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LINE-FOCUS SOLAR CENTRAL POWER SYSTEM—PHASE I

Final Report, Volume 2: Text

By
Arthur J. Slemmons

April 1980

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LINE-FOCUS SOLAR CENTRAL POWER SYSTEM— PHASE I

Final Report

Volume II: Text

April 1980

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**FRONTISPIECE THE FOUR GENERIC TYPES OF SOLAR THERMAL POWER SYSTEMS
BEING STUDIED BY THE U.S. DEPARTMENT OF ENERGY**

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PREFACE

This report is submitted by SRI International to the Department of Energy, under Contract EY-76-C-03-0115, as the final document for the Phase I study of the Line-Focus Central Power System. It contains a summary of the system analysis; parametric analysis, selection and conceptual design of the optimized system, and determination of the commercial market for solar line-focus central power plants.

The report is submitted in three volumes, as follows:

Volume I Executive Overview

Volume II Conceptual Design of the Line-Focus Central Power System

Volume III Appendices

Specific tasks performed by members of the SRI International team were as follows:

- SRI International
 - System analysis
 - System optimization
 - System configuration selection
 - Master control subsystem analysis and selection
 - Receiver subsystem experiments
 - Cost and performance analysis
 - Commercial assessment
- Bechtel National Inc.--Analysis, optimization, selection, and conceptual design of:
 - Heat transport subsystem
 - Energy storage subsystem
 - Electrical power generating subsystem

- Receiver towers
- Heliostat foundations
- Foster Wheeler Development Corporation--Analysis, optimization, selection, and conceptual design of the receiver subsystem
- Acurex/Aerotherm Corporation--Analysis, optimization, selection, and conceptual design of the collector subsystem
- Pacific Gas and Electric Company--Consultant to SRI International

SUMMARY

The Department of Energy, through its Division of Solar Technology, is engaged in an effort to develop the technology for the practical and economic conversion of sunlight into electricity. The Solar Thermal Power Program presently being conducted by the Division of Solar Technology includes several contracts with industrial teams for the conceptual design, development, testing, and demonstration of a variety of Solar Thermal Power System concepts and applications. These efforts are being made in two areas: central power applications and dispersed power applications. This Phase I contract falls into the central power applications area.

As prime contractor, SRI analyzed the Line-Focus Central Power System, the complement to the other three generic solar thermal power systems being studied by Department of Energy contractors (see Frontispiece). The SRI team included Bechtel National, Incorporated; the Foster Wheeler Development Corporation; and the Acurex/Aerotherm Corporation. The Pacific Gas and Electric Company acted as consultant to SRI. The team has completed a subsystem and system parametric analysis, a 100-MW_e commercial plant conceptual design, a cost and performance analysis, and a commercial assessment.

The high-temperature line-focus (HTLF) central power system, when applied to intermediate load service, can operate in an economically competitive range in the year 2000 for most of the western and a very small fraction of the eastern United States, if the cost of coal escalates at an average rate of 8% per year (3% above the assumed general inflation). Depending on whether the initial coal price is taken to be \$1.00 or \$1.40/MMBtu, it is estimated that a potential total of from 160 to over 360 units could be installed by the year 2010. These totals will be limited mainly by the reluctance of utilities to commit large fractions of their installed capacity to a solar insolation dependent

source. At a lower coal price escalation rate (6%), the HTLF would be likely to capture only a very small share of the intermediate-load market. It is estimated that several units, located in the most favorable locations, would be an expected level of penetration by 2010. Significant penetration of eastern United States markets, which are characterized by relatively low solar insolation levels, would require extremely high coal price escalation rates of more than 12% per year (7% per year greater than the assumed general inflation rate).

SRI studies indicate that if production or cost is not restrained by U.S. policy, coal prices of 6 or 8% are considered likely. SRI has concentrated on assessing the markets for intermediate-load units, based in the western United States. The assessment results indicate a good potential for application at the higher (8%) coal price escalation rate. The following table provides a summary of the potential line-focus central power plant market penetration for the western United States, with coal prices of \$1.00 and \$1.40/MMBtu.

SUMMARY OF LINE-FOCUS CENTRAL RECEIVER SOLAR THERMAL POWER PLANT
MARKET PENETRATION IN WESTERN U.S. MARKETS
(In Cumulative Numbers of 100 MW Units)

<u>Year</u>	<u>\$1.00/MMBtu Coal Price</u>		<u>\$1.40/MMBtu Coal Price</u>	
	<u>Coal Price Escalation Rate</u>		<u>Coal Price Escalation Rate</u>	
	<u>6%/yr</u>	<u>8%/yr</u>	<u>6%/yr</u>	<u>8%/yr</u>
1990	0	0	0	0
1995	0	2	0	12
2000	0	24	0	92
2010	0	162	2-4	362

1.0 INTRODUCTION AND SUMMARY

1.1 Introduction

The Department of Energy, through its Division of Solar Technology, is developing the technology for the practical and economic conversion of sunlight into electricity. The solar thermal power program, presently being conducted by the Division of Solar Technology, includes several contracts with industrial teams for the conceptual design, development, testing, and demonstration of a variety of solar thermal power system concepts and applications. Efforts have been expended in two areas: central power applications and dispersed power applications. This contract falls into the central power applications area.

The Phase I conceptual design of the line-focus solar thermal central power system was studied under a prime contract to SRI International (SRI), Menlo Park, California. Subcontractors to SRI included: Bechtel National, Inc., San Francisco, California; Foster Wheeler Development Corporation, Livingston, New Jersey; and Acurex/Aerotherm, Sunnyvale, California. Pacific Gas and Electric Company of San Francisco was consultant to SRI.

The project started 1 October 1978 and the completion date was 30 April 1980. The DOE Headquarters Program Manager was Mr. Joseph Weisiger, Jr. The contract was awarded by DOE/SAN, with Mr. Larry Prince acting as project officer. The technical effort was monitored by Mr. John Coggi of the Aerospace Corporation.

The general objective of the study was to make a systematic investigation of the concept, and provide data that would allow it to be compared to the first generation point-focus (power tower) central power system. The specific objectives of the study were:

- Analyze the line-focus solar thermal central power system
- Perform parametric analyses on the various subsystems
- Optimize both the subsystems and the system
- Provide a conceptual design of the optimized system
- Minimize busbar energy cost ($\overline{\text{BBEC}}$)
- Determine the commercial market for line-focus central power plants.

Every attempt was made to use conservative performance and cost figures. For instance, a mirror reflectivity of 0.9 was used, whereas higher values were probably justified. A conservative receiver absorptivity (α_s) of 0.88 was used, and a conservative receiver emissivity ($\epsilon_{525^\circ\text{C}}$) of 0.06 was used. Better values have been reported in the literature, and the relevant technology will probably be available for commercial use in the 1980s. Conservative tube panels for the receiver were used, although lower cost panel coils can probably be used. The learning curves of 0.90 used for the first plant, and 0.95 for the 80th plant, are believed to be conservative for a new technology, such as this. The calculated heliostat cost is conservatively high, because expensive glass-reinforced plastic was used for the glass substrate. Other examples of cost and performance conservatism are included in the body of the report.

Several advances in technology have resulted from this study. The most important are:

- Heat storage in the heliostat field
- Reheat steam cycle for line-focus solar thermal electric power
- Glass-reinforced concrete heliostats
- Line-focus cavity receiver with selective surfaces
- Low-cost solar thermal electric power.

1.2 Contract Tasks

The study was divided into 10 tasks to permit a systematic investigation of the concept. The tasks and their relation to each other are shown in Figure 1-1.

FIGURE 1-1 PROJECT WORK FLOW

Task 1: Requirements Definition. The purpose of this task was to provide a line-focus solar central power systems requirements definition of the concept, that would allow it to be compared directly to the first generation point-focus (power tower) central power system.

Task 2: Parametric Analyses. The purpose of this task was to conduct a preliminary system analysis to characterize the plant and identify subsystem operating modes and requirements. A parametric analysis of all subsystems in the proposed concept was conducted to obtain preliminary estimates of the cost, performance, and reliability of a commercial scale, storage-coupled plant (100 MW_e nominal). Development problems were identified, and design alternatives analyzed, to define a line-focus solar central power system that satisfied the requirements definition. The engineering tradeoff studies and analyses necessary for defining and selecting the subsystem were also performed.

Task 3: Commercial System Configuration Selection. The purpose of this task was to perform those system level engineering tradeoff studies and analyses necessary for defining the interactions and possibly competitive requirements of the subsystems. The final line-focus solar central power system configuration was chosen by using selected results from the subsystem parametric analyses of Task 2, and the system level analyses. During the selection, the maximum potential worth of the system to a utility was emphasized. This was done by minimizing the cost (per electrical kilowatt-hour) of the energy produced, and maximizing the savings (in terms of the capital investment and operating costs) of displaced conventional generation. The sensitivity of the selected configuration to changes in ambient temperature, wind speed, startup and shutdown rates, working fluid conditions, and the reference insolation level was also explored.

Task 4: Commercial Plant Conceptual Design. The purpose of this task was to prepare a commercial scale (100 MW_e nominal) plant conceptual design for the line-focus solar central power system concept

selected in Task 3. The conceptual design was more detailed than those developed prior to this contract and permitted a more definitive assessment of the technical and economic feasibility of the concept than was possible in Tasks 2 and 3.

Task 5: Commercial Plant Cost and Performance Estimates. The purpose of this task was to provide preliminary cost and performance estimates for the commercial scale line-focus central power plant conceptual design developed in Task 4.

Task 6: Limited Subsystem and Component Experiments. The purpose of this task was to identify, design, fabricate, and perform limited key subsystem level, or component level, research experiments. These experiments were to complement previous development work on the concept and the parametric analysis performed in the previous tasks. The experiments, although limited in nature, addressed critical performance areas and uncertainties in the design. The performance data resulting from these experiments were used in Task 7 to assess the feasibility of the concept.

Task 7: Assessment of Commercial Scale Line-Focus Central Power System. The purpose of this task was to assess the selected solar configuration. The assessment studies were made in sufficient detail to identify the full potential of the proposed line-focus central power technology and to establish technical and cost objectives. The assessment addressed the following topics:

- Potential future improvements of the line-focus central power concept in the areas of performance and cost.
- Potential limitations of the line-focus central power concept.
- A preliminary market survey and analysis was conducted to identify potential markets, in terms of size and penetration, that take advantage of special features of the line-focus technology.

Task 8: Development Plan. The stages of development required for the assessment, and verification of the technical, economic, and environmental viability of the proposed concept, relative to first generation point-focus solar central receiver concepts were to be identified. However, this task was deleted by DOE-SAN during the tenth month of the study.

Task 9: Phase II Plan and Proposal. That portion of the Task 8 Development Plan dealing with Phase II was to be expanded into a proposal for continuation of the contractor's effort into Phase II. This task was also deleted during the tenth month of the study.

Task 10: Phase I Program Planning and Management. The object of this task was to provide the planning, management, and reporting required to accomplish Phase I of the program.

1.3 Summary

The results of the parametric analysis, conceptual design, and market analysis indicate that the line-focus central power system is a potentially viable solar energy technology. Based on the average daily insolation of $10 \text{ kWh/m}^2/\text{day}$ specified in the RFP, the estimated average annual busbar energy cost (BBEC) is 61 mil/kWh for the 80th plant (at 0.6 average daily capacity factor). The estimated BBEC for the first plant is 67 mil/kWh.

These values compare favorably with coal-fired plants, even at the current low cost of fossil fuels. The analysis was done for the 100 MW_e power rating. The modularity of the concepts allows the conceptual design to be optimized for other scales and for use with other energy sources, such as process heat. The assessment of the market for commercial scale solar thermal electric plants of the line-focus central power plant design in areas having lesser insolation shows a potential market as high as 362 100 MW_e line-focus central power systems, by the year 2010.

2.0 PARAMETRIC ANALYSIS

2.1 General

Parametric analysis (Task 2) was the first of the two major activities of this study. The results of the parametric analysis and configuration selection formed the basis for the second major activity, the conceptual design (Task 4).

Parametric analysis (Task 2) and commercial system configuration selection (Task 3) were parallel activities, as was indicated in Figure 1-1. Although the analyses of Task 2 were at the subsystem level (collector, receiver, energy-storage system, heat-transport system, EPGS, and master control), they were guided by system criteria generated in Task 3. Initial analyses of subsystem design, and operational and interfacing alternatives, together with the associated performance/cost/sensitivity data and development issues, permitted intersubsystem and system-level decisions to be made early in Task 3. The feedback to Task 2 of more tightly defined guidelines directed further subsystem parametric analyses, which were aimed at further defining the remaining subsystem design options. This interactive analysis and selection process was needed to choose, in an expeditious yet methodical way, a preferred system configuration for conceptual design and evaluation in subsequent tasks, while also documenting the critical tradeoffs leading to the selected configuration.

During the parametric analyses, the following subsystem parameters were optimized:

- Heliostats
 - Row number
 - Width
 - Spacing (pitch)
 - Cost

- Drive
- Accuracy
- Field size (capacity factor)
- Length
- Module size
- Foundation type and size
- Materials
- Focus (fixed/variable)
- Receiver
 - Cavity, diameter and aperture size
 - Length
 - Absorber module length
 - Tower spacing
 - Insulation thickness
 - Insulation material
 - Aperture angle to the horizontal
 - Tube size, I.D. and thickness
 - Tube material
 - Tube panel size
 - Tube panel flow path
- Heat Transport Subsystem
 - Pipe sizes, I.D. and thickness
 - Pipe materials
 - Pipe insulation
 - Methods for expansion accommodation
 - Trench construction
- Energy Storage Subsystem
 - Type of system (thermocline vs hot and cold tanks)
 - Tank number
 - Tank size
 - Tank insulation
 - Tank materials
 - Storage temperature

- Electrical Power Generating Subsystem
 - Cycle temperatures
 - Inlet water temperature
 - Throttle pressure
 - Throttle and reheat temperatures
 - Working fluid (salt/steam vs water/steam)
 - Reheat vs non-reheat cycle
- Master Control Subsystem
 - Distributed hierarchy configuration
 - Functions performed at each level in the heirarchy
 - Redundancy requirements
 - Technology for data communication
 - Heliostat elevation measurement
 - Cable installation

The initial work on the parametric analysis was modified from that originally outlined in the proposal to include consideration of a system using $\text{KNO}_3/\text{NaNO}_3$ draw salt as a heat transport and energy storage medium. This tradeoff study, to consider an alternative to the proposed water/steam system, resulted in some additional project effort, which necessitated a moderate reduction in the level of effort of the balance of the parametric analysis. However, the study results more than justified the decision to proceed with the tradeoff study. The salt system proved more economical than the proposed water/steam system, resulting in a considerable reduction in the capital cost of the power plant, and also a reduction in the cost of electricity.

The system originally proposed included a water/steam receiver and heat transport subsystem (HTS) with a Caloria[®] and rock thermocline energy storage subsystem (ESS). The initial selection of these subsystems was based on the work performed by the McDonnell Douglas Astronautics Corporation for central receiver systems (Ref. 2-1). However, after the proposal selection a recent study by Martin Marietta Corporation (Ref. 2-2) indicated that significant improvements in electrical power generating subsystems (EPGS) and the energy storage subsystem (ESS) performance could be realized by using molten draw salt

as both the heat transport and energy storage medium. Consequently, it was decided to perform an initial tradeoff study between the water/steam and salt systems.

Based on the McDonnell Douglas and Martin Marietta designs (Refs. 2-1 and 2-2), preconceptual designs and cost estimates were prepared for two 100-MW_e plants based on a solar multiple of 2.1. The coolant flow rate per unit length of receiver and the corresponding field size were defined by the SRI system performance computer model. Table 2-1 gives a summary of performance for the two 100-MW_e plants.

From a performance standpoint it is apparent that the salt system is superior to the water/steam system. The net cycle efficiency of 40% for the salt system is realized not only for direct operation, but also for operation from storage, because salt at a temperature of 839 K (1050°F) is used as the heat source in both. This higher efficiency is realized primarily as a result of the use of a reheat cycle and a higher throttle steam pressure. Reheat is feasible with the salt system, since the steam generator is located adjacent to the EPGS, thereby minimizing the reheat steam piping needed and the resulting pressure drop. Reheat for the water/steam system would be impractical because of the excessively long line lengths needed to transport the reheat steam to and from the receiver field.

A summary of the direct costs in 1973 dollars, for the EPGS, ESS, and HTS is given in Table 2-2.

The EPGS costs for both systems were similar. The additional cost for the reheat in the salt system turbine was offset by the additional costs associated with dual pressure admission for the water/steam system turbine. However, the ESS for the salt system is considerably less expensive than the ESS for the water/steam system, due to the much smaller capacity of the required tankage. The smaller tankage requirement results from a combination of fluid properties, temperature swings, and system efficiencies.

SALT VS WATER/STEAM TRADEOFF PERFORMANCE SUMMARY

*Mega Pascals gauge

Table 2-2

SALT VS WATER/STEAM TRADEOFF CAPITAL COST SUMMARY
(1978 Dollars)

	\$ (Thousands)	
	Salt	Water/Steam
EPGS	19,000	19,000
ESS	9,500	18,200
HTS	<u>18,200*</u>	<u>13,800</u>
Total	46,700	51,000

* Includes approximately \$2 million for electric heat tracing to prevent solidification of the salt in the pipes.

The HTS cost for the salt system includes piping, heat tracing (to prevent solidification of the salt in the pipes), and the steam generator. The cost for the water/steam system includes piping only, as the heat tracing and a steam generator are not required. Piping costs for the two systems are approximately equal. The salt system piping is somewhat larger in diameter than the corresponding piping in the water/steam system. Also, the Incoloy 800 high-temperature headers in the salt system are more expensive than the low-carbon steel, high-temperature headers of the water/steam system. However, these considerations are offset by the higher pressure requirements of the water/steam system, which result in a thicker pipe wall. The overall cost difference, \$4.4 million less for the water/steam HTS, is due primarily to the electric heat tracing and steam generator required for the salt system.

