,000 (2)
- Steam Piping	"	$138,000	"	Not included
- Insulation	"	$47,000 (1)	"	$193,000 (2)
- Trace Heating			"	$213,000
- Preheat System	"	$110,000	"	Not applicable
- Instrumentation and Valves	"	$181,000	"	$503,000 (2)
- Support Steel	"	$246,000	"	$560,000 (2)
- Engineering	Included in total	$52,000	$600,000	$1,215,000
- Shipping	"	Included in total	Not included	$80,000
- Installation	Not included	$53,000	"	$299,000
- General Activities	Included in total	Included in total	Included in total	$286,000
- Contingency	"	"	$971,000 (3)	$1,121,000
- Home office costs	"	"	Included in total	Included in total
- Construction management	"	"	Not included	"
- Fee	"	"	$466,000 (4)	"
Total	$4,150,000 (5)	$6,131,000	$6,291,000 (6)	$8,593,000

1) Includes insulation and heat tracing
2) Includes installation labor costs
3) 20 percent of above costs
4) 8 percent of above costs
5) Cost for partial system; $7,400,000 estimated cost for complete system
6) Cost for partial system; $9,500,000 estimated cost for complete system

Table 3-7 (Continued)
COMPARISON OF STEAM GENERATOR COST ESTIMATES

	ABB Lummus	Struthers Wells	Foster Wheeler	B&W / SAIC
OPERATING COST				
Hot Salt Pumps				
- Flow rate, lb/sec	1,179	1,292	1,364	1,383
- Pressure drop, ft	123	65	166	162
- Power demand, kWe (7)	280	160	430	430
- Annual energy demand, kWhe (8)	980,000	560,000	1,505,000	1,505,000
- Annual energy cost (9)	$107,000	$62,000	$166,000	$164,000
- Equivalent capital cost (10)	$1,019,000	$590,000	$1,581,000	$1,562,000
Evaporator Recirculation Pump				
- Flow rate, lb/sec	Not required	Not required	Not required	102
- Pressure drop, ft				74
- Power demand, kWe (11)				14
- Annual energy demand, kWhe (8)				49,000
- Annual energy cost (9)				$5,000
- Equivalent capital cost (10)				$48,000
TOTAL OPERATING COST	$1,019,000	$590,000	$1,581,000	$1,610,000
TOTAL DESIGN, PROCUREMENT, INSTALLATION, AND OPERATING COST	$5,169,000 (12) $8,400,000 (13)	$6,721,000	$7,872,000 (12) $11,100,000 (13)	$10,203,000

7) Based on pump efficiency of 78 percent, motor efficiency of 95 percent, and variable speed drive efficiency of 96 percent.
8) Based on annual operating time of 3,500 hours
9) Based on auxiliary energy cost of $0.11/kWhe
10) Based on levelized capital carrying charge (fixed charge rate) of 10.5 percent
11) Based on pump efficiency of 70 percent and motor efficiency of 92 percent
12) Heat exchangers plus operating cost only
13) Estimated cost for complete system plus operating cost

Table 3-8
COMPARISON OF HEAT EXCHANGER UNIT COSTS

Item	ABB Lummus	Struthers Wells	Foster Wheeler[1]	B&W / SAIC[2]
Preheater				
- Heat exchange area, m^2	1,343	1,320	1,959	2,049
- Shell side fluid	Low pressure	High pressure	Low pressure	Low pressure
- Weight, kg	43,000	52,600	54,000	35,700
- Cost estimate	$370,000	$415,000	$1,515,000	$794,000
- Unit costs				
$/m²	280	310	770	390
$/kg	8.6	7.9	28.1	22.2
Evaporator and steam drum				
- Heat exchange area, m	1,550	2,288	1,853	2,152
- Shell side fluid	High pressure	High pressure	Low pressure	Low pressure
- Weight, kg	170,000	218,000	122,400	119,400
- Cost estimate	$1,230,000	$1,125,000	$2,828,000	$1,351,000
- Unit costs				
$/m²	790	490	1,530	630
$/kg	7.2	5.2	23.1	11.3
Superheater				
- Heat exchange area, m	1,162	1,556	827	566
- Shell side fluid	Low pressure	High pressure	Low pressure	Low pressure
- Weight, kg	194,000	65,300	28,500	12,100
- Cost estimate	$1,950,000	$1,979,000	$1,087,000	$518,000
- Unit costs				
$/m²	1,680	1,270	1,310	920
$/kg	10.0	30.3	38.1	42.8
Reheater				
- Heat exchange area, m	1,006	1,568	576	788
- Shell side fluid	High pressure	High pressure	Low pressure	Low pressure
- Weight, kg	57,000	62,200	20,900	12,700
- Cost estimate	$600,000	$1,668,000	$861,000	$632,000
- Unit costs				
$/m²	600	1,060	1,490	800
$/kg	10.5	26.8	41.2	49.8

1) Heat exchanger costs include other shop costs, engineering, management, contingency, and fee
2) Heat exchanger costs include a contingency of 15 percent

- Reheater - The consequences of placing the high pressure steam on the shell side of the ABB Lummus and Struthers Wells reheaters are evident. The surface areas are 2 to 3 times the Foster Wheeler and B&W / SAIC requirements, and the heat exchanger weights are 3 to 5 times as high. However, this pattern does not hold for the unit costs.

Third, a comparison of the estimates from B&W / SAIC and Struthers Wells shows wide variations in the costs for salt piping, insulation, heat tracing, instrumentation, engineering, and installation. For example, the figures for engineering and installation vary by factors of 25 and 6, respectively. These two costs may reflect extensive experience at Struthers Wells in the design, fabrication, and installation of similar heat exchangers, while the higher B&W / SAIC estimate may be an indication that only one of this type has been fabricated. Alternately, the higher B&W / SAIC estimate may reflect a more extensive background with, and a more thorough knowledge of, the costs associated with nitrate salt systems. Nonetheless, the Struthers Wells estimate is probably optimistic in several areas. For example, the surface area of the heat exchanger shells is approximately 370 m^2 (4,000 ft^2) and the estimated cost for insulation and heat tracing is \$47,000. This is equivalent to a unit cost of \$125/m^2 (\$12/ft^2). In comparison, typical insulation costs (without heat tracing) used by Bechtel for conceptual estimates are in the range of \$200 to \$325/m^2 (\$20 to \$30/ft^2).

Finally, in an attempt to place the ABB Lummus and Foster Wheeler scopes of supply and cost estimates on the same basis as the other vendors, costs were added for the following: salt piping and attemperators, insulation and heat tracing, instrumentation and valves, support steel, shipping, installation, and contingency. For the purposes of this study, the B&W / SAIC costs were used in each category. The adjustment totaled \$3,250,000, which brought the ABB Lummus design, fabrication, and installation cost to \$7,400,000, and the Foster Wheeler cost to \$9,400,000.

Using the adjusted figures for the ABB Lummus and Foster Wheeler estimates, the 4 steam generation system estimates are within 20 percent of their average. This is in excellent agreement at this level of engineering definition.

Operating Cost Estimates

The operating cost for the steam generator is the decrease in annual plant output and revenue due to the auxiliary electric demand of the following pumps:

- Hot salt pumps to overcome the pressure drop through the heat exchangers
- Evaporator recirculation pumps to overcome the pressure drop through the evaporator-steam drum circuit (B&W / SAIC design only)

The operating cost was converted to an equivalent capital cost, and this cost added to the design, procurement, and installation cost, to evaluate the overall economics of each design.

The hot salt pump auxiliary power demand was calculated using the following:

- Salt flow rate and heat exchanger pressure drops as stated in each vendor report

- A control valve pressure drop of 8.5 m (28 ft), as listed in the B&W / SAIC report, was used in all steam generator systems for consistency

- Pump efficiency of 78 percent, motor efficiency of 95 percent, and a variable speed drive efficiency of 96 percent

The hot salt pump power demands included 160 kWe (215 hp) for the Struthers Wells steam generator, 280 kWe (375 hp) for ABB Lummus, and 430 kWe (575 hp) for Foster Wheeler and B&W / SAIC.

The B&W / SAIC evaporator recirculation pump demand was calculated using the following:

- Recirculation flow rate of 46.4 kg/sec (368,000 lb/hr)

- Total developed head of 23 m (75 ft) to compensate for the 138 kPa (20 psi) pressure drop in the evaporator

- Pump efficiency of 70 percent and a motor efficiency of 92 percent

This resulted in a pump power demand of 16 kWe (21 hp)

A summary of the annual energy demand and operating costs for the four steam generator designs is shown in Table 3-7. The operating costs were converted to equivalent capital costs using the following equation

$$\text{Equivalent Capital Cost} = \frac{\text{Electric energy demand} * \text{Marginal electric energy cost} * \text{Annual operating time}}{\text{Levelized Capital Carrying Charge}}$$

where

- The marginal cost of electric energy is assumed to be \$0.11/kWhe. This is the levelized cost of energy developed for the first commercial 100 MWe plant in Phase 1 of the central receiver Utility Studies (Ref. 1-1)

- The annual operating time of the hot salt pumps was assumed to be 3,500 hours

- The levelized capital carrying charge (fixed charge rate) was 10.5 percent. The rate, based on standard utility project financing and a constant year dollar analysis, was that used during Phase 1 of the Utility Studies

The results of the calculations are also shown in Table 3-7. The equivalent capital cost for operation ranged from 6 to 20 percent of the design, procurement, and installation cost. Thus, the pressure drop through the heat exchangers can influence the relative economics of competing designs. Note that the higher pressure drop in the Foster Wheeler and B&W / SAIC designs entail an economic penalty of approximately \$500,000 relative to the ABB Lummus design and \$1,000,000 relative to the Struthers Wells approach

Design, Procurement, Installation, and Operating Cost Estimates

The sum of the design, fabrication, installation, and operating cost estimates are also shown in Table 3-7. From a review of the estimates, the following observations can be made:

- There is good agreement among the vendors regarding the costs of the heat exchangers; the divergence in the estimates occurs in the auxiliary equipment, engineering, and installation required for a complete system.

- The subcontract price developed by Babcock & Wilcox for the U-tube/U-shell steam generator in Phase 1 of the Utility Studies was $11,128,000 (third quarter 1987 dollars). Escalating this price to first quarter 1993 dollars using an annual rate of 4 percent yields an estimate of $13,800,000. The steam generators developed for this study, including the U-tube/U-shell approach, are considerably less expensive than the Utility Studies design. This may be attributed to the successful use of relatively lower cost kettle boilers in the Luz parabolic trough solar power plants, and renewed vendor interest in commercial central receiver projects following the start of the Solar Two Project.

- It appears that a steam generator for a 100 MWe commercial project can be fabricated and installed for approximately $8 million.

Section 4
Thermal Storage System Hot Salt Tank Designs and Cost Estimates

Three conceptual hot salt storage tank designs and cost estimates were developed during this study. Two of the designs, one developed by Chicago Bridge and Iron Technical Services Company (CBI) and a second by Pitt-Des Moines, Inc. (PDM), employed a stainless steel tank with external insulation. The third design, developed by S. N. Technigaz (a French company), used a carbon steel tank with external insulation. To limit the carbon steel shell temperature to acceptable values, a layer of internal refractory insulation was required. In addition, a thin Incoloy liner was required to protect the refractory from the corrosive effects of the nitrate salt at 566 C (1,050 F).

The discussion which follows reviews the storage tank specification, design features, warranty provisions, and cost estimate for each of the concepts.

SPECIFICATION

The principal specifications used in the design of all of the tanks is presented in Table 4-1. The nominal storage capacity is 1,560 MWht, which translates to an active volume of 7,690 m^3 (272,000 ft^3). An inactive volume of salt at the bottom of the tank (heel) with a depth of 0.9 m (3 ft) was specified to minimize periodic thermal transients in the joint between the floor and wall. A 1.2 m (4 ft) high space at the top of the tank was also specified to hold the heel from the cold storage tank and the salt inventory in the receiver and thermal storage systems. Freeboard above the 1.2 m space, if any, was to be selected by the vendor to accommodate liquid movement during an earthquake. A value of $1,700/kWt was assigned to heat loss through the tank to assist the vendor in selecting the optimum insulation thicknesses.

Nitrate salt tanks operating at this combination of size and temperature have yet to be fabricated and tested. However, several tanks have been built over the past several years that meet or exceed the size or temperature requirements of the hot salt storage tank. Representative tanks, with external insulation, include the following:

- Four bitumen tanks, each 88 m (288 ft) in diameter and 15 m (48 ft) high, were fabricated for Bechtel at the Syncrude Tar Sands Project in Mildred Lake, Canada. The externally insulated tanks operated at 175 to 230 C (350 to 450 F) and used forced air circulation to cool the foundations.

- A nitrate salt tank, 14 m (45 ft) in diameter and 2.8 m (9 ft) high, was fabricated by CBI for a proprietary chemical process plant in Texas. The externally insulated tank operates at 260 to 450 C (500 to 842 F) and uses natural convection air circulation to cool the foundation.

- The thermal storage system for the Luz Solar Electric Generating Station I parabolic trough solar power plant near Barstow, California. The cold tank, 21.0 m (69 ft) in diameter and 12.2 m (40 ft) high, stores a synthetic oil at 250 C (480 F), and the hot tank, 22.1 (72.5 ft) in diameter and 12.2 m (40 ft) high, stores oil at 315 C (600 F).

Table 4-1
HOT SALT TANK PERFORMANCE SPECIFICATION

Active tank volume	7,693 m^3 (271,674 ft^3)
Additional tank volumes	
Heel	0.91 m (3 ft)
Drain down from receiver and thermal storage systems	1.22 m (4 ft)
Freeboard	To be selected by vendor
Nitrate salt density	2.090-0.000636*(Temp, C); g/cm^3 (131.2-0.02221*(Temp, F); lb_m/ft^3)
Equivalent capital cost of heat loss through the insulation	$1,700/kWt
Maximum temperature of insulation exposed to ambient	60 C (140 F)
Heat tracing system	Electric elements to be used at 50 percent of rating; spare circuit to be installed
Leak detection system	To be specified by vendor
Cooled foundation	To be specified by vendor
Seismic accelerations	API Standard 650 Zone 3
Wind loads	40 m/sec (90 mph) at 10 m above grade
Soil bearing capacity	0.24 MPa at 1.5 m below grade (5,000 psf at 5 ft) 0.48 MPa at 3.0 m below grade (10,000 psf at 10 ft)

- Oil and asphalt storage tanks for American Petrofina in Port Arthur, Texas. The cold tank, 45.7 m (150 ft) in diameter and 14.6 m (48 ft) high, operates at 175 C (350 F), and the hot tank, 24.4 (80 ft) in diameter and 14.6 m (48 ft) high, operates at 260 C (500 F)

- Oil and asphalt storage tanks for ARAMCO in Qasim, Saudi Arabia. The cold and hot tanks, operating at 175 C and 220 C (350 F and 430 F), respectively, have dimensions of 57.9 m (190 ft) in diameter and 4.6 m (15 ft) high

- Nitrite salt thermal storage tanks for the MRI / SOLERAS solar desalination plant in Yanbu, Saudi Arabia. The cold and hot tanks, operating at 250 C and 315 C (480 F and 600 F), respectively, have dimensions of 4.9 m (16 ft) in diameter and 4.9 m (16 ft) high

- 112 MWht thermal storage tank for the 10 MWe Solar One pilot plant near Barstow, California. The tank operated on the thermocline principle, and contained 6,180 metric tons (6,800 tons) of rock and sand and 910 m^3 (240,000 gallons) of synthetic oil. During the charging cycle, oil entered the tank at 305 C (580 F), and during the discharging cycle, oil entered at 220 C (425 F).

The alternate tank design, using a low cost carbon steel shell with internal refractory insulation, has been proposed by Martin Marietta Corporation. The liner concept was originally developed by Technigaz for liquified natural gas storage tanks, and has been successfully used in 15 ship and 20 shore facilities during the past 20 years. The idea was extended to high temperature nitrate salt storage by Martin Marietta Corporation and successfully tested in the 7 MWht thermal storage system Subsystem Research Experiment at Sandia National Laboratories. The experimental tank, 3 m (10 ft) in diameter and 6.2 m (20.5 ft) high, operated at 566 C (1,050 F) and used forced water circulation to cool the foundation.

Thus, the externally and internally insulated tank designs proposed in this study can be viewed as moderate extrapolations of current experience.

DESIGN FEATURES

Elevation drawings for the CBI, PDM, and Technigaz tank designs are shown in Figures 4-1 through 4-3, respectively. The principal design features of the three concepts are summarized in Table 4-2.

Externally Insulated Tanks

As might be expected, the two externally insulated tank designs were quite similar. Each tank was 29.0 m (95 ft) in diameter, fabricated from 316 stainless steel, insulated with mineral wool, and supported at the walls by a perimeter ring wall. In addition, the foundations were cooled by air passages to limit the temperature of the natural soil, and the shell and floor plate thicknesses in one design were within 25 percent of the thicknesses in the other. The principal differences were as follows:

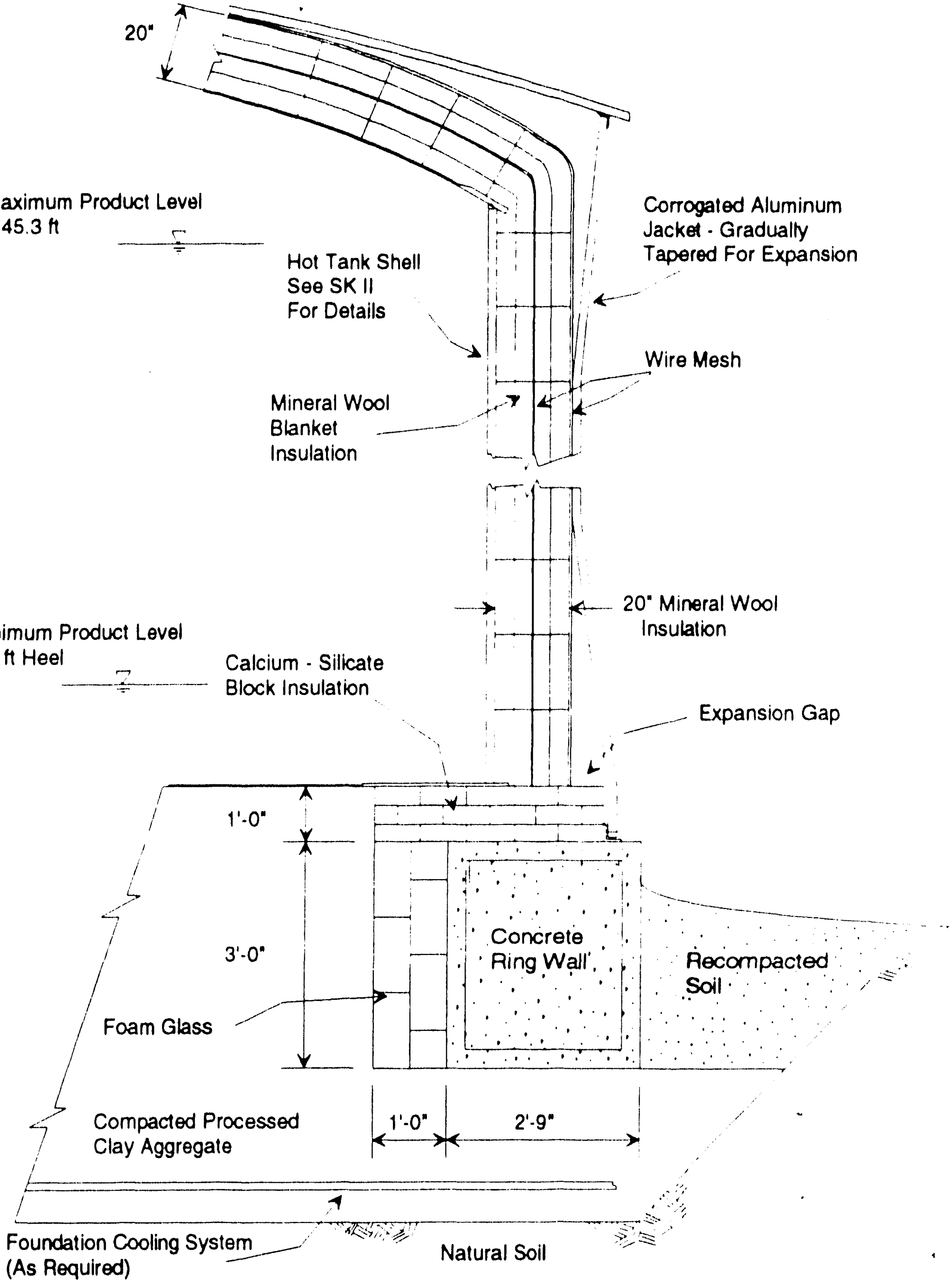

Figure 4-1 Chicago Bridge and Iron Hot Salt Tank Elevation Diagram (Sheet 1 of 2)