However, as seen in Table 2-2, the total EPGS, ESS, and HTS costs for the salt system are \$4.3 million less than for the water/steam system. This advantage is in addition to the performance advantages outlined in Table 2-1.

Using the information in Tables 2-1 and 2-2 in conjunction with the associated cost and performance data for the balance of the line-focus plant, it was determined that a significant overall cost advantage exists in favor of the salt heat transport and energy storage system. Based on this finding, the decision was made to change the originally proposed baseline water/steam system to a salt system, and to proceed accordingly with the parametric analysis.

The development of conceptual designs and cost estimates for each subsystem combination considered in the parametric analysis would be expensive and inappropriate for the study requirements. Therefore, an alternative approach was required to determine the performance and cost figures for each system combination. The approach used was to first estimate the results of the parametric analysis in order to select a baseline approach. Second, a reasonably detailed conceptual design and cost estimate was prepared for a baseline plant. Third, performance estimates for the various system combinations were made by modifying portions of the baseline plant design. Fourth, cost estimates for the system combinations were determined by applying exponential cost functions to portions of the baseline plant estimate. The performance and cost estimates for the system combinations were provided as inputs to the overall system computer model. The model determined the combination of system parameters which minimized the busbar cost of electric energy. These selected system parameters were then used in the conceptual plant design.

Operating parameters initially selected for the baseline plant design and cost estimate were:

- 16.5 MPag/783 K/783 K (2400 psig/950°F/950°F) reheat steam cycle
- 506 K (450°F) final feedwater temperature
- 10.1 kPa (3.0 inches HgA) condenser pressure
- 2.10 solar multiple
- 7.0 hours of thermal storage
- 839 K (1050°F) maximum salt temperature

- 20 collector rows per receiver row
- Dispersed location, thermocline energy storage system
- Incoloy 800 hot salt pipe material, and carbon steel cold salt pipe material.

During the parametric analysis, solar multiple changes were studied to determine the effect of the number of hours of thermal storage on the cost of electric energy. A solar multiple range of 1.4 to 2.9 (corresponding to 2.4 to 12 hours of thermal storage) were reviewed. An average solar multiple value of 2.1 was selected for the baseline plant design.

The steam turbine operating conditions, peak salt temperature, and salt pipe material were selected, based on information presented in the Martin Marietta advanced central receiver design study. The location of the energy storage system was based on a preliminary tradeoff between tank costs and salt pump power requirements. A simplified plot plan for the baseline plant design is shown in Figure 2-1.

2.1.1 Parametric Analysis Methodology

Figure 2-2 shows the computer technique for using the results of the subsystem parametric analyses in choosing the most cost-effective combination of subsystem designs. Starting with the assumed minimum capacity (100 MW_e net at two hours after solar noon on winter solstice day), the system computer model calculates the energy directed into the cavity receiver by the heliostat field, per unit length of the system. All optical losses are considered. Also calculated is the amount of energy, per unit length of the system, absorbed by the molten salt in the cavity receiver. Conduction, wind, reflection, and radiation losses are also considered. The losses in the heat transport subsystem and heat storage subsystem are then calculated, on a per-unit-length basis. For a given EPGS efficiency (at the chosen steam inlet temperature) the length, L , of the system needed to produce 100 MW_e net is calculated. Then, for hourly intervals, the computer calculates the energy collected and transformed into net kilowatt hours for winter solstice, summer solstice, and equinox. A yearly kilowatt-per-hour total is then

- ESS THERMOCLINE TANKS
- ▲ MOLTEN SALT HOLDING TANKS
- EPGS AND STEAM GENERATOR
- RECEIVER HEADERS
- - - HOT SALT MAIN HEADER
- COLD SALT MAIN HEADER

FIGURE 2-1 PLOT PLAN BASELINE PLANT DESIGN

FIGURE 2-2 FLOWCHART FOR CALCULATING BUSBAR ENERGY COST (BBEC)

calculated. Using the cost parameters of the various subsystems, estimated on a cost-per-unit-length, (where applicable), and using known fixed and operating costs, the 30-year levelized busbar energy cost, ($\overline{\text{BBEC}}$), is calculated. The sensitivity of $\overline{\text{BBEC}}$ to variations of the subsystem parameters can be studied and optimized on the basis of $\overline{\text{BBEC}}$.

2.1.2 Operating Parameters from the Parametric Analysis

From the parametric analysis, the following parameters were selected for review in Task 3 (commercial system configuration selection).

- Electrical Power Generating Subsystem (EPGS)
 - Throttle steam pressure of 13.8 MPa_g (2000 psig)
 - Throttle and reheat steam temperatures of 783 K (950°F)
 - Final feedwater temperature of 506 K (450°F)
 - Condenser pressure of 10.1 kPa (3.0 inches HgA)
- Energy Storage Subsystem (ESS)
 - Thermocline tank storage system
 - Dispersed location tanks
 - Thermal storage capacity equivalent to 700 MW_e h
- Heat Transport Subsystem (HTS)
 - Hot salt pipe; Incoloy 800 material
 - Cold salt pipe; A-106 Grade B carbon steel
 - Solar multiple of 2.10
 - Single riser and downcomer in each receiver row
 - Preheater pinch-point temperature difference of 11 K (20°F)
- Heliostat Subsystem (HS)
 - Glass-reinforced concrete heliostat [four 3 m x 12.2 m (10 ft x 40 ft) modules]
 - Ball screw/4-bar linkage elevation control

- Augered caisson collector foundations spaced 12.2 m (40 ft) center to center
 - Caisson dimensions of 0.38 m (1.25 ft) diameter and 3.66 m (12 ft)
 - Number of collector rows per receiver row: 20. (This was increased during Task 3 to 24 rows.)
- Receiver Subsystem (RS)
 - 1.8 m (6 ft) diameter cavity with 1.2 m (4 ft) aperture
 - Total receiver length of 17,560 m (57,600 ft)
 - Tower height of 61 m (200 ft)
 - Tower spacing of 45.7 m (150 ft)
 - Each tower foundation requires an area of 15.2 m x 15.2 m (50 ft x 50 ft)
 - Salt inlet and outlet temperatures of 572 K (569°F) and 839 K (1050°F), respectively.
 - Master Control Subsystem (MCS)
 - Central computer, with multiple distributed controllers
 - Controllers interconnected by two-way, low-data-rate serial paths.

2.2 Heliostat Subsystem

2.2.1 Heliostat Subsystem Optics

The basic heliostat subsystem optical computer program was developed by SRI, specifically for the line-focus central receiver system. It calculates the flux density variation during the day on the inside of the linear cavity receivers for the winter and summer solstices, and for the equinoxes. The heliostat computer program has been integrated into the cavity receiver program as a subroutine. The receiver program calls upon the heliostat program to generate the solar fluxes inside the cavity during the day.

Heliostat input variables are:

- Independent variables
 - Time of day and year
 - Latitude
 - Tower height
 - Number of heliostats
 - Position of heliostats
 - Heliostat width
 - Curvature of heliostats (focal length)
 - Mirror reflectivity
 - Atmospheric attenuation
 - Standard deviation for tracking error
 - Standard deviation for surface roughness
- Variables in common with the receiver program
 - Receiver diameter
 - Receiver aperture width
 - Receiver aperture orientation
 - Number of receiver elements (nodes) considered.

The optics problem is formulated in terms of a vector description of rays projected from the sun. These rays are reflected from the mirrors of each heliostat row to intercept an east-west line, representing the center of the aperture of the linear cavity receiver at an

arbitrary height, H , above the ground. The position of each row is defined in terms of an arbitrary row spacing parameter. The slope angle of each heliostat row is determined by the requirement that the projected image be centered on the aperture. Image size is determined by projecting the rays propagated from the limb of the sun.

The focal length, which is different for each row, is a function of the distance of the row from the receiver tower, the height of the tower, and the angle of incidence in the plane normal to the row (receiver) axis. It is, however, independent of the angle of incidence along the row. Thus, for an east-west oriented mirror row, the focus depends not on the east-west position of the sun, but on the height of the sun above the southern horizon. Since this height varies hourly, a minimum image size would theoretically require minor focus adjustment of each heliostat during the day. Projected power densities have been determined for two mirror systems: mirrors having fixed radii of curvature (but different for each row); and mirrors having adjustable focusing. Provision for this cost tradeoff was anticipated during the formulation of the optics computer programs.

Image distortion (e.g., off-axis astigmatism) is formulated in terms of a ray-trace from different elements of arc along the concave mirror surface. To determine the imaged power density at the receiver for each mirror row, image distortion was evaluated, using the ray-trace and convolved with the image due to the finite angle subtended by the sun.

As the image size was determined primarily by the finite solar subtense angle as projected onto the plane normal to the heliostat (receiver) axis, the lateral variation of the image size was calculated during the day, being minimal at solar noon.

The computer plot in Figure 2-3 shows the solar position and mirror orientation as a function of time of day at summer solstice. A field of 30 mirror rows is shown. The angular position of the sun is drawn with respect to the mirror directly below the tower. Solar angles are the same for each row.

FIGURE 2-3 PICTORIAL REPRESENTATION OF RECEIVER/MIRRORS

The computer run in Table 2-3 shows the J^{th} mirror position $[X(J)]$, and its slope angle $[S(J)]$, the magnitude of the reflected ray vector to the tower (RMAG), and image size (SIZIM) for a field of 30 mirror rows. The case shown is for noon at winter solstice. Lengths are measured in meters, angles in radians.

2.2.2 Heliostat Subsystem Parametric Analysis

2.2.2.1 Heliostat Performance

The optics program delivers the incident flux on each cavity receiver element, which is used by the receiver program, and a listing of the losses. Figure 2-3 shows the heliostat optics computer program results for noon on summer solstice for the following parameters:

- Tower Height = 50.0 m
- Mirror width = 3.0 m
- Mirror spacing = 4.0 m
- Receiver radius = 0.5 m
- Number of receiver segments = 24
- Receiver aperture = 180° (horizontal)

Figure 2-3 also shows some aspects of heliostat subsystem performance for a particular design configuration. For example, it shows the cosine and shading factors, and compares the maximum solar energy available with the energy actually incident on the receiver aperture. The rapid dropoff for the outer rows is due to shading and blocking due to a nonoptimized spacing.

For a given design and time, the heliostat subsystem optics program calculates the losses per unit length due to:

- Mirror surface reflection
- Cosine losses
- Blocking and shading
- Atmospheric attenuation
- Cutoff at the receiver aperture.

Table 2-3

EXAMPLE OF COMPUTER OUTPUT

TIME= 12.00

.54	.54	.53
-.84	-.84	-.85
.00		

X(J)	S(J)	RMAG	SIZIM
-60	.1082	84.85	.849
-54	.1345	80.72	.807
-48	.1635	76.84	.768
-42	.1955	73.24	.732
-36	.2307	69.97	.700
-30	.2691	67.08	.671
-24	.3107	64.62	.646
-18	.3552	62.64	.626
-12	.4022	61.19	.612
-6.	.4511	60.30	.603
.	.5009	60.00	.600
6.	.5507	60.30	.603
12.	.5996	61.19	.612
18.	.6466	62.64	.626
24.	.6912	64.62	.646
30.	.7327	67.08	.671
36.	.7711	69.97	.700
42.	.8063	73.24	.732
48.	.8383	76.84	.768
54.	.8673	80.72	.807
60.	.8936	84.85	.849
66.	.9174	89.20	.892
72.	.9389	93.72	.937
78.	.9585	98.41	.984
84.	.9762	103.23	1.032
90.	.9923	108.17	1.082
96.	1.0070	113.21	1.132
102	1.0204	118.34	1.183
108	1.0328	123.55	1.235
114	1.0441	128.83	1.288

The optics program also considers the mirror focus, the aiming error, and the mirror surface error. This program's development was a critical step for the project. Its early completion allowed the parametric analysis to proceed on schedule.

Figure 2-4 shows the effect of focus on the energy delivered through the receiver aperture. The average daily energy collected per heliostat is greatly affected by its position, but rather little by the type of focus. Equinox focus gives the best average daily performance, from most heliostat positions.

FIGURE 2-4 ENERGY FROM A HELIOSTAT AS A FUNCTION OF FOCUS

Fixed focus heliostats were selected, since the minor improvement in the amount of energy collected does not justify the estimated additional cost (~15%) for heliostats with a variable focus.

Figure 2-5 shows the effect on $\overline{\text{BBEC}}$ of changing the spacing (pitch) between the heliostats. Closer spacing increases blocking and shadowing and lowers energy; wider spacing creates greater losses because of image distortion and aiming errors.

Figure 2-6 shows the effect on $\overline{\text{BBEC}}$ of changing the number of heliostat rows per receiver. Clearly, energy cost is not sensitive to the number of heliostats. The minimum cost occurs at about 24 rows of heliostats.

FIGURE 2-5 HELIOSTAT SPACING

Figure 2-7 shows that, initially, combined fixed-focus heliostat errors can be significantly reduced by increasing heliostat sophistication and cost. However, further reductions in error require a greater investment in heliostat construction. Figure 2-7 also shows that the greater cost heliostat with the lesser aiming error is preferred.

FIGURE 2-6 EFFECT OF HELIOSTAT QUANTITY

FIGURE 2-7 HELIOSTAT PERFORMANCE vs COST

FIGURE 2-8 EFFECT OF HELIOSTAT PERFORMANCE: 24 ROWS

Figure 2-8 shows that the increased investment for this greater accuracy is fully justified, within the ranges studied, by the reduction in $\overline{\text{BBEC}}$. The optimum combined heliostat aiming error for the high-temperature line-focus system appears to be 3 mr.

2.2.2.2 Heliostat Parametric Analysis

The heliostats for the line-focus central receiver are a single-axis design (elevation-tracking). They must withstand the environmental exposure outlined in the requirements definition for the high-temperature line-focus contract for Phase I (December 1, 1978 edition) and must be of the lowest practical cost, consistent with the system performance specifications for 100 MW_e output. Cost and performance targets were identified and established for this design by reviewing the application concept, namely single-axis tracking long-focal-length reflecting heliostats, and extrapolating from previous solar hardware optimization and design studies. These design targets are summarized in Table 2-4,

Table 2-4

HELIOSTAT DESIGN TARGETS

Parameter	Value
Performance:	
Mirror specularity	≤ 1.0 mr
Slope error	1-5 mr
Structural deflection	1-5 mr
Drive-pointing error	≤ 2 mr
Foundation deflection	≤ 0.1 mr
Reflectance	≤ 0.9
Cost:	
Low installed capital cost	
Minimum operation and maintenance cost	

To achieve these targets, the following heliostat design features were considered most important:

- Optimum heliostat configuration and geometry
- Cost-effective materials
- Hardware simplification, reduced number of parts
- Mass-producible, modular design
- Simple maintenance

To address and define these features, the heliostat was separated into subsystem parameters and component parameters. The subsystem variables which affect interfaces with the rest of the power plant are variable- vs fixed-focus heliostats, module size, and the grouping of modules to each drive assembly. These variables primarily affect the control subsystem for focusing mechanisms. The heliostat size affects the number and size of drives and parasitics.

In order to properly design the heliostat module, subassemblies had to be defined. These included the reflective panels, structure, and drives (including foundations). For each subassembly, various design configurations and construction materials were considered. Several design approaches were evaluated, along with typical construction materials, for the heliostat module structure. The reflective panels are composite structures, typically consisting of a reflective surface and a structural backing (substrate) that determines the surface profile and optical characteristics. Various materials and configurations were analyzed.

The drive system is composed of a power supply and power transmission components. Various approaches and configurations were considered. Figure 2-9 summarizes the matrix of design alternatives deemed appropriate for the line-focus central power system heliostat modules.

The initial tradeoff was for fixed focal length vs variable focal length. Varying the focal length continuously during the day would maximize performance, at the expense of needing more complex tracking controls, along with complicated hinged or flexible mirror supports, in the module structure. The calculated difference in collectable energy, on an annual basis, between the two schemes is approximately 2%. Cost estimates showed the installed cost of variable-focus heliostats would be 10 to 20% higher than for fixed-focus heliostats. It was concluded that a fixed focus design is more cost-effective and this concept was used in subsequent design studies.

2.2.2.2.1 Heliostat Design Loads

The predominant design loads are the structure dead weight, and the wind loads. In accordance with the system requirements definition, all dead loads and live loads due to wind, ice, and seismic forces were determined. Table 2-5 summarizes these loads. To calculate these loads, three heliostat configurations were defined: four 60-ft modules; four 40-ft modules; and six 20-ft modules to a central drive unit. Two overall load conditions were of importance, either of which might

FIGURE 2-9 HELIOSTAT DESIGN PARAMETERS

Table 2-5

HELIOSTAT DESIGN LOAD SUMMARY

<u>Heliostat Configuration</u>	<u>Dead Load (lb)</u>	<u>Ice Load (lb)</u>	<u>Wind Vertical (lb)</u>	<u>Wind Lateral (lb)</u>	<u>Wind Overturning Moment (ft-lb)</u>
50 mi/h slew/90 mi/h stow					
240/4	3600	5600	9100	7600	45400
160/4	2400	3700	5800	5000	30200
120/6	1200	1900	3000	2500	15100
13 mi/h operating					
240/4	3600	5600	180	300	1800
160/4	2400	3700	120	200	1200
120/6	1200	1900	60	100	600

dominate the resulting design. The survival condition defines the ultimate strength or capacity of a structure or component, and the operating condition typically defines the required stiffness or maximum allowable deflection to maintain acceptable performance. Three operating loads were determined to verify each design: 90 mi/h stow; 50 mi/h slew-to-stow; and the operating wind condition. The design wind specification (of the system requirements definition) was analyzed to define the time-weighted, average wind speed. It was decided that 13 mi/h typifies the daily heliostat operating wind speed.

All wind speeds are reported at the standard elevation of 10 m (33 ft) above the local terrain. To calculate ground-level wind loads, lift, drag and overturning-moment coefficients were determined, as were the effects of the ground boundary layer. Table 2-6 summarizes the coefficients used, based on flat plates at various angles of attack. The boundary-layer effect reduces loads, according to height above the ground, by the one-seventh power. A nominal height of 6 ft was used in all calculations; this translates to a 0.678 wind speed reduction

Table 2-6
WIND LOAD COEFFICIENTS

	<u>Lift</u>	<u>Drag</u>	<u>Pitch Moment</u>
90 mi/h stow	1.2	0.80	0.15
50 mi/h slew-to-stow	1.2	1.2	0.15
13 mi/h operation	1.2	1.2	0.15

into the heliostat. The following equation was used to calculate all loads:

$$F = C_x A \frac{\rho V^2}{2}$$

where

C_x = coefficient (lift, drag, etc.)

A = aperture area

ρ = air density

V = wind speed at height

The following equation was used to calculate actual wind speed into heliostats located 6 ft above the ground.

$$V = \left(\frac{6}{33}\right)^{0.143}$$

$$V_{ref} = (0.784 V_{ref})$$

After the design wind loads governing this heliostat design were defined, a report from Martin-Marietta was obtained. This report presented wind tunnel test results on the effects of field layout (spacing between adjacent two-axis tracking type heliostats), and the effects of various perimeter wind fences, on windloads.

The shielding of interior heliostats by perimeter heliostats and the use of a wind fence for the perimeter heliostats resulted in reported load reductions on the order of 50%. Extrapolating these results to the single-axis heliostats for the line-focus system, such load reductions could result in some diminution in the size of the foundations, drive components, structure, and reflective panels. A brief analysis indicates a cost reduction in the range of 10% could be expected, allowing for the cost of a wind fence. Future designs should take into account these shielding effects. However, more detailed wind tunnel testing should be performed to quantify these effects on a heliostat configuration that more closely resembles a single-axis type.

2.2.2.2.2 Foundations and Support Structure

Several foundation design candidates have been analyzed in previous studies (Refs. 2-3 and 2-4). These include driven wood, concrete and steel piles, on- and below-grade beams of structural steel and concrete, poured-in-place concrete footings and caissons, and steel structures using guy wires. For this application the most cost-effective approach appears to be the cast concrete caisson, poured in place. Two versions were considered: one with a 6-ft reinforced concrete pedestal integrally cast on top of the caisson; and one with a simple caisson (at grade level) to which is bolted a 6-ft steel post (Figure 2-10). The cost comparison includes field installation labor for integrally casting the pedestal with the caisson to save the cost, both of the steel post and of the field labor to set and align it. The all-concrete design requires alignment of only the protruding anchor bolts. However, the steel post requires alignment of both the anchor bolts and the post. The installed cost of the concrete pedestal and caisson was estimated as being 5 to 10% less than that of the steel post and caisson.

The resulting caisson sizes for the three heliostat configurations were dictated by the 90 mi/h stow loads for maximum capacity, rather than the 1/8-inch deflection (≈ 1.5 mr rotation) limit at the nominal 13 mi/h operating condition. It was anticipated that 40- or 60-ft-long modules would be the optimum size, for which the medium-sized caisson was selected.

FIGURE 2-10 CONCENTRATOR SUPPORT AND FOUNDATION

2.2.2.2.3 Module Structure

The three heliostat module structure designs considered were torque tube, space frame, and monocoque. Reviewing the operating conditions (wind loads) and estimating the dead load of the module (with reflective panels and structure at approximately 6 lb/ft^2), the design loads are primarily flexural (weight and wind) with a moderate torsional contribution (wind). The essential design conditions are minimum practical structural deflections at 13 mi/h operation, while being able to survive a 90 mi/h stow and/or 50 mi/h stow-to-stow. Either criterion may dictate the size of structural members, depending on the materials and overall design used. Table 2-7 lists the three concepts, along with comments on key design parameters. In consideration of the loads expected during operation and stow, the best design should yield good flexural stiffness for good optical performance, and should have a minimum weight to achieve low cost and reduce dead loads.

The best compromise for this application is the glass-fiber-reinforced concrete (GRC) torque-tube design. To evaluate these three designs, a layout of each concept was generated. The calculated live and dead loads were used to determine the size of the major structural members, for the deflection limits outlined in Table 2-5. The costing for each design was done on a per-pound-of-weight basis. The specific structural design details are in Volume III, Appendices. The primary reason for the attractiveness of the glass-fiber reinforced-concrete design is the very low material cost, relative to structural steel and other construction materials. In this design, subsystems of the heliostat module can be isolated for optimization and design. The reflector panels are designed to support the reflector surface uniformly and accurately. The structure is designed for accurate pointing of the reflector panels.

2.2.2.2.4 Reflector Panels

The panels that were considered for the reflectors incorporate a second-surface silvered mirror, made of a thin glass laminated to a rigid structural backing. Table 2-8 lists the material considered.

Table 2-7

MODULE STRUCTURE CONCEPT COMPARISON

Concept	Description	Structural Weight (lb/ft ²)	Strength Torsion/Flexure	Strength Stow Loads	Optical Effects	Reflector Interface	Relative Fabrication Cost
Torque tube	Central beam with lateral ribs to support reflector panels	2.7	High/moderate	Moderate, with large tubes sized for flexure	Designed to support and point accurate panel correctly	Attachment points at rib ends	1.00
Space frame	Girder tube of framework for point-support of panels	2.4	Moderate/high	Low, due to low solidity of structure	Designed to support and point accurate panel	Attachment points distributed on structure	0.89
Monocoque	Triangular, rectangular, or "D" tube. Size of full width reflector	5.80	High/high	High, due to large projected area	Differential thermal expansion and deflection can yield high reflector slope errors	Uniformly bonded onto flat side of structure	2.15
GRC torque tube	Triangular cross-section, minimum wall thickness, 2/3 panel width	7.5	High/high	Moderate, due to solid projected area	Designed to support and point accurate panel correctly	Panel attachment points at apexes	0.35

Table 2-8

REFLECTIVE MATERIAL COMPARISON

<u>Reflective Material</u>	<u>Reflectivity</u>	<u>Dollar/ft²</u>	<u>Comments</u>
Thin glass (second surface silvered)	0.92	To be determined	High performance High resistance to cleaning damage
Thin low-iron glass (second surface silvered)	0.94	0.83	High resistance to cleaning damage (the chosen material)
FEK 163	0.85	1.00	Good performance (new); highly susceptible to degradation in cleaning; blisters, adhesive failure
Aluminized FEP	0.87	1.00/1.50	Lower specularly than glass
Silvered FEP	0.95	Experimental	Lower specularly than glass
Front surface aluminized mylar	0.88	0.20	Lower specularly than glass
Alzak	0.85 (new) 0.80 (old)	1.50	Moderately high reflectance; good weatherability; lower specularly than glass; slope errors

Cost and reflectance values were estimated based on past Acurex/Aerotherm collector design experience and data from either reflective material manufacturers or Sandia Laboratories. Glass is the most promising material, based on its high reflectance and specularity, good durability, and relatively low cost.

Table 2-9 lists the panel backing materials investigated. Included are sheet molding compound, variable and constant density cellular glasses, aluminum and paper honeycombs, sheet metal, plate glass, laminated wood products, and fiber-reinforced concrete.

Although very costly, a material of demonstrated performance is sheet molding compound (SMC). The important factors for an accurate reflector contour are high rigidity, low cost, and good forming properties (surface finish, springback, sink marks, etc.). Also important is long-term dimensional stability during exposure to typical temperature, moisture, ultraviolet radiation, and load cycles in service.

Of interest for a large future cost reduction is the use of fiber-reinforced concrete for the reflector backing material. As in the case of the module torque tube structure, the principal advantage of this material is very low cost. To be evaluated for future design are expansion coefficient matching with the flex-glass reflector, and upgrading the module structure and foundation for the added weight of the panels.

2.2.2.2.5 Drive Unit

In the approach to drive design, the sources of power and power transmission for heliostat tracking were evaluated. Considering the likely size and layout of the distributed heliostat field, hydraulic and pneumatic schemes resulted in high capital and maintenance costs for the central power and fluid distribution components. Electric motors were found to be reliable and also had competitive initial costs.

Table 2-10 lists the drive actuators considered for precisely positioning and tracking the heliostat groups. Included in the

Table 2-9

PANEL STRUCTURAL MATERIALS

Structural Material	Slope Error (mrad)	Panel Weight (lb/ft ²)	Cost with Reflector (Dollar/ft ²)	Comments
Sheet molding compound (SMC)	1.0	2.4	2.73	State of the art; expansion matched with glass; good handling, stiffness; expensive (the chosen material)
Cellular glass	1.0	>2.5	3.00	Good potential, but not developed; moisture seal required; no fabrication history
Plate glass	1.0	6.5	2.23	Good durability; heavy dead load for structure
Aluminum honeycomb	1.8	1.5	4.20	Light weight, but expensive; expansion matching difficult
Paper honeycomb	2.4	2.2	2.70	High moisture sensitivity; expansion matching difficult
Fiberglass honeycomb	2.5	2.0	3.20	Expensive
Laminated wood products	7.9	3.2	1.55	High moisture sensitivity
Glass reinforced concrete	1.0	10.0	1.13	Larger dead load for matching may be required; potential for a significant cost reduction

Table 2-10

HELIOSTAT DRIVE TYPE

Drive	Hydraulic Rotary Actuator	Four-Bar		Scissor Mechanism			
		Ballscrew	Hydraulic Cylinder	Ballscrew	Hydraulic Cylinder	Morse	FMC
Initial cost (\$/2400 ft ²)	1,800	740	660	7,100	1,000	1,600	2,030
Parasitic energy (labor/day)	0.2	0.10	0.10	0.2	0.2	0.15	0.15
Emergency shutdown (kW _e)	0.10	0.50	0.50	1.0	1.0	0.75	0.75
Backlash (mr total at 13 mi/h)	1.75	1.6	1.4	1.3	1.1	2.0	2.6
Reliability	Good	Good	Excellent	Good	Excellent	Good	Good
Description	Paired hydraulic cylinder con- nected by rack and pinion; costs include hydraulic power supply (based on one drive)	Ballscrew driven by a synchronous stepper driving through four-bar linkage	Hydraulic cylinder driving through four-bar linkage	Ballscrew driving through scissors mechanism by ac stepper	Hydraulic cylinder driving scissor mechanism	Double reduction helical, single reduction worm, with clutch and ac stepper	Double reduction worm

evaluation were first cost, recurring maintenance and replacement costs, parasitic power requirements, and drive backlash or pointing error. Of the approaches considered, the four-bar mechanism driven through a recirculating ballscrew is considered the most desirable. A sketch of the unit is shown in Figure 2-11.

2.2.2.3 Heliostat Module Sizing

Assuming that the 3.05 m (10-ft) width range for the module was compatible with the cost-optimum system configuration, consideration was given to both 2.44-m and 3.66-m (8- and 12-ft) widths, to note any significant cost sensitivity. Direct labor and materials, tooling, and indirect costs were generated for the 3.05-m (10-ft) heliostat by subgroup (reflective panel, structure, drive, foundation, and installation). These costs were then scaled for 2.44-m and 3.66-m (8- and 12-ft) widths, to account for differences in panel and structure stiffness, drive and foundation component strengths, and shipping and installation labor requirements. Figure 2-12 is a plot of installed module cost in dollars per square foot of aperture, as a function of module width. A weak dependence is seen which indicates a 3.05-m to 3.66-m (10- to 12-ft) module width is desirable. A similar analysis for module length resulted in choosing the 12.2-m (40-ft) length.

The overriding considerations in the choice of the 3.05-m (10-ft) width turned out to be plastic molding size limitations and practical handling and shipping limitations. Currently, in the plastics industry, 3.05-m (10-ft) molds can be fabricated, but they require the full capacity of the largest presses available, approximately 2.7×10^6 kg (2500 tons).

Shipping by common carrier would limit panel widths to approximately 2.44 m (8 ft). Special permits and slight equipment modifications to a dedicated truck fleet would allow shipping 3.05-m (10-ft) widths. An actual shipping cost reduction is realized for this oversize load.

FIGURE 2-11 HELIOSTAT DRIVE MECHANISM

FIGURE 2-12 HELIOSTAT COST

2.2.3 HelioStat Foundations

Parametric heliostat foundation designs (and cost data) were developed to assist in selecting the heliostat foundation design. Previous work by Bechtel (Ref. 2-5) has shown that the least expensive foundation, under relatively light loads (assuming favorable soil conditions) would be an augered caisson. Although, in situations where heavy gravel and boulders are encountered, a spread footing would be required, only augered caissons were investigated in the parametric analysis.

Three heliostat configurations with different foundation spacings were defined. These formed the basis for nine foundation designs. The design of a concrete pedestal supported by the caisson and extending 1.8 m (6 ft) above grade was also included for each foundation.

2.2.3.1 Design Criteria

The design criteria were derived from a combination of recognized codes and standards, the requirements definition document, and the heliostat design requirements.

2.2.3.1.1 Materials

Acceptance criteria for materials are:

- Concrete compressive strength at 28 days: 20.7 MPa (3000 psi)
- Reinforcing steel yield stress: 414 MPa (60,000 psi)
- Site soil: UBC Class 3 sandy gravel

2.2.3.1.2 Operating Conditions

The key condition for the performance of the foundations was a deflection limit of 3.2 mm (1/8 inch) maximum horizontal deflection at 1.8 m (6 ft) above grade.

2.2.3.1.3 Survival Conditions

No performance criteria were specified for the foundations except that failure should not occur.

2.2.3.2 Loading Criteria

2.2.3.2.1 Design Loads

Loads were determined by each caisson type, representative of different collector spans. These loads are summarized in Table 2-11. Live loads due to cleaning operations were considered negligible and were not included.

2.2.3.2.2 Load Combinations for Drilled Caissons

The load combinations assumed for the operating and survival conditions are given below. The most critical of these load combinations determined the design; each combination is critical when allowable stresses are reached in the foundation soils. Seismic loads for these low superstructures are negligible and are not shown in the load combinations.

Table 2-11

COLLECTOR FOUNDATION LOADING

	Caisson Type		
	A	B	C
Foundation spacing, m (ft)	18 (60)	12 (40)	6 (20)
Collector dead load, kg (lb)	1600 (3600)	1100 (2400)	550 (1200)
Ice load, kg (lb)	2550 (5616)	1700 (3744)	850 (1872)
Operating conditions			
Vertical wind, $\pm$ kg ($\pm$ lb)	90 (200)	55 (120)	27 (60)
Lateral wind, kg (lb)	136 (300)	90 (200)	45 (100)
Survival conditions			
Vertical wind, $\pm$ kg ($\pm$ lb)	4100 (9100)	2600 (5800)	1100 (2900)
Lateral wind, kg (lb)	3500 (7600)	2300 (5000)	1100 (2500)

- Operating conditions
 - D + W
 - D + I
 - $0.75(D + I + W)$
- Survival conditions
 - $0.67(D + W)$
 - $0.5(D + I + W)$

where

D = dead loads

W = wind loads for operating (or survival) conditions

I = ice loads

2.2.3.3 Design Procedure

From preliminary calculations it was found that a survival loading condition for the three caisson types would govern the design. Three caisson diameters were assumed for each foundation spacing and, based on the lateral load, their corresponding embedment lengths were calculated in accordance with The Uniform Building Code (UBC) Section 2907(f). The general design arrangement is shown in Figure 2-13. The vertical loads, both upward and downward, were checked using the

FIGURE 2-13 HELIOSTAT FOUNDATION ARRANGEMENT

assumption that resistance was provided by skin friction and end bearing. The lateral deflection under static loads was checked, for each design, under operating conditions. The resultant sizes are listed in Tables 2-12 and 2-13 for the drilled pier and pedestal, respectively.

Table 2-12

SIZE OF HELIOSTAT FOUNDATIONS

Caisson Type*	Diameter m (ft)	Depth m (ft)	Concrete m ³ (yd ³)	Rebar kg (lb)
A-1	0.91 (3)	3.05 (10)	2.0 (2.6)	90 (200)
A-2	0.76 (2.5)	3.28 (10.75)	1.5 (2.0)	77 (170)
A-3	0.61 (2)	3.58 (11.75)	1.1 (1.4)	64 (140)
B-1	0.76 (2.5)	2.74 (9)	1.2 (1.6)	50 (110)
B-2	0.61 (2)	3.05 (10)	0.9 (1.2)	55 (120)
B-3	0.38 (1.25)	3.66 (12)	0.5 (0.6)	45 (100)
C-1	0.71 (2.33)	2.13 (7)	0.8 (1.1)	50 (110)
C-2	0.61 (2)	2.29 (7.5)	0.7 (0.9)	41 (90)
C-3	0.38 (1.25)	2.74 (9)	0.3 (0.4)	32 (70)

*See Table 2-11.