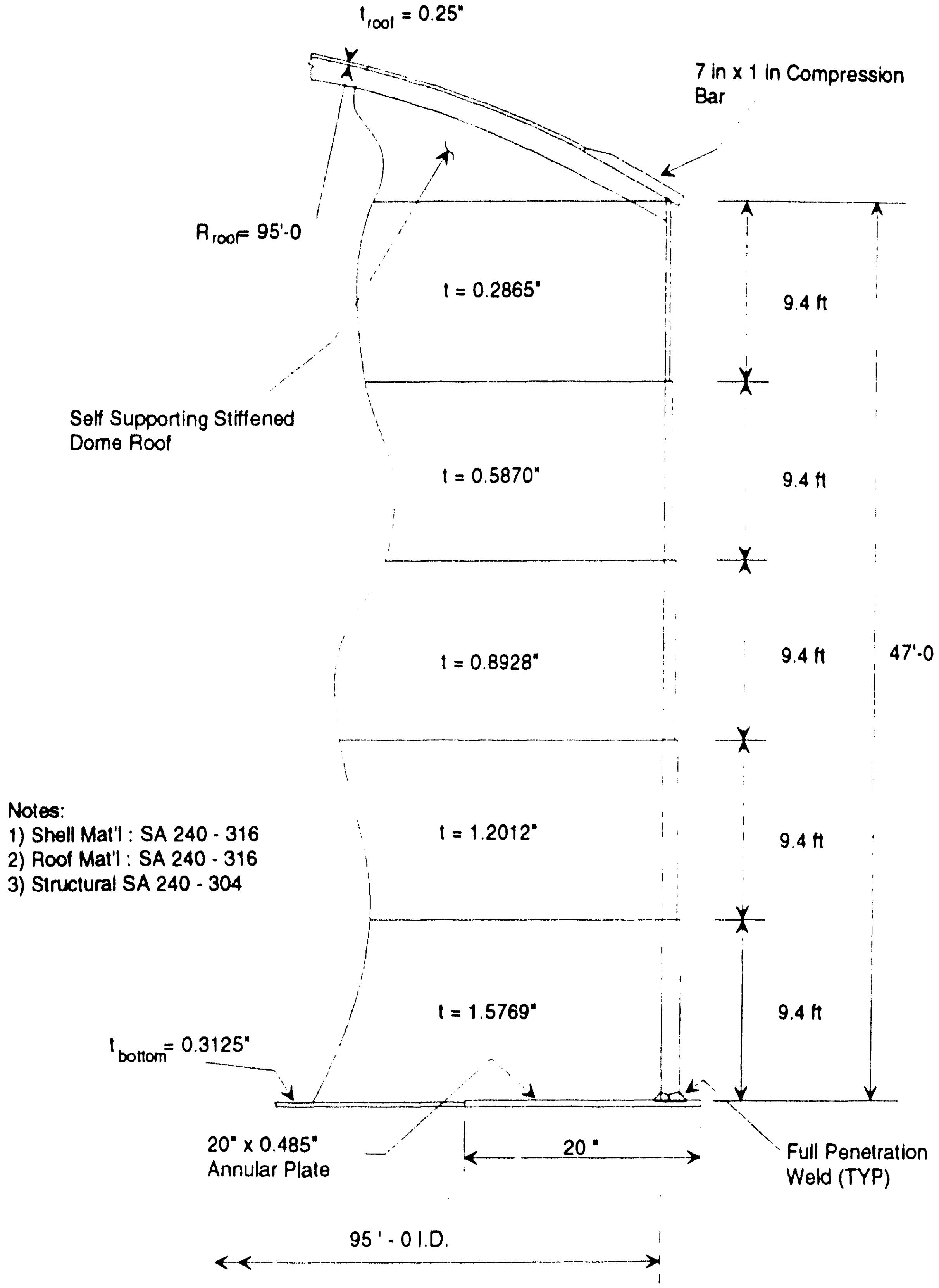

Figure 4-1 Chicago Bridge and Iron Hot Salt Tank Elevation Diagram (Sheet 2 of 2)

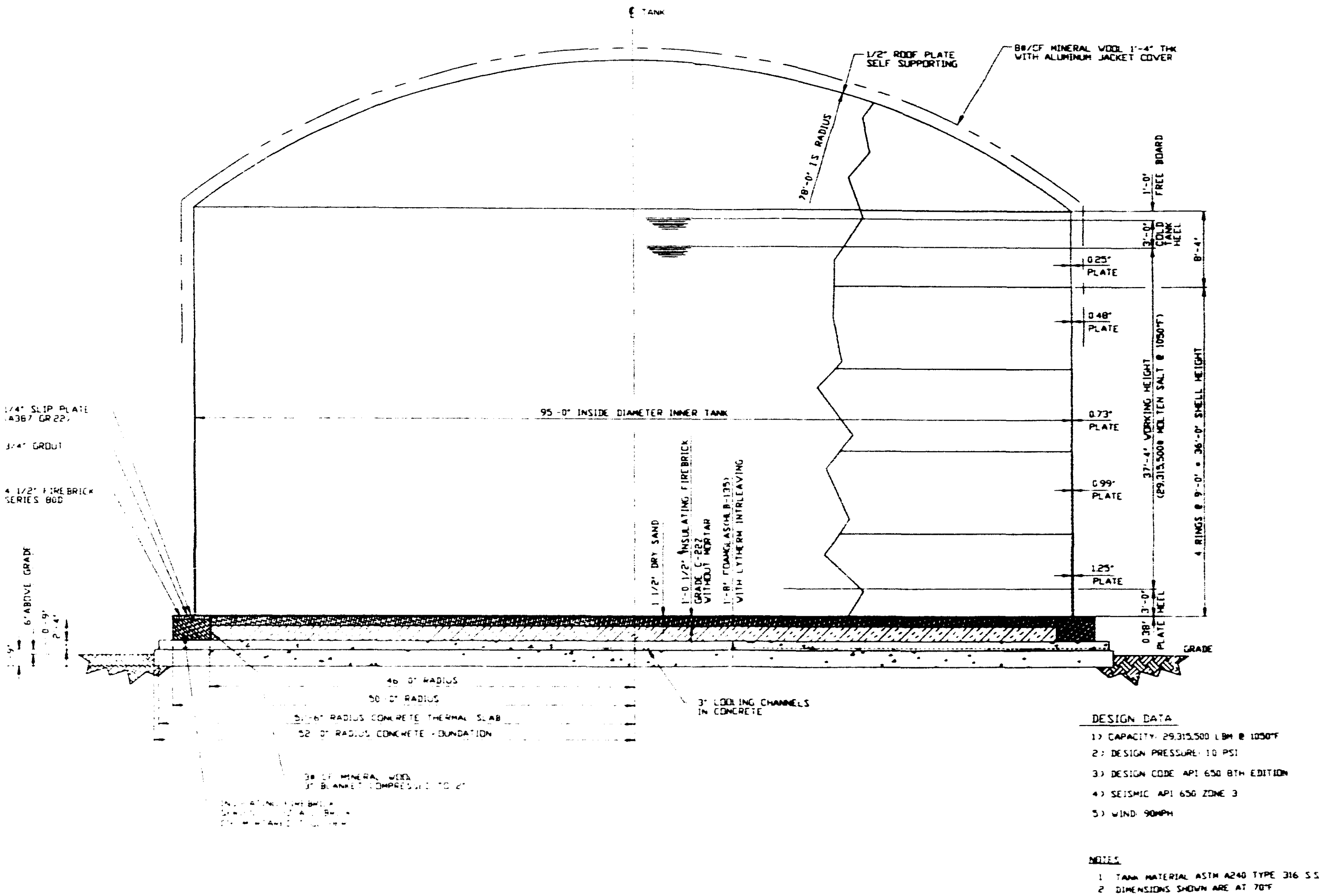

Figure 4-2 Pitt-Des Moines Hot Salt Tank Elevation Diagram

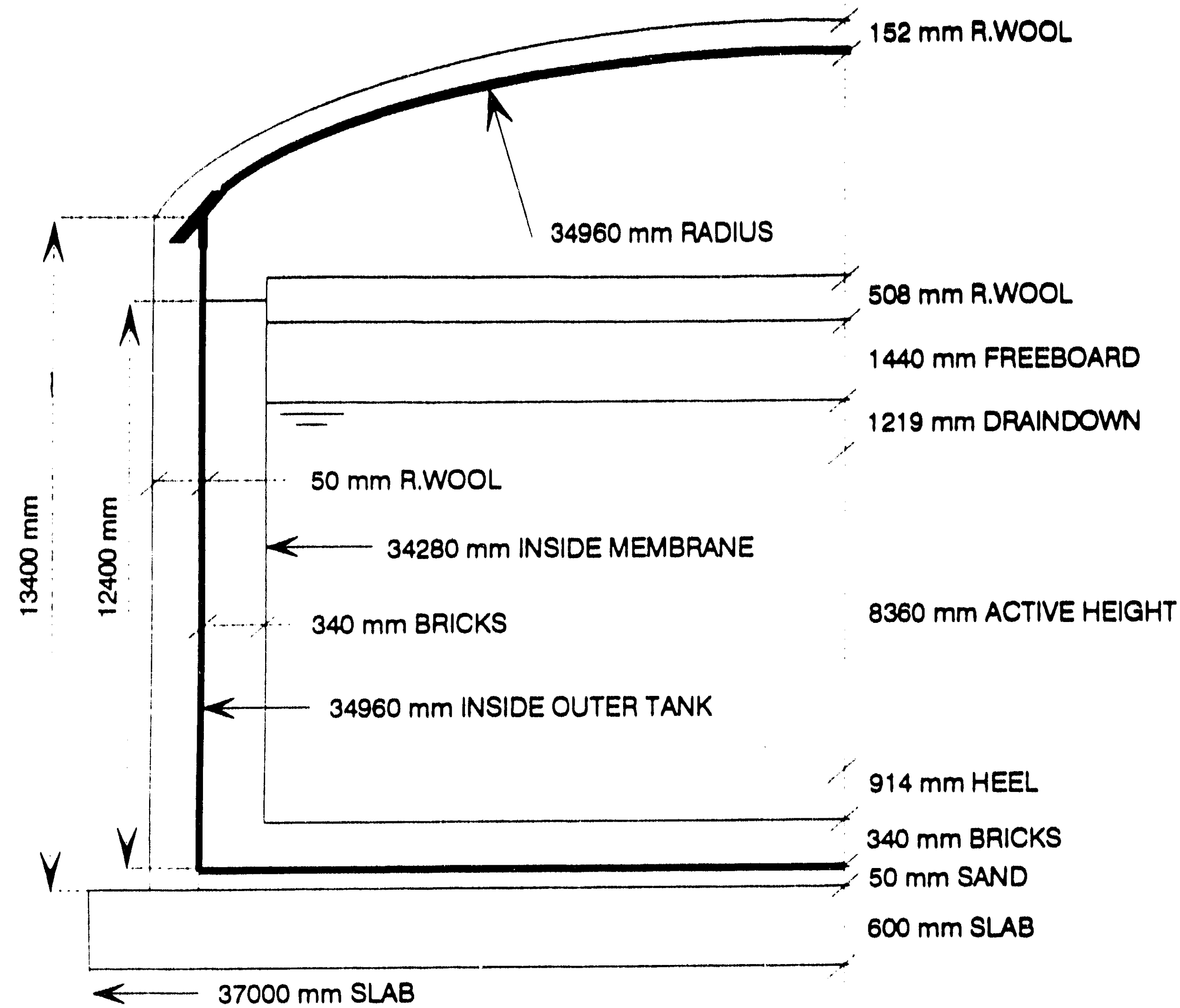

Figure 4-3 Technigaz Hot Salt Tank Elevation Diagram (Sheet 1 of 2)