Table 2-13

SIZE OF HELIOSTAT PEDESTALS

Type	Size mm (inches)	Concrete m ³ (yd ³)	Rebar kg (lb)
A	305 × 381 (12 × 15)	0.2 (0.3)	45 (100)
B	203 × 330 (8 × 13)	0.15 (0.2)	41 (90)
C	203 × 330 (8 × 13)	0.15 (0.2)	41 (90)

2.2.3.4 Cost Estimates

Based on the previous technical description, cost estimates were prepared for the nine alternative foundation designs. Initial estimates were made, based on conventional construction practices, and utilizing a subcontractor for the augering work. Costs were derived from previous Bechtel construction experience and oral quotations from various contractors and material suppliers in the southwestern U.S. Due to the highly repetitious nature of the work involved, the focus of the estimate shifted to examining various cost reduction possibilities. Implicit in this examination was the assumption that the cost estimate was representative of the first commercial plant. The possibility was explored of realizing a cost reduction by using advanced equipment to reduce the labor input by mechanizing the augering, rebar placement, and concrete filling processes. It was concluded by Bechtel that the cost of automation equipment outweighed the possible labor saving advantages for the first plant. However, it was concluded that for future generations of plants, more advanced construction equipment could be employed at an overall cost reduction, since a large potential commercial market was established during the market analysis (Section 7.0). The equipment cost could then be written off over a large number of construction projects.

For the first generation commercial plant, it is conceivable that labor saving could be achieved over the standard installation rates for this type of work through the use of multiple crew units. In view of the extent of the repetitions involved, it was determined that a 95% learning curve could reasonably be employed over the duration of the work, resulting in an approximate 25% labor manhour saving. As used in this study, a 95% learning curve implies that each time the cumulative production is doubled, the labor required to install each foundation is reduced 5%. Table 2-14, which includes the learning curve effect, contains the results of the estimate for the first generation commercial plant. The estimates include material and direct labor costs. Engineering costs are estimated at 10% of the total direct costs. The estimates are in mid-1978 dollars, with delivery to a jobsite in the southwestern U.S.

Table 2-14

HELIOSTAT FOUNDATION COSTS
(1978 Dollars)

Type	Cost per Unit (\$)	Units per 100 MW _e Plant	Total Cost (\$1000)
A-1	273	19,520	5,330
A-2	236	19,520	4,610
A-3	202	19,520	3,940
B-1	192	29,120	5,590
B-2	166	29,120	4,830
B-3	127	29,120	3,700
B-3*	94	29,120	2,740
C-1	138	57,920	7,990
C-2	123	57,920	7,120
C-3	93	57,920	5,390

* Without pedestal

From Table 2-14, it is apparent that the smallest diameter, longest length foundations are the most cost effective. The type B-3 foundation, a 0.38-m (1.25-ft) diameter caisson, 3.66 m (12 ft) deep, is the most economical.

2.3 Receiver Subsystem Parametric Analysis

2.3.1 Performance Analysis

The receiver analysis subprogram utilizes the flux profiles, as calculated by the optics subprogram, to predict the draw-salt mass-flow that meets the desired temperature boundary conditions at the inlet and outlet. The subprogram, as called by the main program, for each hour of heliostat operation during each of three specific days, predicts the receiver performance as a balance of three processes:

- The redistribution of incident solar flux from reflections within the cavity.
- The redistribution of infrared radiation from emission and reflection within the cavity.
- The conduction through the tube walls (and the subsequent convection) into the working fluid.

Wind losses from both free and forced convection are accounted for, after a solution is determined.

The mathematical description of these processes does not lend itself to a closed form solution; hence, an iterative process is employed. The general methodology is as follows:

- Assume that the receiver may be modeled as a two-dimensional element, with working fluid flowing circumferentially through two flow paths from each edge of the aperture towards a common outlet at the back of the receiver. The circumferential paths can then be divided into a finite number of segments for calculation. It is further assumed that properties are uniform in each segment.
- Calculate the redistribution of solar flux from reflections within the cavity, and from losses reflected back through the aperture.
- To initiate the iterative process, assume that the incident solar flux heats the working fluid of each node directly (i.e., radiation losses are neglected). The flow rates can then be calculated from the relation:

$$\dot{m}_1 = \frac{1}{\Delta H} \int_{s_1} \psi(s) ds; \quad \dot{m}_2 = \frac{1}{\Delta H} \int_{s_2} \psi(s) ds$$

where

$\dot{m}_1$ = mass flow through path 1

ΔH = enthalpy change from inlet to outlet

$\psi(s)$ = net flux into working fluid (in this case, equal to the redistributed solar flux)

s = flow path length

- Using the calculated mass flow, the specific properties at each node may be established. In particular, the enthalpy is calculated as flux adds energy to the working fluid, and the temperature is determined by the state of the fluid at the pressure and enthalpy at each node.

$$H_j = H_{j-1} + \psi_j \cdot \frac{\Delta s}{\dot{m}}$$

$$T_j = f(H_j, p_j)$$

where

H_j = enthalpy at segment j

Δs = path length associated with segment j

T_j = bulk fluid temperature at segment j

p_j = local pressure at segment j

- The outside wall temperature of the fluid piping is calculated by using equations for the fluid convection to the pipe, and then for the thermal conduction through the pipe wall.

$$T_{w_j} = \frac{x_j}{h_f} + T_j$$

$$T_{w_j} = \frac{x_j \cdot t}{k} + T_{w_j}$$

where

T_{w_j} = inside wall temperature

T_{w_j} = outside wall temperature

h_f = heat transfer coefficient of working fluid

k = thermal conductivity of pipe

t = pipe wall thickness

- The net flux into the working fluid can now be recalculated, using the following relation which combines both the solar influx and the complex radiation in the cavity.

$$\psi_j = \epsilon_j \sum_{i=1}^N B_i F_{j-i} + \phi_j - \epsilon_j EB_j$$

where

ϵ_j = emissivity at segment j

B_i = radiosity at segment i

F_{j-i} = view factor from segment j to i

ϕ_j = redistributed solar flux at node j

EB_j = emissive power at node j , $\sigma(T_{w_j} + 460)^4$

- Finally, the new ψ_j is compared to the previous ψ_j . The process of estimating fluid mass flows, through calculation of the net flux, is continued until the entire ψ vector converges within an established criterion.

The mass flow per unit length (provided by the subprogram) becomes the input to the main program, for calculations of energy processed through the energy storage subsystem, the energy transport subsystem, and the electric power generating subsystem.

2.3.1.1 Symbol Description

The following symbols are used in the preceding analysis:

h_f	= heat transfer coefficient, $\text{Btu/h-ft}^2\text{-}^\circ\text{R}$
H	= enthalpy, Btu/lbm
k	= thermal conductivity of receiver pipe, $\text{Btu/h-ft-}^\circ\text{R}$
$\dot{m}_{1,2}$	= mass flow of working fluid through paths 1 or 2, per receiver unit width, lbm/h-ft
P	= pressure, psi
s	= path length, ft
t	= pipe thickness, ft
T_j	= fluid temperature at node j , $^\circ\text{R}$
T_w	= outside wall temperature, $^\circ\text{R}$
T_ω	= inside wall temperature, $^\circ\text{R}$
ΔH	= change in enthalpy from inlet to outlet, Btu/lbm
ϵ	= emissivity of wall for infrared temperatures
ψ	= heat flux at node, Btu/h-ft^2
ϕ'	= redistributed solar flux, Btu/h-ft^2
B	= radiosity, Btu/h-ft^2
EB	= emissive power, Btu/h-ft^2

A detailed description of this analysis is provided in Volume III, Appendices.

2.3.1.2 Specifications

The following specifications were introduced as inputs to the receiver program:

- Variables in common with the heliostat program
 - Receiver diameter
 - Receiver aperture width
 - Receiver aperture angle
 - Number of elements considered
- Variable derived from the heliostat program
 - Incident flux on each element

- Independent variables
 - Solar absorptivity per element
 - Emissivity per element
- Variables in common with the energy transport subsystem
 - Choice of working fluid (water or salt) and fluid characteristics
 - Input enthalpy
 - Exit enthalpy

The output of the receiver subroutine is the mass flow per unit length of the receiver, and a listing of losses.

The receiver cavity reduces reflection and radiation losses, and it accounts for about 5% of the efficiency. Figure 2-14 shows how practical receiver efficiencies are achieved at relatively low solar concentrations. At high concentrations, receiver efficiency is relatively insensitive to the emissivity of the receiver surface. However, as the

FIGURE 2-14 SOLAR RECEIVER EFFICIENCY

concentration decreases the receiver efficiency becomes more sensitive to the emissivity of the receiver surface. As the emissivity approaches zero, the receiver efficiency approaches the value of the solar absorptance, α_s , for all solar concentration ratios. SRI chose to use a conservative value of $\epsilon_{525^\circ\text{C}} = 0.06$ for the tradeoff studies, although passivated selective surfaces have been shown to retain an emissivity ($\epsilon_{500^\circ\text{C}}$) of 0.045 for the 500°C blackbody spectrum, after 1000 h at 500°C . (Ref. 2-6) Heretofore, only point-focus systems were able to achieve these high receiver efficiencies. Line-focus systems will benefit greatly from the development of these selective surfaces. Their commercial availability has been projected for the mid-1980s.

2.3.2 Receiver Parametric Analysis

Preliminary receiver calculations were performed, based on flux plots provided by the heliostat program. The 6-ft-diameter receiver selected for analysis had a typical cross section as shown in Figure 2-15

FIGURE 2-15 SCHEMATIC OF RECEIVER CROSS SECTION

and was divided into 24 zones as shown in Figure 2-16. The molten salt enters at 550°F in zones 1 and 24 and leaves from zone 14 at the point of peak heat flux. The receiver molten salt mass flow rate selected by

FIGURE 2-16 RECEIVER ZONES

SRI for the parametric analysis was 222 lb/h-ft of receiver length, of which 55% flows in the upper portion and 45% in the lower portion of the receiver. Thus, the 10 upper zones carry 55% of the duty and the 14 lower zones carry 45%.

2.3.2.1 Requirements and Data

Early in the parametric analysis, it became clear that several constraints were critical in designing the receiver:

- A maximum receiver module length of 100 ft, imposed by shipping limitations.
- A minimum tube wall thickness of 0.120 inch if the tubes were to be longitudinally welded together, as in Monowall® construction.
- A minimum tube outside diameter of at least 0.5 inch whenever possible. As the tube O.D. decreases below 0.5 inch, the tube-to-tube and tube-to-header welds become more difficult and expensive.
- A minimum tube wall thickness of at least 0.035 inch because of manufacturing and handling constraints.

Table 2-15 gives the design and operating characteristics of the receiver design, using cylindrical tubing for the heat-absorbing surfaces.

Due to the moderate flux densities in the line-focus central receiver, further cost reduction is possible by using heat transfer panel technology. This technology has been used for years in refrigerators and hot process tanks, and has been improved in recent years for more demanding applications. In this application, two sheets of metal would be used. The flow path would be stamped into one or both, and the sheets brazed, or spot welded, and seam welded around the edges. Since many techniques are available for forming and assembling, the selection would depend on compatibility with the working fluid. This type of receiver is recommended as the subject of further cost reduction study.

Table 2-15

RECEIVER DESCRIPTION

Design Characteristics

Absorber tube	
Outside diameter	25.4 mm (1.00 in)
Inside diameter	22.9 mm (0.902 in)
Wall	1.24 mm (0.049 in)
Receiver module tube length	27.43 mm (90.0 in)
Number of receiver modules	486
Tubes*	136
Upper circuit	57
Lower circuit	79
Receiver materials	Carbon steel; Incoloy 800
Total* mass of tubes	3068 kg (6,750 lb)
Insulation material	Fiberfrax TM
Total* mass of insulation	1,473 kg (3,240 lb)
Shell material	Carbon steel
Total* mass of shell and eyelid	2,291 kg (5,040 lb)
Total* mass of corrugated aluminum siding	368 kg (810 lb)
Receiver pressure drop correlations	Moody friction factors [†]

Operating Characteristics

Annual receiver efficiency	0.72
Receiver working fluid pressure drop	0.69 MPa (100 psi)
Receiver working fluid velocities, maximum	0.78 m/s (2.49 ft/s)
Peak metal temperature	601°C (1114°F)
Peak working stress in receiver	To be determined
Fatigue life	More than 13,000 cycles
Number of cycles	13,000
Design life	30 yr
Receiver heat transfer coefficients at design power	
Tube wall	17.227 W/m ² -°C (3036 Btu/h-ft ² -°F)
Receiver working fluid	1,969 W/m ² -°C (347 Btu/h-ft ² -°F)
Overall	1,765 W/m ² -°C (311 Btu/h-ft ² -°F)

*Per receiver module

[†]Moody, F.F., Friction Factors for Pipe Flow, Trans. ASME Vol. 66, p. 671, 1944.

2.3.2.2 Receiver Convection Losses

The cavity receiver convective heat loss, h , is difficult to calculate in a receiver having the proposed geometry. For the worst case (a flat plate), the width of the aperture was taken at an average temperature of 700°F above ambient. The worst case (free convection) is given by $h = 0.22 (\Delta T)^{1/3}$ Btu/h-ft²-°F for a flat plate facing upwards. This orientation gives a convective loss of 1.6 kW_{th}/ft of receiver length. The cryogenic wind tunnel tests at the University of Illinois indicated that actual free convection losses are about one-half of this value.

To estimate the worst possible case (forced convection), the Reynolds number power coefficient from the Nusselt-Reynolds correlation was taken to be 0.731, a value taken from Jakob (Ref. 2-7) for a vertical flat plate. The crossover between free and forced convection was estimated at 5 mi/h. The best case assumption is that the convective loss is negligible.

The best and worst case convective loss assumptions were first used for the parametric analysis in the computer model and system trade-off studies. Figure 2-17 compares the best and worst case designs; differences in performance can be seen as differences in the value of BBEC. The tradeoff studies used a convective loss that was one-half the worst case, a value believed to be conservative. Discussion of cryogenic wind tunnel tests and water table tests of the receiver models are discussed in Section 6.

Other system requirements that affected the receiver design are listed below:

- The molten salt must exit the receiver at the point of maximum heat flux (roughly at the end of zone 14). This required that the already low flow be split in two, one for the upper circuit (55%) and the rest for the lower circuit (45%).
- There should be no gaps or spaces between the heat transfer tubes, and all heat flux must impinge upon active heat transfer surfaces. Therefore, this requires that the tubes either be touching each other, or that some heat transfer surface (such as fins or metal strips, as in Monowall® construction) must cover any space between the tubes.

FIGURE 2-17 EFFECTS OF CONVECTION LOSSES

- Maximum receiver fluid pressure drop is limited to 100 psi.
- Maximum outside tube metal temperature is limited to 1150°F.

Values for the molten salt fluid properties were taken from the literature and are given in Table 2-16. To simplify analyses, they were curve-fit to within the range of the receiver working temperature.

Variables that were investigated during the parametric analysis included tube O.D., tube wall thickness, receiver length, and number of tubes per panel. As the parametric analysis proceeded, some limits on the range of those variables became apparent. This helped reduce the range of interest, as follows:

- Tubes with less than 0.5 inch O.D. gave excessive pressure drops. Tubes with O.D. greater than 1.5 inch gave very low convection heat transfer coefficients, resulting in high tube wall temperatures. This increased receiver heat loss.
- Each panel should have no more than 10 tubes, thereby avoiding large differences in both enthalpy pick-up and thermal expansion within the panel.

Table 2-16

MOLTEN SALT PROPERTIES

T (°F)	ρ (lb/ft ³)	C_p (Btu/lb-°F)	$\mu \times 10^3$ (lb/ft-s)	k (Btu/h-ft-°F)	Prandtl Number (Pr)
500	120.4	0.371	2.69	0.23	15.62
600	117.9	0.371	1.88	0.23	10.92
700	115.4	0.371	1.38	0.23	8.01
800	113.6	0.371	1.11	0.23	6.45
900	111.7	0.371	0.974	0.23	5.66
1000	108.7	0.371	0.674	0.23	3.91

2.3.2.3 Tube Bundle

A receiver made of circular tubes was first considered. However, finned tubes were selected over tangential circular tubes. This reduced the required number of tubes, thereby reducing the number of tube bends and tube-to-header welds. The lesser number of tubes required also reduced the pressure drop. Furthermore, the support of tangential circular tubes presented serious design problems.

MonowallTM construction, shown in Figure 2-18, was dropped from further consideration because it required very thick tube walls (0.12 inch minimum), resulting in a very heavy, costly receiver.

Circular tubes with extruded fins and flattened tubes were considered next. Typical cross sections are shown in Figure 2-18. The weights of steel and salt per foot of receiver length were estimated for a series of parametric cases. The results (given in Table 2-17 for a finned circular tube and in Table 2-18 for a flattened tube) show that the flattened tube design required fewer tubes, which resulted in some savings in molten salt weight per foot of receiver. However, the expected tube weight savings were offset by the heavier wall thickness. There is also a cost penalty associated with the extra labor required to flatten the tubes.

MONOWALLTM CONSTRUCTION

FINNED TUBES

FLATTENED FINNED TUBES

FIGURE 2-18 TUBE CONFIGURATIONS

Table 2-17

WEIGHTS FOR FINNED CIRCULAR TUBES

Wall (mil)	O.D. (in)	Number of Tubes	Tube Weight (lb)	Salt Weight per ft (lb)	Total* Weight per ft (lb)
35	0.5	226	47.0	25.9	72.9
	1.0	136	54.2	72.8	127.0
	1.5	97	57.2	122.8	180.0
	2.0	75	58.5	173.0	231.5
49	0.5	226	64.1	22.6	86.7
	1.0	136	75.0	68.5	143.5
	1.5	97	79.3	118.1	197.4
	2.0	75	81.4	167.9	249.3
65	0.5	226	82.4	19.2	101.6
	1.0	136	97.9	63.7	161.6
	1.5	97	104.2	112.7	216.9
	2.0	75	106.2	162.3	269.4

* Tubes and salt only.