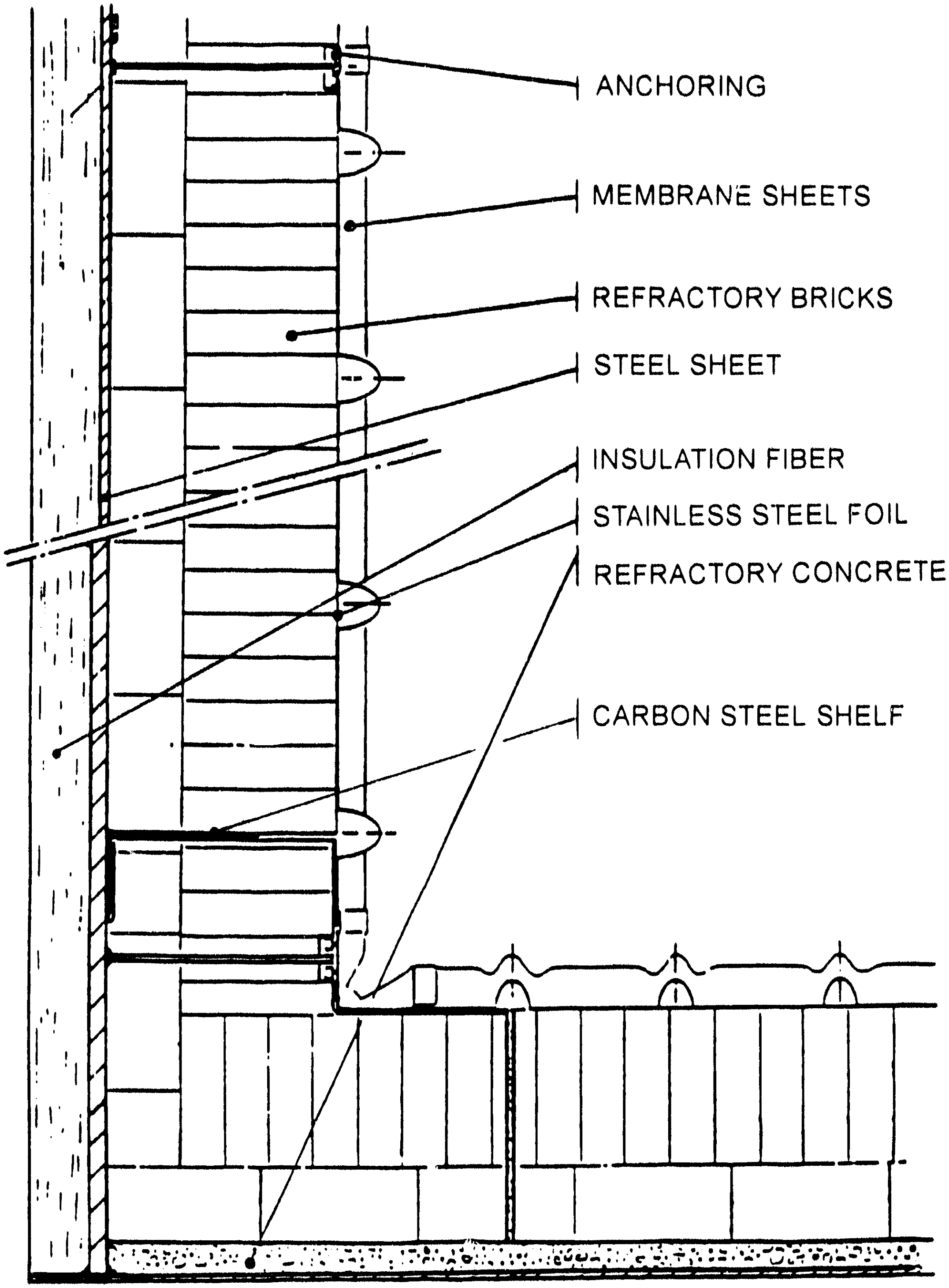

Figure 4-3 Technigaz Hot Salt Tank Elevation Diagram (Sheet 2 of 2)

Table 4-2
COMPARISON OF HOT SALT TANK TECHNICAL CHARACTERISTICS

	Chicago Bridge and Iron	Pitt-Des Moines	Technigaz
DIMENSIONS, ft			
- Outer tank diameter	95.0	95.0	114.8
- Inner tank diameter	Not applicable	Not applicable	112.5
- Height	47.0	44.3	44.0
- Roof radius	95.0	78.0	114.7
PLATE THICKNESS, in.			
- Roof	0.25	0.50	0.25
- Shell: top	0.2865	0.25	0.25
bottom	1.5769	1.25	1.2795
- Floor	0.3125	0.38	0.3740
MATERIALS			
- Roof	316 stainless steel	316 stainless steel	Carbon steel - A516 Gr 70
- Shell	316 stainless steel	316 stainless steel	Carbon steel - A516 Gr 70
- Liner	Not applicable	Not applicable	Incoloy 800 (0.05 in.) with stainless steel foil back (0.01 in.)
- Floor	316 stainless steel	316 stainless steel	Carbon steel - A516 Gr 70
- Insulation			
External	Roof and shell - mineral wool (20 in.)	Roof and shell - mineral wool (16 in.)	Mineral wool - roof (6 in.); shell (2 in.)
Internal	Not applicable	Not applicable	Roof - mineral wool (20 in.) Shell and floor - refractory brick (13.4 in.)
- Foundation (top to bottom)			
Perimeter	Calcium silicate block (12 in.) Reinforced concrete (33 in. W x 36 in. H) Foamglas (12 in. W x 36 in. H)	Steel slip plate (1/4 in.) Grout (3/4 in.) Firebrick (4 1/2 in.) Insulating firebrick (28 in.)	Not specified
Center	Compacted local soil (48 in.)	Dry sand (1 1/2 in.) Insulating firebrick (12 1/2 in.) Foamglas (20 in.) Thermal concrete (9 in.) Reinforced concrete (21 in.)	Dry sand (2 in.) Reinforced concrete (24 in.)
Soil	Compacted local soil (12 in.)	Compacted local soil (48 in.)	Compacted local soil (36 in.)

- CBI included a freeboard of approximately 0.5 m (1.7 ft), while PDM selected a height of 0.30 m (1.0 ft). As discussed in the CBI report, the freeboard was included to accommodate liquid movement during an earthquake.

- The technical specification called for a height of 1.22 m (4 ft) to store the drain down from the receiver and thermal storage systems. However, PDM included a height of only 0.91 m (3 ft).

- CBI selected an optimum insulation thickness of 50 cm (20 in.), while PDM used 40 cm (16 in.). The difference was likely due to different assumptions regarding unit insulation costs, as discussed below under CAPITAL COST ESTIMATE.

- The PDM design used a steel slip plate at the ring wall and a layer of sand near the foundation center to reduce friction loads due to thermal expansion; the CBI floor at the ring wall rested directly on calcium silicate block insulation.

- Under the center of the PDM tank, the foundation consisted of layers of insulating firebrick (320 mm (12½ in.)), foamglas (510 mm (20 in.)), thermal concrete (230 mm (9 in.)), and reinforced concrete (530 mm (21 in.)). Cooling air ducts, 75 mm (3 in.) wide, passed through the thermal concrete layer to limit the temperature of the reinforced concrete, and native soil beneath the concrete, to acceptable levels.

- Under the center of the CBI tank, the foundation consisted of at least 1.2 m (4 ft.) of compacted clay aggregate. The final thickness was to be determined from a detailed thermal analysis during final design. A foundation cooling system, consisting of water or forced air pipes located near the bottom of the clay, would be provided if the selected insulation thickness did not limit the native soil temperatures to less than 100 C (212 F).

- Leaks in the bottom of the PDM tank were to be detected by continuous lengths of temperature sensitive elements located under the top layer of the foamglas insulation. The elements were installed on a 3 m by 3 m (10 ft by 10 ft) grid, and would activate when the temperature of any 50 mm (2 in.) portion exceeded 480 C (900 F).

- Leaks in the bottom of the CBI tank were to be detected by rows of equally spaced thermocouples located in the clay foundation. The thermocouples would be installed in conduits to simplify repair or replacement.

A review of the CBI and PDM reports by a tank designer within Bechtel provided the following observations:

- Stainless steel tanks, operating at this size and temperature and using conventional shell-to-floor joints, should be feasible.

- The foundation bearing pressure of 290 kPa (6,000 lb/ft^2) on the calcium silicate and foam glass was at the upper end of conventional practice.

- The bricks in the ring wall may be subject to settlement, which could lead to stress gradients in the tank bottom. The recommended approach would substitute light weight refractory concrete for the bricks

- The sand in contact with the tank bottom in the PDM design would need to be free of chlorides.

Internally Insulated Tank

The Technigaz concept was quite different from the CBI and PDM approaches. The design was based on the criteria that 1) stainless steel tanks are quite expensive, 2) the joint where the wall meets the floor is sensitive to fatigue failure, and 3) there may be a need during the life of the plant to rapidly transfer salt from the cold tank to the hot tank. To satisfy these criteria, the following approach was used:

- The pressure boundary (floor, wall, and roof) was fabricated from carbon steel

- A high temperature refractory lining was installed inside the carbon steel tank to transfer hydrostatic loads to the pressure boundary and to provide sufficient resistance to conduction heat transfer such that the carbon steel temperature did not exceed 370 C (700 F). The lining consisted of 512,000 bricks, each 23 cm by 11.5 cm by 6.5 cm (9 in. x 4½ in. x 2½ in.)