Table 2-18

WEIGHTS FOR FLATTENED TUBES

t* (in)	Wall (mil)	O.D. (in)	Number of Tubes	Tube Weight per ft (lb)	Salt Weight per ft (lb)	Total† Weight per ft (lb)
0.25	83	1.0	108	95.4	23.7	119.1
	134	1.5	74	151.7	25.3	177.0
	188	2.0	56	211.0	26.0	237.0
0.50	65	1.0	116	83.1	44.5	127.6
	120	1.5	77	143.1	48.1	191.2
	180	2.0	58	210.2	49.8	260.0
0.75	65	1.0	129	92.6	59.4	152.0
	109	1.5	82	139.9	69.0	208.9
	156	2.0	61	194.7	74.2	268.9

† Tubes and salt only.

Table 2-19 shows typical heat transfer coefficients and Reynolds numbers, Re, for circular tubes. For 0.5- and 2-inch I.D. tubes, no values could be obtained in the turbulent region ($Re > 10,000$) within the constraints of a 100-psi maximum pressure drop, and a maximum receiver length of 100 ft.

Next, Panelcoil[®] heat transfer technology was considered. Panelcoils have been used for years in refrigerators and hot process tanks. As shown in Table 2-20, vertical Panelcoils gave very low Reynolds numbers (in the laminar flow region) and heat transfer coefficients. This resulted in high tube wall temperatures, which increased receiver heat losses and required the use of tubes made from expensive high-temperature alloys, such as Incoloy 800. Vertical Panelcoils are also undesirable because they are of one-piece construction, precluding the use of less expensive materials in the low-temperature regions.

Table 2-19
PARAMETRIC PERFORMANCE VALUES FOR CIRCULAR TUBES

Wall (mil)	O.D. (in)	Number of Tubes	Length (ft)	h (Btu/h-ft ² -°F)	Reynolds Number (Re)	V (ft/s)	ΔP (psi)
65	1.0	1	42	348	18366	2.48	100
		2	85	352	18664	2.52	100
	1.5	1	67	223	18600	1.60	30
	1.5	1	100	309	27901	2.39	90
49	1.0	1	46	279	18857	2.45	100
		2	90	347	19167	2.49	100
	1.5	1	69	221	18957	1.59	30
	1.5	1	100	295	27176	2.27	80
35	1.0	1	47	340	19207	2.45	100
		2	95	343	19603	2.47	100
	1.5	1	72	220	19242	1.58	30
	1.5	1	98	282	26253	2.15	70

Table 2-20

PARAMETRIC PERFORMANCE VALUES FOR VERTICAL PANELCOILS[®]

O.D. (in)	Reynolds Number (Re)	h (Btu/h-ft ² -°F)	Maximum Wall Temperature (°F)
0.250	134	86.5	1281
0.375	87	50.0	1450
0.500	68	35.9	1608

Horizontal Panelcoils gave Reynolds numbers and heat transfer coefficients similar to those obtained with circular tubes (Table 2-19). Figures 2-19 and 2-20 show the horizontal Panelcoil assembly conceptual design and the developed view of the Panelcoil receiver assembly, respectively. Preliminary cost estimates for the Panelcoils ranged from \$51/ft² to \$185/ft² from one vendor. The great range in the cost estimate suggests that the vendor was geared toward his own small markets. This concept should be pursued further, independently of any particular manufacturer. Lower costs than tube construction are believed possible.

An analysis of the heat flux distribution showed that zones 1 through 4 accounted for almost 21% of the receiver area, yet received only about 2% of the total energy. It was decided to remove the tubes from these zones, saving receiver weight and cost and pumping power costs. The energy incident on zones 1 through 4 will now be reflected and reradiated to the other zones, and only a small amount will be lost through the aperture. SRI BBEC analysis showed that the savings in receiver and tower costs outweighed the small increase in loss through the aperture.

2.3.2.4 Receiver Conduction Losses

Conduction losses, per linear foot of receiver, were estimated as a function of the insulation thickness and the outside heat transfer coefficients for an ambient air temperature of 26°C (80°F). The results

FIGURE 2-19 CONCEPTUAL DESIGN FOR THE HORIZONTAL PANELCOIL®

FIGURE 2-20 DEVELOPED VIEW OF THE HORIZONTAL PANELCOIL® RECEIVER ASSEMBLY

are given in Table 2-21. It was found that the thermal resistance of the outside air film to heat flow was small compared to the insulation resistance. A review of the table shows that the estimated heat losses changed by less than 11% for the range of the outside air heat transfer coefficients considered. Thus, the uncertainty associated with the outside air convective heat transfer coefficient acting on the outside surface of the receiver is not a critical factor, and the conservative assumption that the outside surface of the receiver is at ambient air temperature is a reasonable approximation.

An average insulation thickness of 6 inches was assumed for costing purposes. Table 2-22 gives the conduction losses, per linear foot of receiver, as a function of the ambient temperature. It was assumed that the receiver outside surface is at the ambient air temperature, since the effect of the heat transfer coefficient, and therefore the wind speed, is negligible.

2.3.3 Receiver Towers and Bridges

A parametric performance and cost study was conducted on three alternative designs for the towers and connecting bridges necessary for supporting the cylindrical solar line-focus receiver at an elevation of 61 m (200 ft) above grade. This tower height was selected for two reasons. First, tall towers are desirable to maximize the number of collector rows per receiver row. However, there is an optical and economic limit to the tower height, which is set by the accuracy of the collector surfaces. Second, high lift cranes are available for installing equipment up to a maximum height of 61 m (200 ft). Although this height is below the optical and economic limit, it was selected as a viable compromise between collector-receiver performance and tower installation costs.

The tower spacing was varied to determine the optimum size of the bridges and towers. A Warren-type truss, supported on the top of the towers, was analyzed for the receiver support system because it was considered the least expensive of the parallel chord trusses. Augered,

Table 2-21

CONDUCTION HEAT LOSSES vs INSULATION THICKNESS

Average Insulation Thickness (in)	Insulation Weight (lb/ft)	Insulation Cost (\$/ft)	Conduction Heat Losses					
			h = 2		h = 30		h = ∞	
			kW/ft	%*	kW/ft	%*	kW/ft	%*
3	40.8	16.60	0.622	3.902	0.683	4.290	0.689	4.321
6	69.0	28.54	0.340	2.133	0.356	2.236	0.358	2.244
9	93.3	40.46	0.239	1.499	0.246	1.547	0.246	1.550
12	125.6	52.38	0.187	1.174	0.191	1.200	0.191	1.202
15	154.0	64.30	0.155	0.975	0.158	0.992	0.158	0.993
18	182.3	76.22	0.134	0.840	0.135	0.853	0.136	0.853

Notes:

h = Outside air heat transfer coefficient in $\text{Btu/h-ft}^2\text{-}^\circ\text{F}$ Ambient temperature = 80°F

All values per linear foot of receiver

*Percent of incident energy

Table 2-22

CONDUCTION HEAT LOSSES vs AMBIENT TEMPERATURE
(Average Insulation Thickness = 6 in)

Ambient Air Temperature		Conduction Heat Losses (kW/ft)	Percent of Incident Energy
°C	°F		
-20	-4	0.404	2.534
0	32	0.384	2.410
20	68	0.364	2.286
40	104	0.344	2.160
60	140	0.325	2.036

belled caissons were used for the tower foundations for two reasons. First, the caissons provided the required uplift resistance to the lateral loads of the design. Second, they avoided the high labor costs involved in constructing spread footings. Although it should provide savings in steel weight, an internally guyed tower was not investigated because it might be considerably more flexible than a self-supporting structure and could possibly allow too much receiver structure deflection. Further analysis of this construction could be productive.

2.3.3.1 Design Criteria

The design criteria were obtained from a combination of recognized codes and standards and the requirements definition document.

Materials. Acceptance criteria for materials are similar to those listed in Section 2.6.1 for soil, concrete, and reinforcing steel. ASTM-A36 structural steel and A325 bolts were assumed to be used for the purposes of the design analysis of the towers and bridges.

Operating Conditions. The requirements definition document operating conditions were used, with the exception of the ice loading, which was reduced to be consistent with contemporary tower design practice. This provided a cost reduction for the structure. Key operating conditions are listed below:

- Wind: 5.8 m/s (13 mi/h) at 9.2 m (30 ft), varying exponentially with height above the ground, as per the requirements definition document
- Ice: 12.7 mm (1/2 inch) thickness
- Seismic: Uniform Building Code (UBC) Zone 3 conditions

Survival Conditions

- Wind: 40.3 m/s (90 mi/h) at 9.2 m (30 ft), varying exponentially with height above the ground, as per the requirements definition document
- Ice and seismic: as above

2.3.3.2 Loading Criteria

Design Loads. The dead loads, calculated for the selected structural sections and materials, included a solar receiver linear weight of 522 kg/m (350 lb/ft). Maximum wind forces, calculated at the top of the tower, are:

- Operating: 393 Pa (8.2 lb/ft²)
- Survival: 2739 Pa (57.2 lb/ft²)

Load Combinations. The load combinations assumed for the operating and survival conditions are listed below. The most critical of these load combinations determined the design for each case.

- Operating Conditions
 - 1.27 (D + I)
 - 0.75 (D + E)
 - 1.27 (D + I) + 2.54W
- Survival Conditions
 - 1.27 (D + W)

where

D = dead loads

W = wind loads for operating or survival conditions

I = ice loads

E = seismic loads

These load combinations are appropriate for tower structures according to the methods of ASCE Manual 52 (Ref. 2-8).

2.3.3.3 Design Procedure

Because of its similarity to a transmission tower, the receiver tower design was based on the formulae and procedures outlined in ASCE Manual 52. The wind force was calculated on 175% of the projected area of flat surfaces, and on 67% of the area of cylindrical surfaces. Three truss spans and towers were analyzed, corresponding to tower spacings of 46 m (150 ft), 61 m (200 ft), and 76 m (250 ft). In each analysis, the survival wind case governed the design. The foundation design was governed by uplift, and included a safety factor of two. The general structural configuration is shown in Figure 2-21. The computed structural weights are listed in Table 2-23, and the number of required towers and bridges is outlined in Table 2-24.

2.3.3.4 Cost Parametric Study

Based on the previous technical description, estimates were made for the three alternative tower and bridge designs listed in Table 2-23. A material takeoff estimated the quantities of excavation, concrete, rebar, and steel required. The capital cost estimate was based on previous Bechtel construction experience, as well as on oral quotations from various contractors and material suppliers.

To calculate the cost of the various tower and bridge configurations, the estimate focused on the tradeoff between shop and field assembly of the steelwork. It was determined that the first alternative would be to order the steel components fabricated, galvanized, and bundled in the shop. Final assembly would then take place in the field.

FIGURE 2-21 GENERAL ARRANGEMENT OF TOWERS AND BRIDGE TRUSSES

Table 2-23

SUMMARY OF TOWER AND BRIDGE STRUCTURAL QUANTITIES

	Span, m (ft)		
	46 (150)	61 (200)	76 (250)
Structural steel, tonnes (tons)			
Bridge	10.5 (11.6)	19.0 (20.9)	36.4 (40.0)
Tower	14.6 (16.1)	16.8 (18.5)	20.1 (22.1)
Tower foundation (per tower)			
Concrete, m ³ (yd ³)	4.0 (5.2)	5.7 (7.4)	6.7 (8.8)
Rebar, kg (lb)	236 (520)	336 (740)	400 (880)

Table 2-24

REQUIRED TOWER AND BRIDGE QUANTITIES
(16 Rows)

Span m (ft)	Towers per Row	Total Towers	Bridges per Row	Total Bridges
45.6 (150)	25	400	24	384
61.0 (200)	19	304	18	288
76.1 (250)	15	240	14	224

The costs of the three alternative designs (based on the quantities listed in Table 2-24), are given in Table 2-25. The cost estimates include material and direct labor costs. Engineering costs are estimated at 10% of the total direct costs. The estimates are made in mid-1978 dollars, with delivery to a jobsite in the southwestern U.S.

Table 2-25

FOUNDATION, TOWER AND BRIDGE COSTS
(In Thousands of Mid-1978 Dollars)

<u>Span m (ft)</u>	<u>Foundations</u>	<u>Towers</u>	<u>Bridges</u>	<u>Total</u>
45.6 (150)	720	8340	5840	14,900
61.0 (200)	730	7320	7740	15,800
76.1 (250)	810	6930	11460	19,200

It is apparent from the cost summary that the shortest span results in the least expensive combination. The 45.6-m (150-ft) span, therefore, was selected for the conceptual design. The relative cost of the shortest span should be even less than shown, because the greater number of bridges and towers will provide for the larger "learning curve" cost reduction.

2.4 Heat Transport Subsystem (HTS)

The parametric analysis for the HTS involves three factors. First is the optimization of the HTS for the baseline plant design. Second is the influence of the final feedwater temperature and throttle steam pressure of the electrical power generating systems (EPGS) on the cost of the HTS. The third influence is that of the solar multiple on the cost of the HTS for the baseline plant design. As outlined on the requirements definition document, the HTS includes the field salt piping and the steam generator. A schematic flow diagram for the HTS is shown in Figure 2-22.

The optimization of the HTS for the baseline plant design includes optimizing the field piping cost and the steam generator conceptual design. The optimization of the field piping is based on selecting the field configuration, the riser and downcomer configuration, and the pipe diameter in each section of the main headers and the receiver headers.

The location of the EPGS in the collector field should have only a small influence on the total HTS pipe length. However, locating the EPGS at the center of the field should minimize the diameter of each pipe run. Therefore, this configuration was selected for the parametric analysis. Estimates were prepared, based on anticipated salt velocities, of the total weight of both the main and receiver headers for field configurations of 10 to 20 receiver rows. A capital cost optimum was initially found for a field layout with 16 rows; therefore, this configuration was selected for the parametric analysis. A brief economic comparison was made between the following receiver header designs: (1) locating the receiver headers on the ground, with risers and downcomers to each receiver module; and (2) locating the receiver headers adjacent to the receiver modules, with a single riser and downcomer for each receiver row. The latter arrangement was shown to be considerably less expensive, and was selected for the parametric analysis.

FIGURE 2-22 HEAT TRANSPORT SUBSYSTEM FLOW DIAGRAM

The optimization of the field piping diameters was based on selecting the line size in each section of both the main and receiver headers which would minimize the total cost of the section. The total cost was composed of the following four components:

- Pipe material and installation cost. Based on preliminary materials considerations and pressure drops, Incoloy 800, Schedule 10 has been selected for the hot salt piping and carbon steel, Schedule 40 has been selected for the cold salt piping. In the conceptual design, the use of less expensive materials for the hot salt piping was investigated.
- Insulation material and installation cost. The insulation considered for the field piping was calcium silicate, with an aluminum jacket for weather protection. Optimum insulation thicknesses ranged from 51 mm (2 inches) for the cold receiver headers to 102 mm (4 inches) for the hot main headers.
- Incremental collector field cost (to compensate for pipe heat losses).
- Incremental collector field and EPGS cost (to compensate for salt pump power requirements).

The optimization of the pipe insulation thickness is carried out first. The costs of the optimum insulation thicknesses, and the corresponding heat losses, are inputs to the pipe diameter optimization.

The results of the line size optimization, as shown in Table 2-26, define the HTS for the baseline plant design, and provide the basis for the HTS parametric cost model. The pipe diameters shown apply to various sections within each header.

Table 2-26

LINE SIZE OPTIMIZATION RESULTS

Line	Section Diameter, mm (inch)
Hot main header	152 (6), 203 (8), 254 (10)
Cold main header	203 (8), 254 (10), 305 (12), 356 (14)
Hot receiver header	51 (2), 76 (3), 102, (4)
Cold receiver header	76 (3), 102 (4)

The conceptual design of the steam generator includes separate shell-and-tube heat-exchangers for the preheater, boiler, superheater, and reheater. A conventional steam drum is located between the boiler and the superheater. The design is based on an 11 K (20°F) pinch point temperature difference in the preheater and a 56 K (100°F) approach temperature difference between the salt and steam temperatures in the superheater and reheater. Relative to the other systems within the plant, the \$3 million cost of the steam generator is small. Therefore, the present conceptual design, outlined in Table 2-27, has been used for the baseline plant design and the HTS parametric cost model. The parametric analysis for optimizing the pinch point and the salt-steam approach temperatures was performed in Task 3 (selection of the commercial system configuration).

To provide an input to the systems analysis, which selected the optimum throttle steam pressure and final feedwater temperature for the EPGS, a parametric cost model for the HTS was developed. This model was based on early design estimates, which specified that the salt flow rate, per foot of receiver length, was a constant. The salt flow rates used in the model included the effect of net EPGS efficiency, and the change in salt temperature between the inlet and the outlet of the receivers. The exponential cost functions of the model components are:

- Field Piping

$$\begin{aligned} \text{EPGS main headers cost} = & \$514,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{0.50} + \\ & \$1,446,000 \left(\frac{\text{WID}}{\text{WSID}} \right)^{0.70}, \end{aligned}$$

where

WSI = the cycle salt flow rate on the main headers, between the EPGS and the ESS

WSID = the baseline plant design salt flow rate in the main headers between the EPGS and the ESS.

(The first term is the cold main header cost, and the second term is the hot main header cost.)

Table 2-27

STEAM GENERATOR FOR THE BASELINE PLANT DESIGN

Section	Number of Shells	Dimensions L x D, m (in)	Materials		Duty MW (10 ⁶ Btu/h)	Area Per Shell m ² (ft ²)
			Shell	Tube		
Preheater	2	10.67 x 0.97 (420 x 38)	Carbon steel	Carbon steel	67.2 (229.2)	750 (8,100)
Boiler	2	8.74 x 1.07 (344 x 42)	Carbon steel	347 Stainless steel	67.8 (231.3)	540 (5,800)
Steam drum	1	3.74 x 0.91 (344 x 36)	Carbon steel	- -	- -	- -
Superheater	1	9.20 x 1.32 (362 x 52)	Incoloy 800	Incoloy 800	73.7 (251.4)	950 (10,200)
Reheater	1	4.17 x 1.37 (164 x 54)	Incoloy 800	Incoloy 800	43.4 (148.0)	570 (6,100)

$$\begin{aligned} \text{EPGS receiver headers cost} &= \$2,298,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{1.325} + \\ &\quad \$3,898,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{1.50}, \end{aligned}$$

(The first term is the cold receiver header cost, and the second term is the hot receiver header cost, for the receiver headers between the EPGS and the ESS.)

$$\begin{aligned} \text{ESS main headers cost} &= \$546,000 \left(\frac{\text{OR}}{8} \right) \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.50} + \\ &\quad \$1,060,000 \left(\frac{\text{OR}}{8} \right) \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.70}, \end{aligned}$$

where

WSO = the cycle salt flow rate in the main headers outside the ESS.

WSOD = the baseline plant design salt flow rate in the main headers outside the ESS.

OR = the number of receiver rows outside the ESS (a function of the solar multiple).

(The first term is the cold main header cost, and the second term is the hot main header cost.)

$$\begin{aligned} \text{ESS receiver headers cost} &= \$287,200 (\text{OR}) \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{1.325} + \\ &\quad \$487,200 (\text{OR}) \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{1.50}, \end{aligned}$$

(The first term is the cold receiver header cost, and the second term is the hot receiver header cost, for the receiver headers outside the ESS.)

- Salt Pumps

$$\text{EPGS cold salt pump cost} = \$297,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{0.80}$$

$$\begin{aligned} \text{ESS cold and hot salt pump cost} &= \$453,000 \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.80} + \\ &\quad \$240,000 \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.80} \end{aligned}$$

- Salt Inventory

$$\text{HTS salt inventory cost} = \$499,000 \left(\frac{\text{WSI}}{\text{WSID}} \right) \text{SM},$$

where

SM = the solar multiple.

- Steam Generator

$$\text{Preheater cost} = \$228,000 \left(\frac{\text{DUTY}}{\text{LMTD}} \right)^{0.60}$$

where

DUTY = the cycle preheater duty in MWth

LMTD = the cycle preheater log mean temperature difference in K.

(DUTY and LMTD are calculated from the cycle heat balance.)

$$\text{Boiler cost} = \$308,000 \left(\frac{\text{DUTY}}{\text{LMTD}} \right)^{0.65}$$

$$\text{Steam drum cost} = \$500,000$$

$$\text{Superheater cost} = \$279,000 \left(\frac{\text{DUTY}}{\text{LMTD}} \right)^{0.65}$$

$$\text{Reheater cost} = \$359,000 \left(\frac{\text{DUTY}}{\text{LMTD}} \right)^{0.65}$$

The results of the parametric analysis for the HTS, showing the HTS cost as a function of final feedwater temperature for 13.8 MPag (2000 psig) and 16.5 MPag (2400 psig) throttle steam conditions, are shown in Figure 2-23. The rationale for selecting the throttle steam conditions and final feedwater temperature is outlined in Section 2.6.

The solar multiple will also have an effect on the cost of the HTS. As the solar multiple is increased, the collector field and the length of the field piping will increase. The influence of the solar multiple on the cost of the HTS was analyzed only for the baseline plant design. The results of the analysis are shown in Figure 2-24. These data were analyzed by SRI to determine the optimum solar multiple.

FIGURE 2-23 HTS COST AS A FUNCTION OF THROTTLE STEAM PRESSURE AND FINAL FEEDWATER TEMPERATURE

FIGURE 2-24 HTS COST AS A FUNCTION OF SOLAR MULTIPLE

2.5 Energy Storage Subsystem (ESS)

As outlined in the requirements definition document, the ESS includes the thermal storage tanks, interconnecting pipes, and the ullage control system. The parametric analysis for the ESS included four variables:

- The method of thermal storage.
- The location of the ESS within the collector field.
- The influence on the cost of the ESS of the final feedwater temperature and throttle steam pressure of the EPGs.
- The influence of the solar multiple on the cost of the ESS for the baseline plant design.

Two methods of thermal storage were considered in the parametric analysis: thermocline storage and N+1 hot and cold tank storage.

The thermocline system uses a single tank to store the hot and cold salt. Hot salt is stored at the top of the tank while cold salt is stored at the bottom. Mixing is prevented by the density difference between the hot and cold fluids. The advantage of this system is that it requires the minimum size tank to store all the hot and cold salt. Two disadvantages of this system have been identified:

- The operating thickness and stability of the thermocline layer are not well known.
- An auxiliary storage tank is required for storing salt from the receivers and the HTS when the collector field is drained during an extended shutdown.

The second method of thermal storage involves N+1 hot and cold salt tanks. In this method, N tanks are required to hold the thermal storage inventory. The additional tank is used to store the salt from a hot or cold tank when it is emptied. The additional tank will also be emptied during the day as the storage salt is heated in the heliostat field, or it will be emptied at night as the storage salt is used in the EPGs. Two distinct advantages of this system are apparent:

- The storage tanks have either hot or cold salt in them; a thermocline system is not required.
- The additional tank may be used as the auxiliary salt storage tank during extended shutdowns.

The disadvantage of this system is that the tank cost of the N+1 tanks will be greater than that of the thermocline and auxiliary storage tanks. The selection of the thermal storage system is discussed later in this section.

The second component of the ESS parametric analysis was concerned with the location of the storage tanks. Two cases have been considered. The first case places the storage tanks adjacent to the EPGS, at the center of the heliostat field. The second case places the storage tanks in two locations. Each location is four receiver rows removed from the EPGS, and is adjacent to the main salt headers. Two advantages of the dispersed location ESS tanks can be seen. First, the portion of the main headers running from the ESS to the EPGS can be sized for a salt flow rate corresponding to a solar multiple of 1.0, rather than the 2.1 of the baseline plant design. Second, hot and cold salt pumps will be located at the ESS tanks. This will have the effect of reducing the throttling losses on the receiver headers close to the EPGS, thereby reducing the auxiliary power requirements of the EPGS. The principal disadvantage of the dispersed location ESS is that, although the total storage volume is the same as in the nondispersed case, twice the number of tanks are required, which will increase the cost of the ESS. In the parametric analysis, capital cost estimates were prepared for a number of cases, including central and dispersed thermocline tanks, and central and dispersed N+1 tank systems using values of 1, 2, and 4 for N. Among the N+1 options, the tank arrangement with N equal to 2 was found to have the lowest capital cost. This tank arrangement was then compared with the thermocline tank systems. Figure 2-25 shows the results of the study, including the cost of the tanks, increased main header costs, and increased plant costs (to compensate for additional pumping requirements). The dispersed location thermocline ESS was selected for the parametric analysis, because it had the lowest capital cost.

FIGURE 2-25 SELECTION OF THE TYPE AND LOCATION OF THE ESS BASELINE PLANT DESIGN

The tank design considered in the analysis is based on information presented in Ref. 2-2. Specifically, the tanks are constructed of carbon steel with 6 inches of ceramic fiber internal insulation. The insulation is porous, yet sufficiently dense (approximately 0.19 g/cm^3 or 12 lb/ft^3) to prevent convection within the insulation. Molten and solid salt within the insulation provides the majority of the thermal resistance to heat transfer. The use of internal insulation provides two design advantages. First, the tank walls operate at temperatures near ambient, thus minimizing thermal stresses within the tank due to changing salt temperatures. Second, the low tank wall temperatures means that carbon steel, rather than the more expensive alloy steel, can be used in the tank construction. The major potential disadvantage of this type of storage tank, the long-term stability of the ceramic fiber insulation, was more closely addressed during the conceptual design. The tank designs considered in the parametric analysis are based on a height-to-diameter ratio of 1.0, with tank diameters ranging from 10.4 m (34 ft) to 22.0 m (72 ft). The range of tank diameters corresponds to solar multiples of from 1.4 to 2.9. Large storage tanks, such as those required for the ESS, are generally constructed with a height-to-diameter ratio of 0.5 to 1.0. For the analysis, a ratio of 1.0 was selected to minimize the thermocline volume within the tank.

To provide an input to the system analysis, which selected the optimum throttle steam pressure and final feedwater temperature for the EPGS, a parametric cost model for the dispersed-location thermocline-ESS was developed. The exponential cost functions of the model components are:

$$\text{Thermocline tank cost} = \$943,800 \left[\left(\frac{\text{DTD}}{\text{DT}} \right) \left(\frac{\text{CED}}{\text{CE}} \right) \left(\frac{\text{SM}-1.0}{\text{SMD}-1.0} \right) \right]^{0.68}$$

where

DTD = the baseline plant design salt change in temperature through the steam generator

DT = the cycle salt change in temperature through the steam generator

CED = the baseline plant design net cycle efficiency

CE = the cycle net cycle efficiency

SM = the cycle solar multiple

SMD = the baseline plant design solar multiple.

$$\text{Auxiliary tank cost} = \$373,200 \left[\left(\frac{DTP}{DT} \right) \left(\frac{CED}{CE} \right) \left(\frac{SM-1.0}{SMD-1.0} \right) \right]^{0.68}$$

$$\text{Salt inventory cost} = \$5,748,000 \left[\left(\frac{DTP}{DT} \right) \left(\frac{CED}{CE} \right) \left(\frac{SM-1.0}{SMD-1.0} \right) \right]$$

Note that the salt inventory costs, based on a delivered price of 4.2¢/kg (11¢/lb), dominate the cost of the ESS. The results of the parametric analysis for the ESS, showing ESS cost as a function of final feedwater temperature for 13.8 MPag (2000 psig) and 16.5 MPag (2400 psig) throttle conditions, are shown in Figure 2-26.

In addition to throttle pressure and final feedwater temperature, the solar multiple will also affect the cost of the ESS. As the solar multiple is increased, the collector field, the number of hours of storage, and the capacity of the ESS will increase. As with the HTS parametric analysis, the influence of the solar multiple on the cost of the ESS was analyzed only for the baseline plant design. The results of the analysis are shown in Figure 2-27. These data were used as an input for the systems analysis to determine the optimum solar multiple.

Although the thermocline system selected for the parametric analysis has the lowest capital cost, it was not the appropriate choice for the conceptual design. There are a number of operating details in the thermocline system which are not well understood. These include the stability of the thermocline layer in the presence of eddy currents within the salt; and the heat transfer from the hot to the cold salt, due to conduction along the tank walls and through the thermocline layer. However, the operation of the N+1 tank system can be accurately defined, and its selection will increase the plant capital cost by less than \$500,000. For these reasons, it was decided that the choice of the thermocline system should be made only tentatively, and that these issues be addressed more closely in Task 3 (selection of the commercial system configuration).

FIGURE 2-26 ESS COST AS A FUNCTION OF THROTTLE STEAM PRESSURE AND FINAL FEEDWATER TEMPERATURE

FIGURE 2-27 ESS COST AS A FUNCTION OF SOLAR MULTIPLE

2.6 Electrical Power Generating Subsystem (EPGS)

The EPGS includes a tandem-compound, two-flow, reheat turbine-generator with individual high, intermediate, and low pressure sections, a water-cooled condenser, seven stages of feedwater heating, a deaerator, condensate pumps, boiler feedwater pumps, mechanical draft wet cooling towers, and circulating water pumps. The seven stage feedwater heating selection was based on the throttle and reheat steam conditions and a preliminary economic analysis. A flow diagram for the EPGS is shown in Figure 2-28.

In the effort to determine the optimum operating conditions for the EPGS, computer heat and material balances were run for all combinations of the parameters shown in Table 2-28.

Table 2-28

TURBINE PARAMETERS USED FOR HEAT AND MATERIAL BALANCES

Throttle Pressure MPag (psig)	Throttle and Reheat Steam Temperature, K (°F)	Final Feedwater Temperature, K (°F)
11.0 (1600)	672 (750)	450 (350)
13.8 (2000)	728 (850)	478 (400)
16.5 (2400)	783 (950)	506 (450)
	839 (1050)	533 (500)

Plant auxiliary loads were calculated for each of the cycle combinations. The auxiliary loads that were considered included the following:

- Salt circulation pumps
- Boiler feedwater pumps
- Condensate pumps
- Cooling tower fans

FIGURE 2-28 ELECTRICAL POWER GENERATING SUBSYSTEM (EPGS) FLOW DIAGRAM

- Circulating water pumps
- Heliostat drive motors
- Master control system
- High voltage ac and miscellaneous plant loads.

Based on calculated values for the cycle heat input, the net cycle efficiencies were determined for each of the EPGS combinations.

Receiver efficiency calculations show only a 1.1% decrease in receiver efficiency as the peak salt temperature increases from 728 K (850°F) to 839 K (1050°F). However, the net EPGS cycle efficiency increases by approximately 6.1% as the salt to the steam generator increases in temperature from 728 K (850°F) to 839 K (1050°F). Therefore, it was agreed that the parametric analysis should concentrate on a peak salt temperature of 839 K (1050°F), and a corresponding throttle steam temperature of 783 K (950°F).

The choice of the throttle steam pressure and final feedwater temperature, however, is not readily apparent. To determine the optimum steam pressure and feedwater temperature, a parametric cost model was developed for the EPGS. The model includes the cost of the turbine-generator, condenser, feedwater heater train, and heat rejection system. The exponential cost functions of each of the model components are shown below. All of the costs shown below for the EPGS, HTS, and ESS are in 1978 dollars and include material, direct labor, and engineering costs at 17% of the direct costs. Distributables and contingency are not included.

$$\text{Turbine-generator cost} = \$9,000,000 + \$830,000 \left(\frac{TP - 9.65}{16.5 - 9.65} \right),$$

where

TP = the throttle steam pressure in MPag.

$$\text{Condenser cost} = \$497,000 \left(\frac{HR - 3600}{HRD - 3600} \right),$$

where

HR = the cycle heat rate in kJ/kWh

HRD = the baseline design cycle heat rate.

$$\text{Heat rejection system cost} = \$1,402,000 \left(\frac{\text{HR} - 3600}{\text{HRD} - 3600} \right) + \\ \$696,000 \left(\frac{\text{HR} - 3600}{\text{HRD} - 3600} \right)^{0.5}$$

(The first term is the cooling tower cost and the second term is the circulating water pipe cost.)

$$\text{Feedwater train cost} = \$315,000 \left(\frac{\text{QFD}}{\text{QFW}} \right)^{0.65} + \$109,000 \left(\frac{\text{WF}}{\text{WFD}} \right)^{0.25} + \\ \$543,000 \left(\frac{\text{WF}}{\text{WFD}} \right)^{0.3} + \$970,000 ,$$

where

QFW = the cycle feedwater heater duty

QFWD = the feedwater heater duty for the baseline plant design

WF = the cycle feedwater flow rate

WFD = the feedwater flow rate for the baseline plant design

(The first term is the feedwater heater train cost. The second term is the feedwater pump cost. The third term is the feedwater piping cost. The fourth term is the cost of auxiliary subsystems which support feedwater system operation.)

Preliminary results showed that the 11.0 MPag (1600 psig) cycles are not competitive with the 13.8 MPag (2000 psig) and 16.5 MPag (2400 psig) cycles. Therefore, the parametric analyses for the EPGS, HTS, and ESS concentrated on the 13.8 MPag (2000 psig) and 16.5 MPag (2400 psig) cycles. The parametric analysis results for the EPGS, showing its cost as a function of the final feedwater temperature for 13.8 MPag (2000 psig) and 16.5 MPag (2400 psig) throttle conditions, are graphed in Figure 2-29. These data were used as an input to the systems analysis to determine the optimum EPGS operating conditions.

The solar multiple also has an effect on the cost of the EPGS. As the solar multiple is increased, the collector field will increase in size. The auxiliary power requirements for the salt pumps and, therefore, the capacity of the EPGS will be increased. To keep the number of parametric cases to a manageable level, the influence of the

FIGURE 2-29 EPGS COST AS A FUNCTION OF THROTTLE STEAM PRESSURE AND FINAL FEEDWATER TEMPERATURE

FIGURE 2-30 EPGS COST AS A FUNCTION OF SOLAR MULTIPLE

solar multiple on the cost of the EPGS was analyzed only for the base-line plant design EPGS. The results of the analysis are shown in Figure 2-30. These data were used to determine the optimum solar multiple.

Two final EPGS operating parameters were optimized in Task 3: condenser pressure and preheater pinch point temperature. These parameters were optimized in Task 3 because their effect on electric energy cost was estimated as being small.

2.7 Master Control Subsystem (MCS)

An overall view of the master control subsystem (MCS) is shown in Figure 2-31. The central computer is the focal point for all of the control operations, but the detailed operation of the power plant and storage, the heliostats, and the receivers is managed by controllers directly associated with the individual subsystem elements. The general complexity of the functions performed by the central computer dictates that the computer should be digital. The advent of single-chip microprocessors makes digital techniques the logical choice for the controllers required at the local level. The compatible choice for intermediate levels is likewise digital.

The configuration of the receivers and the heliostat array for the entire line-focus power system is determined primarily by the optical, mechanical, and thermodynamic requirements of the system. This determines the points in the system at which local control devices are required. For this reason, the master control subsystem parametric analysis is principally related to parameters within the master control subsystem itself. Since the overall configuration of the line-focus power system has a distributed and modular character, the parametric analysis readily showed that the MCS should have these same distributed and modular characteristics. Therefore, the MCS parametric analysis became primarily a hierarchical problem; that is, the analysis of input/output tradeoffs at each level in the subsystem and the determination of the optimal control sophistication at each level. This included an analysis at each control point, and at each interface in the various levels of the hierarchy, to determine just how "smart" each should be, and what amount of autonomy should be included.