- A corrugated Incoloy 800 liner, 1.27 mm (0.05 in.) thick, was installed inside the refractory to isolate the refractory from the corrosive effects of nitrate salt at 566 C (1,050 F). The liner also prevented salt migration into cracks in the refractory which could result in local high temperature areas on the carbon steel shell. The corrugations, illustrated in Figure 3-4, allowed the liner to expand and contract, thus ensuring that all of the hydrostatic loads are transferred through the liner to the refractory bricks. The liner flexibility also accommodated rapid temperature transients with minimum fatigue damage. The liner concept was originally developed for liquified natural gas storage tanks and has been successfully used in 15 ships and 20 shore installations over the past 20 years. The concept was extended to high temperature nitrate salt storage by Martin Marietta Corporation and successfully tested in the 7 MWht thermal storage system Subsystem Research Experiment at Sandia National Laboratories

- A stainless steel foil barrier, 0.25 mm (0.01 in.) thick, was installed between the Incoloy liner and the refractory to prevent abrasion of the refractory during thermal transients

- An insulated concrete foundation, cooled by an array of water pipes, was used to limit the carbon steel floor temperature to 370 C (700 F) or less. Heat from the foundation was rejected to the atmosphere by a 350 kWt (1,200,000 Btu/hr) wet, mechanical draft cooling tower

- A suspended ceiling, fabricated from corrugated liner material and backed by 50 cm (20 in.) of mineral wool insulation, was installed inside the tank. The support structure for the ceiling required 29 tons of stainless steel members

- Mineral wool external insulation, 5 cm (2 in.) thick on the wall and 15 cm (6 in.) thick on the roof, limited the heat loss from the carbon steel shell.

- Any leaks through the Incoloy liner are to be detected by a system consisting of a gas chromatograph, 12 circuits of 12 mm (½ in.) stainless steel tubing located behind the liner corrugations, 12 solenoid valves, a vacuum pump, and a data acquisition computer. Gas samples are continuously taken from each of the circuits in succession. Should a leak develop, the chromatograph will identify nitrates in the sample gas and the computer will record the location and rate of change in the nitrate concentration.

The Technigaz design had only one feature in common with the CBI and PDM designs; the tank wall and floor thicknesses are essentially the same as the corresponding thicknesses in the stainless steel tanks. In all other respects, the two approaches were quite different, as described below.

- The height and diameter of the active salt volume in the Technigaz design was 8.36 m (27.4 ft) and 34.3 m (112.5 ft), respectively, for a height-to-diameter ratio of 0.24. The corresponding dimensions in the CBI and PDM designs were approximately 13.4 m (44.1 ft) and 29.0 m (95.0 ft), for an aspect ratio of 0.46.

- To accommodate the 340 mm (13.4 in.) thick internal refractory insulation on the walls and floor, and the 500 mm (20 in.) thick internal mineral wool insulation in the suspended ceiling, the surface area of the carbon steel outer tank must be 111 percent of the surface area of the Incoloy liner.

- The temperature gradient through the mineral wool exterior insulation was approximately 45 C/cm (205 F/in.) at the wall and 15 C/cm (70 F/in.) on the roof. In contrast, the temperature gradient through the wall and roof insulation in the CBI design was 10 C/cm (45 F/in) and through the PDM insulation, 13 C/cm (60 F/in.). The higher gradients through the Technigaz insulation resulted in greater heat losses, but this situation cannot be avoided. Greater insulation thicknesses will result in carbon steel shell temperatures which exceed the design value of 288 C (550 F).

- On a similar basis, the Technigaz foundation required an active cooling system to prevent the tank floor temperature from exceeding 288 C (550 F). Heat loss through the bottom of the tank was approximately 5 times the loss through the bottom of the PDM tank and 8 times the loss through the CBI tank.

- The integrity of the corrugated liner was an essential element in the Technigaz concept. To ensure that leaks were identified as quickly as possible, an active leak detection system was required. In contrast, leaks were detected passively in the CBI and PDM designs.

THERMAL LOSSES AND TRANSIENT PERFORMANCE

One of the principal features cited for the internally insulated tank is its ability to accommodate rapid temperature transients, and thereby avoid the use of heat tracing for maintaining constant temperatures during overnight or extended shutdowns. However, the importance of this feature is perhaps mitigated by two observations.

First, transient thermal storage tank models developed by Sandia National Laboratories predict that an empty hot salt tank will cool overnight at a rate of 1 C (2 F) per hour. The following morning, as salt from the receiver is introduced into the tank at an average temperature of 454 C (850 F), the tank will initially cool at a rate of 55 C (100 F) per hour. During the next 30 minutes, the temperature of the salt from the receiver will increase to the normal outlet value of 566 C (1,050 F). Once this temperature is reached, the tank will heat at a rate of approximately 22 C (40 F) per hour. Discussions with CBI and PDM indicate that large tanks can routinely tolerate temperature ramp rates up to 56 C (100 F) per hour without suffering excessive creep or fatigue damage. In addition, ramp rates greater than this may also be acceptable, but a detailed thermal stress and fatigue damage analysis would be required to verify the operating procedures. Representative experience with large, externally insulated tanks which tolerate temperature transients at least as severe than those anticipated for a commercial solar project can also be found. For example, the thermal storage tanks for the SEGS I parabolic trough solar power plant are 21 m (70 ft) in diameter and routinely accommodate temperature change rates of 40 to 55 C (75 to 100 F) per hour. In addition, a nitrate salt tank 14 m (45 ft) in diameter fabricated by CBI for a proprietary chemical process plant in Texas normally operates at 260 C (500 F), but is periodically filled very quickly with salt at 450 C (842 F).

It can be noted that the transient performance noted above applies to tanks with conventional shell-to-floor joints, in which the vertical shell is joined to the horizontal floor by a full penetration weld. Preliminary creep-fatigue calculations by CBI using ASME Code Case N-47 show the joint stresses to be fully consistent with a 30 year life. However, if a detailed transient thermal and structural analysis shows that this is not the case, an alternate design is available. The alternate uses floor-to-shell transition joints with a double curvature, commonly referred to as "knuckles", which eliminate the orthogonal corner. The vertical radius of the joint is approximately 0.9 m (3 ft) and the horizontal radius is the tank radius, in this case, 14.5 m (47.5 ft). The knuckle joints, which are formed with a large press and die, are often used in the shell-to-roof joints of large petroleum and water tanks. This joint is estimated to increase the price of the tank by only 3 to 5 percent, and may be a feature in the first commercial plant to reduce the technical risk.

Second, heat losses for externally insulated tanks are considerably less than for the internally insulated design as shown in Table 4-3. The losses for the CBI and PDM designs are comparable, with lower values for the CBI design likely due to differences in the insulation thickness (50 cm (20 in.) vs. 40 cm (16 in.)). Losses from the roof of the Technigaz tank were also comparable to the CBI and PDM roofs. This can be traced to the similar insulation materials and thicknesses on all three tanks. However, losses from the Technigaz wall and foundation were significantly greater than the corresponding losses from the externally insulated tanks. This can be traced directly to the relatively high thermal conductivity of the refractory bricks and the need to maintain the carbon steel shell temperature at or below 260 C (550 F).

Table 4-3
COMPARISON OF HOT SALT TANK THERMAL LOSSES

	Chicago Bridge and Iron	Pitt-Des Moines	Technigaz
Roof	73.8	95.1	65.5
Wall	129.0	155.4	311.5
Floor	41.7	70.0	344.3
	-------	-------	-------
Total	244.4	320.6	721.3

During an extended shutdown, the hot tank will cool to 266 C (550 F), at which time electric energy is used to maintain the temperature of the inventory. Following the restart of the receiver, the tank may be subject to a rapid change in the temperature of the inventory. Depending on the results of a detailed thermal analysis, the tank and inventory may need to be preheated prior to the restart of the receiver to avoid excessive thermal stresses. If so, the electric energy for preheating should be included in the comparisons of the tank designs. However, the steady state thermal loss from the internally insulated tank is greater than the loss from an externally insulated design. Therefore, some annual quantity of heat tracing for the externally insulated tank can be used before the annual performance of the two designs is equal. A first order thermal analysis shows the steady state loss from the internally insulated tank to be approximately 2.5 times the average of the thermal losses from the CBI and PDM designs. Assuming a Rankine cycle efficiency of 40 percent, the electric heat tracing on the externally insulated tanks could, in theory, be operated continuously and still offer the same annual thermal efficiency as the internally insulated design. Clearly, tank designs requiring such an operating strategy would not be proposed. However, it is apparent that the periodic use of trace heating on externally insulated tanks, should it be needed, can be justified.

LEAK REPAIR TIMES

The vendors were asked to develop procedures and estimated times to repair a leak. The most complete response was provided by Pitt-Des Moines, as follows:

Activity	Time or Manhours
Tank cool down	24 to 48 hours
Tank opening	40 to 60 manhours
Leak location	8 to 40 manhours
Leak repair	8 to 32 manhours
Non-destructive examination	4 to 8 manhours
Tank closing	40 to 60 manhours
Startup (ambient to 260 C (500 F))	48 to 72 hours

To cool the tank within 24 to 48 hours, two holes are opened in the roof and air is circulated through the interior by means of a fan. Air is also forced through the foundation cooling passages to limit the heat transferred from the foundation into the tank. The labor required to locate and repair a leak is estimated to be 100 to 200 manhours. Assuming that the repair crew consists of 2 men, and 3 shifts work each day, the time to complete the repair should be 2 to 4 days. The tank is then brought from ambient temperature to 260 C (500 F) over the course of 2 to 3 days by means of electric heat tracing. Thus, it appears that a leak could be located and repaired, and the tank filled, in 5 to 9 days.

The Technigaz liner has demonstrated reliable service in numerous liquified natural gas tank installations. However, if a leak should develop in the liner of a nitrate salt tank, it is estimated that the repair procedure would be more lengthy than for an externally insulated tank for two reasons. First, the larger thermal mass of the internally insulated design will extend the cool down period of the tank. A first order analysis was based on the following:

- The weight of the PDM tank was approximately 313,000 kg (690,000 lb). Assuming a stainless steel specific heat of 460 J/kg-C (0.11 Btu/lb_m-F) and a temperature change of 556 C (1,000 F), the thermal mass of the tank was on the order of 22 MWht (76 million Btu). As noted above, the cool down period was 1 to 2 days

- The weight of the Technigaz tank and refractory were 446,000 kg (984,000 lb) and 1,700,000 kg (3,750,000), respectively. Assuming a carbon steel specific heat of 460 J/kg-C (0.11 Btu/lb_m-F), a tank temperature change of 280 C (500 F), a refractory specific heat of 920 J/kg-C (0.22 Btu/lb_m-F), and a refractory temperature change of 445 C (800 F), the combined thermal mass of the carbon steel tank and refractory was 210 MWht (710 million Btu). Assuming that the cool down period is proportional to the thermal mass, it may take 10 to 20 days following the detection of a leak before repair procedures could be started.