Sun tracking is accomplished by simultaneously issuing a single angular increment value to all the heliostats. This means that the computational power of the heliostat local controller is modest, so a simple, low-speed, 8-bit microcomputer (preferably CMOS) is an adequate implementation means for each local controller. A nonvolatile memory (one that does not lose its memory if the power is turned off) will be

FIGURE 2-31 MASTER CONTROL SUBSYSTEM

included in the microcomputer to "remember" the absolute position of each heliostat. The absolute position data are derived from an incremental encoder in the heliostat elevation mechanism. This is a cost saving method, compared to using an absolute position optical shaft encoder. By the mid-1980s it is reasonable to expect that a nonvolatile semiconductor memory will be available for this purpose, as an integral part of a single-chip microcomputer (also called a microcontroller).

A nonvolatile memory technology that is currently receiving considerable attention in the technical press is the magnetic bubble memory. Just now emerging as a commercial product, it is expected to find widespread use in microcomputer applications. However, it requires several discrete semiconductor parts to provide internal control functions and interfacing, so it probably will not be cost effective for the heliostat local controller.

Adding redundancy is a useful method of increasing the reliability of a system. This has been considered for the MCS. In particular, redundant block controllers with redundant data paths from the central computer would enhance reliability. However, present indications are that this type of redundancy will not be required. Instead, replacement redundancy and error detection check bits in a command/response protocol will be used.

Replacement redundancy at the block controller level, and lower, is feasible since a brief loss of control is not catastrophic. It is also economical, because all controllers within a given class are identical--a consequence of the modularity of the MCS. Two spare block controllers should provide an adequate replacement backup. A larger number of replacements will be required for the backup of the simpler heliostat controllers and receiver controllers. In the central computer, there is an array controller that will probably have the same (or similar) hardware implementation as the block controllers, and replacement redundancy will also be used for it, either on a share basis with the block controllers or independently.

The principal individual computer within the central computer complex is duplicated in hardware by an auxiliary computer. Backup operation is achieved by switching from one computer to another, while still maintaining essential operating control. (See Section 4.7 on conceptual design.) A battery-powered uninterruptible power source (UPS) is not deemed necessary for the central computer, since brief control loss at the central computer level can be tolerated by the line-focus power system. Furthermore, manually assisted operation at the block and local controller level is provided for.

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3.0 COMMERCIAL SYSTEM SELECTION

3.1 Introduction

The commercial system design was selected using the same computer program that optimized the subsystems on the basis of 30 year levelized BBEC, described in Section 2.1, Parametric Analysis Methodology. The commercial system design was selected by an iterative process during the subsystem parametric analysis. The BBEC was the design criterion. Further minor system changes were made during Task 4 (plant conceptual design).

The development of conceptual designs and cost estimates for each combination of HS, RS, EPGS, HTS, ESS, and MCS used in the parametric analysis would be prohibitively expensive and time consuming. An alternative approach was required to determine the performance and cost figures for each system combination. The approach used had four steps:

- Estimates of the results of the parametric analysis were made to select a baseline design.
- A reasonably detailed conceptual design and cost estimate was prepared for a baseline plant.
- Performance estimates for the various combinations were made by modifying portions of the baseline plant design.
- Cost estimates for the system combinations were made by applying exponential cost functions to portions of the baseline plant estimate.

The performance and cost estimates for the system combinations became inputs to the overall plant computer model. The model determined the combination of system parameters that minimized BBEC. These selected system parameters were then used in the conceptual plant design.

A number of possible design changes in the CS, RS, EPGS, HTS, ESS and MCS were identified as potential improvements to the system during the parametric analysis. These changes were addressed in Task 3 and include:

- Increase in the throttle steam temperature from 783 K (950°F) to 811 K (1000°F).
- Change in the throttle steam pressure from 13.8 MPag (2000 psig) to either 12.4 MPag (1800 psig) or 16.5 MPag (2400 psig).
- Revision of the HTS cost model from one based on a constant salt flow rate, per meter of receiver length, to one based on a constant change in salt enthalpy, per hour, per meter of receiver length.
- Change in hot-salt pipe material from Incoloy 800 to 316 stainless steel.
- Change in tentative choice of the thermal storage design from thermocline tanks to N+1 hot and cold tanks.
- Change in the condenser pressure from 10.1 kPa (3.0 inches HgA) to 8.45 kPa (2.5 inches HgA) or 11.8 kPa (3.5 inches HgA).
- Change in the preheater pinch point temperature difference from 11 K (20°F) to 5.6 K (10°F)
- Change in the number of receiver rows from 16 to 10.

The results from Task 2 (parametric analysis) together with the results from Task 3 (selection of the commercial system configuration) outlined the design parameters for Task 4 (conceptual design).

3.2 System Analysis

3.2.1 Electrical Power Generating Subsystem (EPGS)

The possible design changes in the EPGS included the increase in the throttle steam temperature from 783 K (950°F) to 811 K (1000°F) and the change in the throttle steam pressure from 13.8 MPag (2000 psig) to 12.4 MPag (1800 psig) or 16.5 MPag (2400 psig).

3.2.1.1 Increase in the Throttle Steam Temperature

Previous solar power plant studies have limited the throttle steam temperature to 783 K (950°F) due to thermal cycling limitations based on the recommendation of turbine-generator manufacturers. The receivers in these studies have been cooled by the feedwater flow from the EPGS. The flow is controlled to maintain a constant steam pressure upstream of the turbine control valves. During a cloudless day (with an

instantaneous solar multiple greater than 1.0) the turbine control valves will almost be fully open and the turbine inlet temperature will remain constant. However, during a day with passing clouds and a changing solar multiple the turbine control valves will throttle the main steam flow, as the solar insolation and turbine output change. This, in turn, will change the turbine inlet steam temperature. It is this short-term temperature cycling which has limited main steam temperatures to 783 K (950°F).

After the decision to select 783 K (950°F) as the baseline throttle and reheat temperature conditions for this study, additional study of the SRI concept showed that short-term temperature cycling at the turbine inlet should not be present. In the SRI design, high-temperature salt from the receiver is directed to the EPGS for power generation and to the ESS for storage. A constant flow rate of salt to the steam generator can be maintained by taking salt from the receivers and from the salt storage tanks. With a constant heat input to the EPGS, the generator load and turbine inlet temperature will each remain constant. Therefore, the turbine temperature is expected to cycle only once each day, since the EPGS is started in the morning and shut down at night. Bechtel research, on the subject of peaking turbine-generators that cycle once each day, has shown that 16.5 MPag/811 K/ 811 K (2400 psig/1000°F/1000°F) cycles are technically feasible, and are now in commercial operation. Therefore, the selection analysis was expanded to include the 811 K (1000°F) turbine cycles.

3.2.1.2 Change in the Throttle Steam Pressure

After the selection of the 13.8 MPag (2000 psig) cycle during the parametric analysis, research on the subject of commercially available 100 MW_e reheat turbine-generators showed that the 13.8 MPag (2000 psig) turbine is not a standard design. Even if the 2000 psig cycle proves to be the eventual economic optimum for the line-focus concept, it cannot be predicted with certainty that these units will be manufactured on a commercial scale by 1985 or 1990. Therefore, it was decided that the

final selection of the throttle steam pressure for the conceptual design should be based on standard turbine designs.

The closest standard throttle steam pressures to 13.8 MPag (2000 psig) are 12.4 MPag (1800 psig) and 16.5 MPag (2400 psig), so the emphasis in the analysis shifted to a comparison of the 12.4 MPag (1800 psig) and 16.5 MPag (2400 psig) cycles. Fortunately, the performance and cost of the plant are only mildly sensitive to changes in the throttle steam pressure. Therefore, it was agreed that the results from the parametric analysis for the 13.8 MPag (2000 psig) cycle would apply equally well to the 12.4 MPag (1800 psig) cycle. Performance and cost data for the 16.5 MPag (2400 psig) cycle had already been developed in the parametric analysis.

Preliminary results from the parametric analysis and the selection of the commercial system configuration indicated that the optimum final feedwater temperature was 506 K (450°F) and that the 811 K/811 K (1000°F/1000°F) cycle was more economic than the 783 K/783 K (950°F/950°F) cycle. Therefore, to select the throttle steam pressure for the conceptual design, parametric performance and cost data were developed for the 12.4 MPag/811 K/811 K (1800 psig/1000°F/1000°F) and 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) cycles with final feedwater temperatures of 506 K (450°F). Although the parametric analysis did not determine the optimum final feedwater temperature for the 811 K/811 K (1000°F/1000°F) cycles, a feedwater temperature of 506 K (450°F) is typical for a 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) fossil-fired plant. The performance and cost data for the 12.4 MPag/811 K/811 K (1800 psig/1000°F/1000°F) and 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) cycles are summarized in Table 3-1, which shows the change in cost from the baseline plant design.

3.2.2 Energy Storage Subsystem (ESS)

After the selection of the thermocline storage system for the parametric analysis, a heat transfer model for the salt thermocline was

Table 3-1

811 K/811 K (1000°F/1000°F) EPGS PARAMETRIC ANALYSIS

<u>Throttle Steam Pressure, MPag (psig)</u>	<u>Net Cycle Efficiency (%)</u>	<u>Change in EPGS Cost (\$1000)</u>
12.4 (1800)	39.02	-681
16.5 (2400)	39.64	-131

developed. The model was used to estimate the amount of heat transfer from the hot to the cold salt within the tank. To compensate for this heat transfer, the size of the collector and receiver systems and the capacity of the salt pumps must be increased over that required for a comparable N+1 hot and cold tank system. The cost for the increased system capacities must then be added to the thermocline system cost in order to obtain an accurate comparison of ESS methods.

The heat transfer model has two components. The first is conduction heat transfer through the thermocline. The solution is modeled by a transient analysis for semi-infinite slabs. The second component is convective heat transfer to the storage tank walls and conduction along them. The mathematical solution is based on a model presented in a Sandia Laboratories thermocline test facility report (Ref. 3-1).

Assuming that a complete thermocline charge-discharge cycle requires 24 hours, and that the thermocline thickness at the start of the cycle is 0.3 m (1 ft), the daily heat transfer within the storage tanks is 7.5 MWh. The cost of the extra collectors and the receivers required to supply this additional 7.5 MWh per day is approximately \$260,000. To collect this incremental energy, the capacity of the salt pumps must be increased by 8 kW_e, and the cost of the plant increased by \$10,000. Also, the salt within the thermocline volume represents a cost liability, since it cannot be used for power generation. Assuming a maximum thermocline thickness of 0.46 m (1.5 ft), the cost of the salt contained within this volume is \$105,000. The total costs for the

thermocline system and a comparable N+1 tank system with N equal to 2, are summarized in Table 3-2.

Table 3-2

ENERGY STORAGE SUBSYSTEM COST COMPARISONS

Component	Cost (\$1000)	
	Thermocline Tanks	N+1 Tanks
Salt tanks	1,515	1,920
Increased costs		
Collectors and receivers	260	--
Plant capacity	10	--
Salt inventory	105	--
Total	1,890	1,920

The small cost benefit may not justify the development risk associated with the thermocline system. Therefore, the N+1 tank system was selected by SRI for the conceptual plant design. ESS costs for the 12.4 MPag/811 K/811 K (1800 psig/1000°F/1000°F) and 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbine cycles, with the final cycles having final feedwater temperatures of 506 K (450°F), are summarized in Table 3-3. The costs are based on the N+1 tank system and show the change in cost from the baseline plant design.

Table 3-3

811 K/811 K (1000°F/1000°F) ESS PARAMETRIC ANALYSIS

Throttle Steam Pressure MPag (psig)	Change in ESS Cost (\$1000)
12.4 (1800)	+279
16.5 (2400)	+195

These data were used to develop the final operating parameters for the EPGS in the conceptual design.

3.2.3 Heat Transport Subsystem (HTS)

The potential design changes in the HTS include a change in the hot salt pipe material and a revision to the HTS cost model. This revised cost model influences the final selection of the EPGS operating parameters.

3.2.3.1 Hot Salt Pipe Material

To explore the possibility of using 316 stainless steel as an alternate to the Incoloy 800 hot salt pipe material selected in the parametric analysis, a number of studies (Refs. 3-2 through 3-6) were reviewed in Task 3. These studies dealt with the corrosive effects of draw salt on 316 stainless steel at temperatures ranging from 756 K (900°F) to 867 K (1100°F). Although the draw salt sensitized the stainless steel, intergranular corrosion and weight loss through flaking were not present. The studies concluded that nonstressed 316 stainless steel is compatible with stagnant draw salt at 839 K (1050°F). On this basis, it was felt that 316 stainless steel would be suitable for use as the hot salt pipe material in the conceptual design.

However, this decision must be regarded as tentative. The final selection of this stainless steel for use in an actual powerplant should depend on additional laboratory tests which will measure the effects of salt velocities, daily pressures, and temperature cycles.

3.2.3.2 Heat Transport Subsystem

An approximate cost model for the HTS was developed in the parametric analysis, based on a constant salt flow rate per meter of receiver length. The model was refined in Task 3, with a more realistic change in salt enthalpy per hour, per meter of receiver length. The model variables are identical to those of the parametric analysis model. The revised cost model is summarized as follows:

- Field Piping

$$\text{EPGS main header cost} = \$514,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{0.50} +$$

$$\$1,446,000 \left(\frac{\text{WSI}}{\text{WSID}} \right)^{0.70}$$

$$\text{EPGS receiver headers cost} = \$2,298,000 \left[\left(\frac{\text{CED}}{\text{CE}} \right) \left(\frac{\text{DT}}{\text{DTD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right) \right]^{1.325} +$$

$$\$3,898,000 \left[\left(\frac{\text{CED}}{\text{CE}} \right) \left(\frac{\text{DT}}{\text{DTD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right) \right]^{1.50}$$

$$\text{ESS main header cost} = \$546,000 \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.50} +$$

$$\$1,060,000 \left(\frac{\text{WSO}}{\text{WSOD}} \right)^{0.70}$$

$$\text{ESS receiver headers cost} = \$2,608,000 \left[\left(\frac{\text{CED}}{\text{CE}} \right) \left(\frac{\text{DT}}{\text{DTD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right) \right]^{1.325} +$$

$$\$4,498,000 \left[\left(\frac{\text{CED}}{\text{CE}} \right) \left(\frac{\text{DT}}{\text{DTD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right) \right]^{1.50}$$

- Salt Inventory

$$\text{HTS salt inventory cost} = \$499,000 (\text{SM}) \left(\frac{\text{WSI}}{\text{WSID}} \right)^{0.50} \left(\frac{\text{CED}}{\text{CE}} \right)$$

The salt pump and steam generator (SG) cost models are identical to those of the parametric analysis.

HTS costs were developed for the 12.4 MPag/811 K/811 K (1800 psig/1000°F/1000°F) and 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbines cycles, with final feedwater temperatures of 506 K (450°F). Steam generator costs were based on a 28 K (50°F) approach temperature difference between the hot salt and main and reheat steam temperatures and a 11 K (20°F) pinch point temperature difference in the preheater. These costs are summarized in Table 3-4, which shows the change in cost from the baseline plant design.

Table 3-4

811 K/811 K (1000°F/1000°F) HTS PARAMETRIC ANALYSIS

Throttle Steam Pressure, MPag (psig)	Change in System Cost (\$1000)	
	HTS	SG
12.4 (1800)	+422	-439
16.5 (2400)	+235	+201

These data were used to develop the final operating parameters for the EPGS in the conceptual design.

3.2.4 Final Selection of the Turbine Cycle Operating Conditions

Based on the performance data outlined in Table 3-1, collector and receiver subsystem capacities and costs corresponding to the 12.4 MPag/811 K/811 K (1800 psig/1000°F/1000°F) and 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbine cycles were developed. Based on these costs, together with the EPGS, ESS, and HTS parametric costs, it was determined that there is approximately a \$460,000 capital cost advantage for the 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbine cycle. Therefore, this cycle was selected for the conceptual design.

Two final EPGS operating parameters were selected after the final throttle steam conditions were chosen. These parameters were the condenser pressure, and the preheater pinch-point temperature.

3.2.4.1 Condenser Pressure

Additional heat balances were run for the 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbine cycle, with condenser pressures of 8.45 kPa (2.5 inches HgA) and 11.8 kPa (3.5 inches HgA) to compare with the base case of 10.1 kPa (3.0 inches HgA). Parametric performance and cost data were developed for each cycle. The data are summarized in Table 3-5, which shows the change in cost from the baseline plant design.

Based on these performance data, heliostat and receiver capacities and costs were calculated for each condenser pressure. From these costs,

Table 3-5

CONDENSER PRESSURE PARAMETRIC ANALYSIS

Condenser Pressure, kPa (inches HgA)	Net Cycle Efficiency (%)	Change in System Cost (\$1000)		
		EPGS	HTS	ESS
8.45 (2.5)	39.79	+691	-86	-165
10.1 (3.0)	39.64	-131	-10	-146
11.8 (3.5)	39.44	-594	+93	-101

it was determined that the optimum condenser pressure remained unchanged at 10.1 kPa (3.0 inches HgA):

3.2.4.2 Preheater Pinch-Point Temperature

Parametric performance and cost data were developed for the 16.5 MPag/811 K/811 K (2400 psig/1000°F/1000°F) turbine cycle, with preheater pinch-point temperature differences of 5.6 K (10°F) and 11 K (20°F). Previous experience with steam generator design was used to select the initial 11 K (20°F) pinch-point temperature difference for the parametric analysis. The 5.6 K (10°F) pinch-point temperature difference was selected as a reasonable minimum for stable steam generator operation. The data are summarized in Table 3-6, which shows the change in cost from the baseline plant design.