Second, the extent to which the refractory was contaminated with salt would need to be determined and those bricks which had absorbed salt would need to be replaced. The replacement time would depend on the number of contaminated bricks, but it is clear that the leak repair procedure would be more time consuming than for the externally insulated designs. Thus, it appears that the time required to cool the tank, locate the leak, replace the refractory, and fill the tank could be in the range of 15 to 30 days.

From this simple analysis, the frequency of leaks in an internally insulated tank can be only one-half to one-third of that in the externally insulated design without suffering a disadvantage in annual availability.

ADVANTAGES AND DISADVANTAGES

Some of the qualitative advantages and disadvantages of each design concept are summarized below in Table 4-4.

Table 4-4
STORAGE TANK CONCEPT ADVANTAGES AND DISADVANTAGES

Item	External Insulation	Internal Insulation
Design		
- Structural analysis	More complex fatigue analysis of wall-to-floor joint	Less demanding fatigue analysis
- Thermal analysis	Less complex	More complex, particularly for potential thermal short circuits to carbon steel shell
Fabrication	Less field manhours, but specific weld procedures required for thick stainless steel sections	Significant field manhours and detailed liner weld quality assurance procedures
Operation	Temperature ramp rates must be monitored to ensure fatigue life is met; periodic use of heat tracing is acceptable due to lower thermal losses	Rapid thermal transients can be accommodated; 2.5 times higher thermal losses
Leak detection	Passive detection methods suitable to identify shell and bottom leaks	Active detection methods required to identify liner leaks as quickly as possible
Leak repair	Drain tank, locate by vacuum box, repair leak, and test by vacuum box	Drain tank, locate by ammonia leak test, repair or replace defective liner section, replace contaminated refractory, and test new welds with ammonia
Leak repair time	5 to 9 days	15 to 30 days

WARRANTY PROVISIONS

The principal provisions in the warranty offered by CBI include the following:

- Any defects caused by faulty design, workmanship, or material furnished by CBI will be repaired for a period of one year from the date of completion

- The guarantee is valid only if a complete and continuous temperature and level history of the tank is maintained

- Any warranty of fitness for a particular purpose or compensation for consequential damages are expressly excluded.

Assuming that PDM would design, fabricate, and install the tank on a turn-key basis, PDM would expect to offer its standard commercial warranty as follows:

- Any defects caused by faulty design, workmanship, or material furnished by PDM will be repaired for a period of one year from the date of completion

- Any warranty of fitness for a particular purpose or compensation for consequential damages are expressly excluded.

The scope of work for Technigaz on this study included only the development of material quantities for the tank and a cost estimate for the liner and its installation. Bechtel developed the estimate for procurement and installation of the refractory bricks, carbon steel shell, foundation, and insulation. As such, Technigaz was not in a position to offer a warranty on the complete hot tank. However, Technigaz anticipates that the liner will have a service life of 30 years without leaks.

CAPITAL COST ESTIMATES

The capital cost estimate for each tank consist of two elements: the investment cost for design, procurement, and installation, and the operating cost of reduced plant output due to thermal losses through the insulation. The later element can be converted to an equivalent capital cost to give an overall assessment of the three designs.

Design, Procurement, and Installation Cost Estimates

Design, procurement, and installation cost estimates for the externally insulated tanks are summarized in Table 4-5. The CBI and PDM estimates of foundation and tank costs are very close; only the insulation costs differ by a significant amount. As shown in Table 4-2, CBI selected an insulation thickness of 50 cm (20 in.) while PDM selected 40 cm (16 in.). The differences in the selected optimums can likely be traced to differences in the unit insulation costs assumed by CBI and PDM.

Table 4-5
COMPARISON OF HOT SALT TANK COST ESTIMATES

	Chicago Bridge and Iron	Pitt-Des Moines	Technigaz
DESIGN, PROCUREMENT, AND INSTALLATION COST			
- Foundation	$450,000	$280,000	$470,000
- Foundation cooling system	Included with foundation	Included with foundation	$400,000
- Tank	$2,750,000	$2,840,000	$950,000
- Liner	Not required	Not required	$4,690,000
- Heat tracing	Not included	Included	$260,000
- Insulation			
Internal	Not required	Not required	$1,100,000
External	$500,000	$1,360,000	$580,000
- Leak detection system	Not included	Integral with foundation	$150,000
- Sales tax (7.5 percent)	Included	$210,000	$210,000
- Engineering	Included	$320,000	$620,000
- Contingency	Included	Included	$940,000
Total	$3,700,000	$5,010,000	$10,370,000
OPERATING COST			
1) Thermal loss, kWt			
Roof	73.8	95.1	65.5
Wall	129.0	155.4	311.5
Floor	41.7	70.0	344.3
Total	244.4	320.6	721.3
Annual thermal loss cost (1)	$44,000	$57,000	$129,000
Equivalent capital cost (2)	$420,000	$540,000	$1,230,000
2) Foundation cooling pump annual electric demand, kWhe (3)	Not applicable	Not applicable	70,000
Equivalent capital cost (4)	Not applicable	Not applicable	$73,000
TOTAL DESIGN, PROCUREMENT, INSTALLATION, AND OPERATING COST	$4,120,000	$5,550,000	$11,670,000

1) Based on thermal energy cost of $0.0204/kWht and annual operating time of 8760 hours
2) Based on levelized capital carrying charge (fixed charge rate) of 10.5 percent
3) Based on pump demand of 8.0 kWe and annual operating time of 8760 hours
4) Based on electric energy cost of $0.11/kWhe and levelized capital carrying charge (fixed charge rate) of 10.5 percent

It should be noted that the CBI estimate does not include heat tracing or a leak detection system, while the PDM estimate includes these items. Thus, the difference in estimates will be somewhat less than shown in the table. Nonetheless, it is encouraging that the two cost estimates, and warranty provisions, are as comparable as they are for this implies that storage tanks for the early commercial plants should be available on a competitive basis.

As noted above, Technigaz developed material quantities for the thermal storage tank, but cost estimates only for the liner and installation. Bechtel was responsible for developing the estimate for the procurement and installation of the complete tank. An estimate summary is shown in Table 4-5, and the details of the estimate are presented in Table 4-6. The basis for the estimate included the following:

- All costs were first quarter 1993 dollars

- Equipment and bulk material prices were based on recent Bechtel construction experience and vendor catalog prices

- Labor costs were based on Barstow, California craft wage rates and labor productivity. The wage rates included fringe benefits, taxes, insurance, and a casual overtime allowance of 5 percent. Sufficient labor was assumed to be available in the immediate area, and therefore, no allowance for travel and subsistence was provided

- Distributable labor and material costs were estimated to be 80 percent of direct labor costs. These costs included the following:

    - Temporary construction building, utility systems, and scaffolding
    - Construction equipment, small tools, equipment maintenance, material handling, consumable supplies, and purchased utilities
    - Crane, earth mover, and truck rentals
    - Field staff providing craft supervision, personnel activities, and warehousing

- In the PDM estimate, engineering costs were approximately 7 percent of the sum of the material and labor costs. For the purposes of this study, engineering costs for the Technigaz design were also estimated to be 7 percent of the sum of the material and labor costs

- The conceptual tank designs outlined in the vendor reports did not include all of the detail which would be available at the completion of final design. To account for items in the cost estimate which were not yet identified, a contingency was added. It was assumed that these contingencies were included in the CBI and PDM estimates. It was further assumed that the level of definition in the Technigaz design is reasonably complete, and that a contingency of 15 percent was sufficient to account for all material and labor costs which have yet to be identified.

Table 4–6
TECHNIGAZ HOT SALT TANK COST ESTIMATE DETAILS

	DESCRIPTION	QTY	UNIT	UNIT COST MAT'L	UNIT COST S/C	UNIT COST LABOR	UNIT MHR	TOTAL MHR	$ MAT'L	$ LABOR	$ S/C	$ TOTAL
1.0	FOUNDATION											
	Excavation	500	YD3			40	0.5	250		18,000		18,000
	Fine grade	13,300	FT2			40	0.03	400		28,800		28,800
	Formwork	800	FT2	1		40	0.5	400	800	28,800		29,600
	Concrete	840	YD3	60		40	1	840	50,400	60,480		110,880
	Reinforcing steel	204	T	600		40	10	2,040	122,400	146,880		269,280
	Embedded metal	750	LB	1.75		40	0.07	53	1,313	3,780		5,093
	Compacted backfill	80	YD3			40	1	80		5,760		5,760
								4,063	174,913	292,500		467,413
2.0	FOUNDATION COOLING SYSTEM											
	Cooling water pipe (4 in., Sch 40, carbon steel)	5,000	FT	13		40	0.30	1,500	64,740	108,000		172,740
	Pipe welds (20 ft lengths)	260	EA			40	1.90	494		35,568		35,568
	Valves – 4 in. gate	2	EA	2,500					5,000			5,000
	– 4 in. check	1	EA	2,000					2,000			2,000
	Pipe supports – 4 in.	12	EA	135					1,620			1,620
	Miscellaneous materials (10 percent) and labor operations (80 percent)	1	LT			40	2871	2,871	7,336	114,854		122,190
	Instrumentation (25 percent of installed pipe cost)	1	LT								30,548	30,548
	Pipe trench excavation and backfill (1,000 ft)	110	YD3		5						550	550
	Cooling water pump (40 gpm, 700 ft tdh, 10 bhp)	1	EA	1,500		40	25	25	1,500	1,800		3,300
	Concrete foundation for cooling water pump	1	YD3		250						250	250
	Cooling tower (wet, mechancial draft; 1,200,000 Btu/hr)	1	EA	15,000		40	100	100	15,000	7,200		22,200
	Concrete foundation for cooling tower	5	YD3		250						1,250	1,250
								4,990	97,196	267,422	32,598	397,216
3.0	TANK STRUCTURE											
	Walls, floor, and roof (A516 Gr. 70 carbon steel)	463	T		1,700						787,100	787,100
	Suspended ceiling (316 stainless steel)	29	T	2,500	3,000				72,500		87,000	159,500
									72,500		874,100	946,600
4.0	INTERNAL INSULATION											
	Refractory bricks (9 in. x 4 1/2 in. x 2 1/2 in.)	512,000	EA		2.14						1,095,000	1,095,000
5.0	CORRUGATED LINER											
	Incoloy 800 liner, 1.27 mm thick (Includes wall and bottom areas, angle pieces, angle corners, central piece, bottom caps, flat caps, dog legs, special expansion bellows between wall and suspended deck, anchor pieces, and	1	LT	1,990,000		40	33,600	33,600	1,990,000	2,419,200		4,409,200

Table 4-6 (Continued)

TECHNIGAZ HOT SALT TANK COST ESTIMATE DETAILS

	DESCRIPTION	QTY	UNIT	---- UNIT COST ---- MAT'L	S/C	LABOR	UNIT MHR	TOTAL MHR	---------- $ ---------- MAT'L	LABOR	S/C	TOTAL
6.0	EXTERNAL INSULATION											
	Walls – mineral wool (2 in. thick)	15,900	FT2		17						270,300	270,300
	Roof – internal mineral wool (20 in. thick)	10,300	FT2		2						20,600	20,600
	– external mineral wool (6 in. thick)	10,400	FT2		28						291,200	291,200
	Aluminum jacket	26,300	FT2	(Included with insulation cost)								
											-----	-----
											582,100	582,100
7.0	HEAT TRACING											
	Mineral insulated resistance cable (150 W/ft)	9,600	FT2	13		36	0.2	1,920	124,800	124,416		249,216
	Termination assemblies	28	EA	53		36	1	28	1,484	1,814		3,298
	Combination thermostat/contactor/junction box	2	EA	1,050		36	5	10	2,100	648		2,748
	Mounting brackets	2	EA	55		36	2	4	110	259		369
	Star connection junction box	2	EA	40		36	5	10	80	648		728
	Mounting brackets	2	EA	55		36	2	4	110	259		369
	Seals	28	EA	0.28		36	0.02	1	8	36		44
								----	-----	----		-----
								1,977	128,692	128,081		256,773
8.0	LEAK DETECTION SYSTEM											
	Gas monitoring piping (1/2 in. tubing)	2,000	FT	15		40	0.65	1,300	30,000	93,600		123,600
	Solenoid valves (1/2 in.)	12	EA	300		40	4	48	3,600	3,456		7,056
	Gas chromatograph	1	EA	8,000		40	80	80	8,000	5,760		13,760
	Computer (80386 with monitor and keyboard)	1	EA	1,500		40	40	40	1,500	2,880		4,380
	Data storage (300 Megabyte external hard disk drive)	1	EA	600		40	20	20	600	1,440		2,040
								----	-----	----		-----
								1,488	43,700	107,136		150,836
								----	-----	-----	-----	-----
	SUBTOTAL							46,117	2,507,000	3,214,339	2,583,798	8,305,137
	Incology liner ocean shipping, import duty, and inland freight								285,000			
	Sales tax (7.5 percent)								209,400			
									-----	-----	-----	-----
	TOTAL								3,001,400	3,214,339	2,583,798	8,799,537

Note: Distributable costs are estimated to be 80 percent of direct labor costs

As expected, the carbon steel vessel in the Technigaz concept was considerably less expensive than the stainless steel vessels required in the CBI and PDM designs. However, in essentially all other categories, the internally insulated design was more expensive. The principal reason for this is the extensive field labor required to install the Incoloy liner and the refractory bricks. Note that the installed cost of just the liner was approximately the same as the complete tank estimates from CBI and PDM. The Technigaz concept was also burdened with an active foundation cooling system and leak detection system that the other two concepts did not require.

Operating Cost Estimates

Operating costs included the economic penalty for heat loss through the tank insulation plus, for the Technigaz concept, the penalty for electric energy use in the foundation cooling system. The thermal losses from each tank, shown in Table 4-5, can be converted to an equivalent capital cost using the following equation:

$$\text{Equivalent Capital Cost} = \frac{\text{Thermal loss} * \text{Marginal thermal energy cost} * \text{Annual operating time}}{\text{Levelized Capital Carrying Charge}}$$

where:

- The marginal cost of the collector and receiver system to supply 1 kWh of thermal energy was estimated to be \$0.020. This was based on a unit heliostat price of \$175/m^2 and a unit receiver system price of \$115/kWt

- The annual operating time of the tank was assumed to be 8,760 hours

- The levelized capital carrying charge (fixed charge rate) was 10.5 percent. The rate, based on standard utility project financing and a constant year dollar analysis, was that used during Phase 1 of the central receiver Utility Studies.

A similar analysis was used to convert the annual electric energy demand of the Technigaz foundation cooling water pump to an equivalent capital cost. The pump power demand was a continuous 8 kWe. The value of electric energy was assumed to be the levelized energy cost for the first commercial plant in Phase 1 of the Utility Studies, or \$0.11/kWhe.

The results of the calculations are shown in Table 4-5. For each tank, the equivalent capital cost for operation was equal to approximately 10 percent of the design, procurement, and installation cost. Note that the higher heat losses through the Technigaz insulation entailed an economic penalty of approximately \$750,000 relative to the CBI and PDM approaches.

Design, Procurement, Installation, and Operating Cost Estimates

The sum of the design, fabrication, installation, and operating cost estimates are also shown in Table 4-5. From a review of the estimates, the following observations can be made:

- The internally insulated tank is approximately twice as expensive as the externally insulated designs
- There is good agreement on the cost estimates from two of the vendors who are potential suppliers to the Solar Two and early commercial projects
- The subcontract price developed by CBI for the externally insulated hot salt tank and foundation in Phase 1 of the Utility Studies was $3,300,000 (third quarter 1987 dollars). Escalating this price to first quarter 1993 dollars using an annual rate of 4 percent yields an estimate of $4,100,000. This price compares very favorably with the average of the CBI and PDM estimates in this study ($4,350,000)
- It appears that a hot salt tank for a 100 MWe commercial project can be fabricated and installed for approximately $5 million.

Section 5
References

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1-3 "Status of Solar-Thermal Electric Technology", H. G. H. Enterprises, Laguna Hills, California, Electric Power Research Institute Report GS-6573, Project 2003-9, December 1989

2-1 "Molten Salt Steam Generator Subsystem Research Experiment, Phase 1 Final Report, Volume 1", Foster Wheeler Solar Development Corporation, Livingston, New Jersey, FWSDC Contract 9-71-9202, Sandia National Laboratories Report 82-8179, October 1984

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Appendix A

Statement of Work for Steam Generator Vendors

Statement of Work
Steam Generator Design and Cost Estimate

PURPOSE

The purpose of this study is to resolve issues related to the design, fabrication, warranty, and capital cost of steam generator systems for commercial nitrate salt central receiver plants. The next central receiver project will be the Solar Two project, a retrofit of the 10 MWe Solar One pilot plant with nitrate salt receiver, thermal storage, and steam generation systems. In addition, it is likely that the Solar Two project will be the only predecessor to the first 100 MWe commercial project. Therefore, the equipment installed at Solar Two should be as representative as possible of the equipment to be installed in the first commercial project. To select the best design for the Solar Two project, an optimum design must be defined for the first 100 MWe project. This study will review and compare the alternate steam generator designs for the first 100 MWe project, and evaluate these designs according to their feasibility, capital cost, performance, warranty terms, and operation and maintenance requirements.

BACKGROUND

The central receiver Utility Studies completed in 1988 proposed a baseline design for all the major systems in the first commercial 100 MWe plant. The steam generator design, developed by Babcock & Wilcox, was a forced recirculation drum type with separate shells for the superheater, reheater, evaporator, and preheater. The heat exchangers used a U-tube/U-shell design, which is highly tolerant of thermal stresses due to transients but is also rather expensive. Other steam generator designs have been proposed which may be suitable and less expensive, but they have not been investigated in the same level of detail. These include the following:

- Natural circulation drum type, with straight tube/straight shell superheater, reheater, evaporator, and preheater components using bellows for thermal expansion. This concept was developed by Foster Wheeler in the early 1980's.

- Kettle evaporator with U-tube/straight shell superheater, reheater, and preheater components. This design is similar to that currently employed by Luz in the SEGS VIII and IX power plants.

In this study, Foster Wheeler will investigate the straight tube/straight shell design, and ABB Lummus and Struthers Wells the kettle evaporator concept.

STEAM GENERATOR SCOPE OF SUPPLY

The steam generator transfers the thermal energy in nitrate salt to thermal energy in main and reheat steam for use in a turbine-generator. The steam generator includes the following items:

- Nitrate salt-to-water and nitrate salt-to-steam heat exchangers

- Steam drum, if required

- Nitrate salt and steam attemperators, as required
- Inter-heat exchanger piping
- Water recirculation pumps, if required
- Electric heat tracing and insulation
- Heat exchanger and piping supports
- Controls and instrumentation.

Performance specifications for the steam generator are summarized in Table A-1. In sizing the heat exchangers, consideration shall be given to optimizing the heat transfer area and salt side pressure drop. For this study, the value (equivalent capital cost) of reducing the pressure drop on the salt side by 1 ft of head is estimated to be $9,000.

STATEMENT OF WORK

The vendor shall review and update the existing steam generator design, describe the advantages and disadvantages of the heat exchanger configuration, and provide an updated capital cost estimate. Specific items to be addressed include the following.

- Heat exchanger arrangement drawings, and section drawings which are representative of the components
- Heat exchanger specifications, including
  - materials
  - heat transfer areas
  - tube and shell side heat transfer coefficients
  - weights of the shell and internals
- Requirements for salt temperature attemperation at the inlet to the superheater, reheater, or evaporator
- Design, fabrication, and delivery schedule
- Estimated start times from cold, warm, and hot conditions
- Overnight thermal conditioning requirements and the ability to respond to daily temperature transients
- Warranty provisions

Table A-1
STEAM GENERATOR PERFORMANCE SPECIFICATION

Nominal Ratings	110 MWe gross plant output 260 MWt steam generator duty
Final Feedwater	236 C (456 F) (As required) MPa (psia) 93.71 kg/sec (743,700 lb/hr); 1% blowdown assumed
Main Steam	540 C (1,004 F) 13.03 MPa (1,890 psia) 92.77 kg/sec (736,300 lb/hr)
Cold Reheat Steam	347 C (656 F) 3.08 MPa (446 psia) 79.92 kg/sec (634,300 lb/hr)
Hot Reheat Steam	538 C (1,000 F) 2.77 MPa (402 psia) 79.92 kg/sec (634,300 lb/hr)
Nitrate Salt	566 C (1,050 F) inlet temperature (As required) MPa (psia) inlet pressure 454 C (850 F) maximum evaporator tube temperature consistent with acceptable corrosion rates for chrome-moly tubes 288 C (550 F) outlet temperature 138 kPa (20 psia) outlet pressure Specific heat $0.345 + (2.28 \times 10^{-5})(\text{Temp, F})$, Btu/$lb_m$-F Density $131.2 - (2.221 \times 10^{-2})(\text{Temp, F})$, lb_m/ft^3 Thermal conductivity $0.25308 + (6.26984 \times 10^{-5})(\text{Temp, F})$, Btu/hr-ft-F Viscosity $60.2844 - (0.17236)(\text{Temp, F}) + (1.76176 \times 10^{-4})(\text{Temp, F})^2 - (6.11408 \times 10^{-8})(\text{Temp, F})^3$, lb_m/ft-hr

- Cost breakdown in sufficient detail to understand how the costs were developed and to permit a comparison with costs from the other vendors. The breakdown should include the following items:

    - Engineering and procurement
    - Material costs for each heat exchanger and the steam drum (if required)
    - Fabrication costs for each heat exchanger and the steam drum (if required)
    - Installation
    - Heat tracing and insulation
    - Controls and instrumentation.

Appendix B

Statement of Work for Thermal Storage System Hot Salt Tank Vendors

Statement of Work
Hot Salt Tank Design Cost Estimate

PURPOSE

The purpose of this study is to resolve issues related to the design, fabrication, warranty, and capital cost of thermal storage systems for commercial nitrate salt central receiver solar power plants. The next central receiver project will be the Solar Two project; a retrofit of the 10 MWe Solar One pilot plant with nitrate salt receiver, thermal storage, and steam generation systems. In addition, it is likely that the Solar Two project will be the only predecessor to the first 100 MWe commercial project, and therefore, the equipment installed at Solar Two should be as representative as possible of the equipment installed in the first commercial project. To select the best design for the Solar Two project, an optimal design must be defined for the first 100 MWe project. This study will review and compare the alternate thermal storage tank designs for the first 100 MWe project, and evaluate the designs according to their feasibility, capital cost, and warranty terms.

BACKGROUND

The central receiver Utility Studies completed in 1988 proposed a baseline design for all of the major systems in the first commercial 100 MWe plant. An externally insulated carbon steel tank was used to store the 550°F cold salt and an externally insulated stainless steel tank was used to store the 1050°F hot salt. Designs and costs for these tanks were provided by CBI Industries and by Pitt-Des Moines.

Earlier studies of Martin Marietta adopted an alternate hot salt tank design based on use of an internally insulated carbon steel tank with an inner waffle-configured stainless steel liner developed by Technigaz. A 7 MWh 1050°F salt tank based on this concept was installed and successfully tested at the Sandia Central Receiver Test Facility at Albuquerque in 1982.

This study is intended to assess the relative feasibility, warranty availability, and capital cost for these two hot salt tank design approaches. Bechtel will compare and evaluate information supplied by Pitt-Des Moines, CBI Industries and Technigaz to determine which designs are suitable for 1050°F service in a 100 MWe central receiver solar power plant.

HOT SALT THERMAL STORAGE TANK SCOPE OF SUPPLY

The hot salt thermal storage tank stores heated salt from the solar receiver until it is pumped to the steam generator for subsequent conversion to electric energy. The thermal storage tank includes the following items:

- A stainless steel or carbon steel tank

- An exterior or interior insulation system

- A stainless steel liner (only for interior insulation system)

- A cooled foundation

- A leak detection system (not a part of Task 4 cost estimate)

- An electrical heat tracing system capable of preheating the empty tank prior to initial charging of tank with salt (not a part of Task 4 cost estimate).

The specification for the hot salt thermal storage tank is presented in Table B-1. It is based on the tank specification from the Utility Studies, with the requirement for a leak detection system added.

STATEMENT OF WORK

Task 1 Design Review and Update

The vendor shall review their previous tank designs that were prepared for the Utility Studies (PDM and CBI) or for the thermal energy storage tank design reported in "Molten Salt Thermal Energy Storage Subsystem Research Experiment", MCR-82-1722, September 1982 (Technigaz). The latter design shall be scaled as needed to satisfy the capacity and other requirements of the Table B-1 specification. Each vendor can make modifications as may be necessary to bring the design in step with the current technology status.

<u>Deliverables</u> Description of updated tank design including a discussion of prominent tank design features and associated advantages and disadvantages with illustrations and/or drawings.

Task 2 Inputs for Tank Design Comparisons

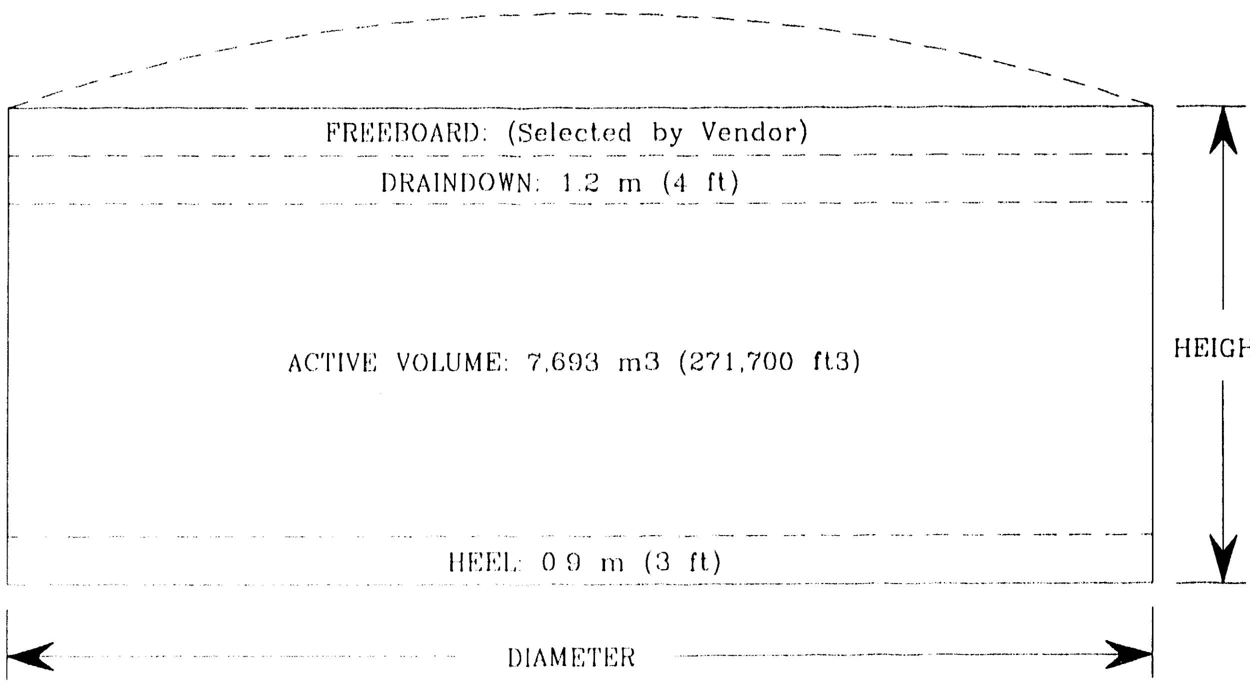
Criteria for the tank design comparisons to be made by Bechtel are indicated below. The vendor shall prepare written discussions of their design covering each of the comparison criteria listed below. Note that two of the criteria are treated in Tasks 3 and 4 and need not be discussed under Task 2.

- Capital cost (discussed under Task 4)
- Fabrication quality assurance
- Warranty provisions
- Accommodation of thermal expansion and heat tracing
- Estimated rate of heat loss
- Foundation design concept
- Inventory charging and inventory/temperature cycling
- Leak detection, location and repair (discussed under Task 3)
- Major maintenance repair anticipated during a 30-year service life.
- Areas of design uncertainty and recommended resolution

<u>Deliverables</u> Written discussions of comparison criteria

Table B-1
HOT SALT TANK SPECIFICATION

Active Tank Volume	7,693 m^3 (271,674 ft^3) (1,560 MWh in 550 F to 1,050 F salt)
Additional Tank Volume	See Figure B-1
Tank Heel	0.914 m (3 ft)
Draindown Volume	1.219 m (4 ft) To allow for draining of receiver inventory and for emergency storage of cold tank heel
Tank Freeboard	(To be selected by supplier)
Equivalent Capital Cost per kWt of Heat Loss (for use in calculating optimal insulation thickness)	$1700/kWt
Insulation Shield Temperature	140°F Maximum
Heat Tracing System	Electric heating elements to be utilized at 50% of rating; redundant circuits are required
Bottom Leak Detection System	(Vendor concept and design)
Cooled Foundation	(Vendor concept and design)
Seismic	API 650, Zone 3
Wind	90 mph @ 10 m above grade
Allowable Soil Bearing Capacity	5,000 psf @ 5 ft below grade 10,000 psf @ 10 ft below grade
Density of Salt	131.2 - 0.02221 * (Temp, °F), lb_m/ft^3



TANK VOLUME = ACTIVE VOLUME + HEEL + DRAINDOWN + FREEBOARD

Figure B-1 Thermal Storage Tank Nomenclature

Task 3 Repair Rationale

We assume that a tank leak will occur at least once during the 30 year life of the plant. The vendor shall provide a discussion of the rationale for leak detection, location and repair. Items to be addressed shall include:

- Recommended leak detection rationales and equipment
- Methods for locating a tank bottom leak and baseline estimate of the required time
- Tank bottom leak repair procedure and baseline estimate of required completion time
- Representative range of labor hours to locate and repair a tank bottom leak (with uncertainties duly noted).

Deliverables Written repair rationale

Task 4 Cost Estimates

The vendor shall prepare an estimate of installed cost of the hot salt tank, covering the entire scope of supply indicated on pages 1 and 2 above except as noted below. Estimated costs shall identify engineering and procurement, tank materials, insulation materials, foundation materials and field fabrication costs in sufficient detail to permit a comparison with cost from other vendors. Heat tracing system and leak detection system costs are not required. Site location is assumed to be Barstow, California. Uncertainties associated with selected elements of the cost estimate should be duly noted.

Estimates in foreign currency should include an approximate estimate of that portion of the materials and of the labor that may become available from United States sources.

Deliverables Cost estimates

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