Table 3-6

PREHEATER PINCH-POINT TEMPERATURE PARAMETRIC ANALYSIS

Pinch Point Temperature, K (°F)	Net Cycle Efficiency (%)	Change in Subsystem Cost (\$1000)		
		EPGS	HTS	ESS
5.6 (10)	39.70	-143	+175	+5
20	39.64	-131	-10	+195

Based on these performance data, collector and receiver capacities and costs were calculated for each pinch-point temperature. From these costs, it was determined that the optimum pinch-point temperature difference was 5.6 K (10°F).

3.2.5 Final Field Configuration

Based on the results of Task 3, the collector and receiver subsystem designs were revised from the parametric analysis as follows:

	<u>Parametric Analysis</u>	<u>Conceptual Design</u>
Solar multiple	2.10	2.49
Total receiver length, m (ft)	17,560 (57,600)	13,475 (44,200)
Collector rows per receiver row	20	24

The preliminary field configuration, based on the conceptual design parameters, had a width-to-length ratio of only 0.28. To increase this ratio, a modified field configuration was developed, based on 10 receiver rows. This latter arrangement, which has a width-to-length ratio of 0.67, was selected for the conceptual design for two reasons. First, a brief analysis showed that the revised layout had a slightly lower field piping cost. Second, and more importantly, each receiver row became longer. This will reduce the receiver cosine losses in the morning and evening. Therefore, it will also increase the annual energy collected by the field. It was felt that the change in the number of receiver rows would not have a substantial effect on the results of the parametric analysis, or on the selection of the commercial system configuration.

3.2.6 Final Operating Parameters for the Conceptual Design

From the Task 2 and Task 3 analyses, the following operating parameters for the conceptual design were selected:

- Electrical Power Generating System
 - Throttle steam pressure of 16.5 MPa (2400 psig)
 - Throttle and reheat steam temperatures of 811 K (1000°F)
 - Final feedwater temperature of 506 K (450°F)
 - Condenser pressure of 10.1 kPa (3.0 inches HgA)
- Energy Storage Subsystem
 - N+1 tank storage system, with N equal to 2
 - Dispersed location tanks
 - Thermal storage capacity equivalent to 750 MW_e h
- Heat Transport Subsystem
 - Hot salt pipe material of 316 stainless steel
 - Cold salt pipe material of A-106 Grade B carbon steel
 - Solar multiple of 2.49
 - Single riser and downcomer in each receiver row
 - Preheater pinch-point temperature difference of 5.6 K (10°F)
- Heliostat Subsystem
 - Augered caisson heliostat foundations spaced 12.2 m (40 ft) center-to-center
 - Caisson dimensions of 0.38 m (1.25 ft) diameter and 3.66 m (12 ft) deep
 - Number of heliostat rows per receiver row is 24
- Receiver Subsystem
 - Total receiver length of 13,475 m (44,200 ft)
 - Receiver rows extend 65 m (213 ft) beyond the end of the collector row
 - Tower height of 61 m (200 ft)
 - Tower spacing of 45.7 m (150 ft)
 - Tower foundation; requires an area of 15.2 m (50 ft) x 15.2 m (50 ft) [231.04 m² (2500 ft²)]
 - Salt inlet and outlet temperatures of 567 K (561°F) and 839 K (1050°F), respectively

3.3 System Tradeoff Parameters

The following 100-MW_e plant design specifications were established:

- 10 rows of collectors, of nominal length 1347 m (4420 ft) each
- 24 row of 3-m-wide heliostats, per 61 m (200 ft) high linear cavity receiver
- ~6 m (19.7 ft) spacing between heliostat foundations
- 273°C (550°F) salt into the receiver
- 551°C (1050°F) salt from the receiver
- 218°C (450°F) feedwater to the steam generator
- 13.8 KPa, 524°C (2000 psi, 1000°F) steam to turbine
- 3.6 KPa, 524°C (520 psi, 1000°F) reheat conditions in turbine
- three equally sized salt storage tanks, capable of hot or cold salt storage, for a system storage capacity of 750 MWh_e.

The resulting capacity factor is 0.6, on an average daily basis.

That is, the plant operates at 100 MW_e for 0.6 of the day. The levelized BBEC based on a heliostat cost of \$5/ft² is 61 mil/kWh_e. The largest remaining opportunity for cost reduction in the plant design is the heliostat cost, and its associated performance. Heliostat costs below \$5/ft² are being projected for the glass-fiber-reinforced concrete heliostat, with second-surface silvered glass on a fiber-reinforced-concrete substrate. The values are in 1980 dollars for the 80th plant, planned to begin operation in 1990.

Figure 3-1 shows the performance of the first and 80th plants in the terms of BBEC vs average daily capacity factor. For a given size storage system, the curves show a minimum in BBEC vs the average daily capacity factor. The curves were generated by selecting a design and changing only the length of the collector field. As additional collector area is needed, the electrical generation system is more fully utilized (the average daily capacity factor increases). Thus, the BBEC decreases as the capacity factor increases. At some point, more energy is collected by the increased collector size than can be used or stored. This excess energy is wasted, and the increased collector field thus increases cost without adding to capacity.

FIGURE 3-1 LEVELIZED BUSBAR ELECTRIC COST (BBEC)

An increase in the storage capacity of a plant allows operation at a larger capacity factor, and generally at a decreased $\overline{\text{BBEC}}$. The $\overline{\text{BBEC}}$ of the envelope decreases until a capacity factor in excess of 0.9 is reached.

This 100 MW_e plant design provides a solar central power system that can still be improved to give better cost and performance. The design shows a minimum in $\overline{\text{BBEC}}$ with respect to changes in nearly all of the current input parameters. Improvements in performance or cost variables should create improvements in the cost effectiveness of an optimized design. In addition, incremental start-up should significantly reduce these costs even further. (Ref. 3-7)

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- 3-7 Conceptual Design of Advanced Central Receiver Power System, Phase I, Martin Marietta Final Report, September 1978, EG-77-C-03-1724, p. II-303.

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4.0 CONCEPTUAL DESIGN

4.1 Introduction

4.1.1 HTLF Central Power System Description

The system designed by SRI International and its subcontractors is the high-temperature line-focus (HTLF) central power system (Figure 4-1). The HTLF system is analogous to the line-focus, parabolic cylinder "trough" collector/receiver, but is much larger. (Similarly, the point-focus, central-power "power tower" concept is analogous to the point-focus, paraboloid "dish" collector/receiver, but much larger.) The HTLF system retains the simplicity of the single-axis parabolic cylinder collector and requires less piping than a distributed system, enabling a reduction in heat loss, cost, and absorber-fluid pressure drop. The cavity receivers of the HTLF-system also significantly reduce the reflection and radiation losses associated with the visually exposed tube receiver of the smaller distributed line-focus system. The cavity provides for a higher stagnation temperature, and results in higher collection and system efficiencies. A schematic drawing of the system is shown in Figure 4-2.

The sun's rays are reflected from second-surface mirror panels mounted on the heliostat frame (Figure 4-3). The single-axis heliostats are each about 3 m (10 ft) wide and 48.8 (160 ft) long. Each heliostat is made of four 12.2 m (40 ft) modules, and is driven by a single four-bar linkage aiming mechanism. Each heliostat can rotate 195° for stowing, with the reflective surfaces facing downward at night, and during inclement weather. The heliostats are controlled by a central computer, through field controllers and local heliostat controllers.

In this configuration, each low-profile, large [148.6 m^2 (1600 ft^2)] single-axis heliostat is aimed by a single motor, and all heliostats move in synchrony. Each heliostat is focused to reduce the sun's image

FIGURE 4-1 100-MW_e HIGH-TEMPERATURE LINE-FOCUS (HTLF) CENTRAL RECEIVER SOLAR POWER SYSTEM

FIGURE 4-2 HIGH-TEMPERATURE LINE-FOCUS (HTLF) CENTRAL RECEIVER SYSTEM

from the 3 m (10 ft) width of the heliostat to the minimum image size for its distance from the receiver aperture.

The HTLF receiver is a horizontal cylindrical cavity 1.8 m (6 ft) in diameter with a 1.2 m (4 ft) aperture. The aperture faces downward to the north at an angle of about 40° to the horizontal (Figure 4-4). The top edge of the aperture has doors, the reflective inside surface of which act as secondary concentrators. At night, the doors close to conserve heat. The inside of the cavity is lined with horizontal heat-exchanger tubes, backed by fiber insulation. The heat absorber tubes are arranged to take advantage of once-through, multipass, counterflow heating of molten draw-salt by locating the incoming tubes near the aperture, with the final pass in the center, where the flux density is

FIGURE 4-3 LINE-FOCUS CENTRAL RECEIVER HELIOSTAT MODULE

FIGURE 4-4 HIGH-TEMPERATURE LINE-FOCUS (HTLF) CENTRAL RECEIVER

the greatest. Each section between towers has an independent modular heat absorber with manifolds that feed the final pass to common downcomers. The hot salt is delivered through insulated downcomers, and an insulated common horizontal pipe, directly to storage tanks and to the heat exchanger that generates steam for the turbine/generator. The power-conversion system uses a reheat steam cycle.

The function of the energy storage subsystem (ESS) is to buffer the electrical power generating subsystem (EPGS) from the results of excessive variations in insolation, and to extend the plant's generating capabilities into periods of low (or no) insolation. The EPGS uses commercial steam turbine technology.

The high-temperature line-focus receiver system has an inherently high reliability, for two main reasons: reliability of the proposed subsystems and redundancy. The redundancy inherent in the heliostat and heat-absorber subsystems also enhances this reliability.

The heliostats and controls are in parallel. One favorable characteristic of the HTLF system is that the use of modular receiver heat absorber units permits partial shutdown (and quick replacement of facility components) during the continuing operation of the rest of the system. The cavity receiver concept has two 23-m heat-absorber modules in each 46-m section between towers. Because these absorber modules have separate controls and feed pumps, the remaining absorbers can still deliver energy to the turbine (or to storage) in the event that one absorber module malfunctions. Figure 4-5 shows the plot plan.

Table 4-1 shows the baseline design for the 100 MW_e line-focus solar central power system.

4.1.2 System Performance

Figure 4-6 shows the system performance in terms of $\overline{BBEC}$ as a function of capacity factor for several different storage capacities. The 750 MW_{th} storage capacity indicates the performance of the plant selected for a capacity factor of 0.6. Figure 4-7 shows the detailed

POINT	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	A	B
FLOW lb/hr	4.74 x 10 ⁶	3.15 x 10 ⁶	1.59 x 10 ⁶	3.15 x 10 ⁶	1.59 x 10 ⁶	4.74 x 10 ⁶	4.74 x 10 ⁶	4.74 x 10 ⁶	790,000	790,000	7.06 x 10 ⁶	7.06 x 10 ⁶	722,000	722,000	722,000	722,000	661,000	661,000	21.2 x 10 ⁶	21.2 x 10 ⁶	2.37 x 10 ⁶	2.37 x 10 ⁶
TEMPERATURE F	1,050	1,050	1,050	030	836	836	650	548	548	1,050	548	1,050	450	640	640	1,000	647	1,000	108	84	1,050	548
PRESSURE psig	100							488	303	30	0.3	71				1,800	421	377			176	57

FIGURE 4-5 SRI LINE-FOCUS SOLAR CENTRAL POWER PLANT PROCESS FLOW DIAGRAM, 100 MW_e NET

Table 4-1

HTLF SOLAR CENTRAL POWER SYSTEM
(100 MW_e Conceptual Design)

Collector

12,192 m (40,000 linear ft)	} 6000 heliostats, 48.8 m (160 ft) long
24 rows of heliostats	
3 m (10 ft) width	
5.5 m (18 ft) row pitch	

Receiver

13,472 m (44,200 linear ft)
10 rows
1.22 m (4 ft) aperture
1.83 m (6 ft) diameter
218°C-274°C (450°-550°F) salt in
551°C (1,050°F) salt out
552 kg/m (350 lb/ft)
Selective surface ($\alpha_s = 0.9$; $\epsilon_{525^\circ\text{C}} = .06$)

Towers

61 m (200 ft) high
45.7 (150 ft) between towers

Heat Transport System

Line thermal losses 0.35 kW/m (0.108 kW/ft) 5.8 MW_{th} hot
0.16 kW/m (0.050 kW/ft) 2.7 MW_{th} cold

Heat Storage Subsystem

7.5 hour storage at 100 MW_e
3 tanks, each usable for hot or cold salt
Tank thermal loss
Hot: 0.33 MW_{th}
Cold: 0.17 MW_{th}

Electrical Power Generating Subsystem

551°C (1,050°F) salt in
274°C (550°F) salt out
218°C (450°F) 2,000 psi feedwater
524°C (1,000°F) steam inlet
3.59 MPa, 524°C (520 psi, 1,000°F) steam reheat
39.4 efficiency, including pumping and auxiliary losses

System Average Daily Capacity Factor

0.6

FIGURE 4-6 LEVELIZED BUSBAR ELECTRIC COST (BBEC)

system schematic, including the energy and mass balance for the operating mode, direct or from storage. Figures 4-8, 4-9, and 4-10 are "waterfall" charts showing the power losses at winter, spring-fall, and summer noon. Figure 4-11 shows the annual total energy losses.

System costs for the 80th plant are estimated in Table 4-2. These cost estimates are based on SRI subcontractor's current designs and practice; a learning factor was applied by SRI for the receiver, towers, trusses, and heat transport system.

The cost of the turbine-generator, steam generator, and heat storage system was based on current costs of turnkey operation. The costs of land and salt are as given in Table 4-2. The cost for the heliostats is based on mass-produced design at a rate of 25,000 heliostats per year.

Table 4-2

LINE-FOCUS SOLAR CENTRAL POWER SYSTEM
PARAMETRIC ANALYSIS DESIGN
(80th Plant Cost)

		Costs (\$ millions)	
		1st Plant	80th Plant
Design basis, 100 MW _e , 0.6 average daily capacity factor			
Collector field (6,600 heliostats)			
44,050-ft length			
24 rows per receiver			
10-ft width	@ \$5.05/ft ²	\$ 52.55	\$ 52.55
Receiver			
48,643-ft length total in 10 rows (extends beyond heliostats 210 ft)	@ \$750/ft (1st) @ \$542/ft (80th)	36.48	26.36
Towers and trusses			
200 ft high	@ \$208/ft (1st)		
150 ft span	@ \$132/ft (80th)	10.12	6.42
Heat transport system		13.53	13.53
Heat storage system	6 Tanks, 7.5 h	11.07	11.07
Steam generator		3.91	3.91
Turbine-generator		22.40	22.40
Land, 752 acres @ \$500/acre		.38	.38
Structures and improvement		3.9	3.9
Salt cost (\$4.04 included in HTS and ESS)	@ 15¢/lb		
Distributables		3.6	3.6
BOP		3.8	3.8
Direct Cost		161.74	147.92
Indirect Cost		32.27	15.94
Total Cost		\$194.01	\$163.86
		1st Plant (mil/kWh)	80th Plant (mil/kWh)
The projected BBEC are:		66.07	60.40

FIGURE 4-7 SRI LINE-FOCUS SOLAR CENTRAL POWER PLANT PLOT PLAN

FIGURE 4-8 FULL DAY ENERGY - WINTER

FIGURE 4-9 FULL DAY ENERGY - SPRING

FIGURE 4-10 FULL DAY ENERGY - SUMMER

FIGURE 4-11 ANNUAL ENERGY PERFORMANCE

4.1.3 System Operation

This section briefly outlines the daily startup, normal operation, and the evening shutdown of the power plant. Under shutdown conditions, the turbine-generator is removed from the grid and is placed on the turning gear. Cold salt from the HSS is circulated through the field piping and the steam generator to provide freeze protection.

To start the plant in the morning, the steam generator is first brought from the standby mode to the minimum operating temperature condition. This is done before dawn, and requires approximately 0.5 hour. The energy for heating the steam generator is supplied from the hot salt storage tanks. Once the steam generator reaches minimum operating conditions, the main steam lines are cleared of condensate and main steam is admitted to the turbine. The turbine-generator is brought to the 3 to 5 MW_e load condition, and is synchronized with the grid. The steam generator is then brought to normal operating conditions. Sunrise should be occurring at this time. The receivers and field piping are brought to operating temperature, and the thermal energy from the receiver subsystem is directed to the steam generator. As the solar insolation increases, the thermal output of the receivers increases, as does the output of the turbine-generator. The output of the receivers reaches a value corresponding to a solar multiple of 1.0 in approximately 1 hour. The output of the turbine-generator increases from the 5 MW_e level to the 107.2 MW_e maximum guaranteed load condition over a period of approximately 1.5 hours.

The output of the receivers increases in the morning, reaches a maximum at noon, and decreases in the afternoon. However, the turbine-generator continues to operate at the maximum guaranteed load condition. The receiver output in excess of that required to drive the turbine-generator is stored in the hot-salt storage tanks.

When the receiver output in the afternoon falls below the level corresponding to a solar multiple of 1.0, hot salt from the ESS is pumped to the steam generator to maintain the turbine-generator at the maximum guaranteed load condition. Within 1 hour, the output of the

receivers approaches zero, and the salt flow in the field piping and receivers is stopped. The hot salt flow from the ESS is now increased to maintain the maximum guaranteed turbine load condition.

The turbine-generator operates during the evening at this load condition until the hot salt tanks are approximately 85% empty. At this time, the load on the turbine-generator is ramped down to the 3 to 5 MW_e level over a period of approximately 1 hour. When this load condition is reached, the turbine-generator is removed from the grid and placed on the turning gear. The steam generator is cooled from operating temperature, and is placed in the standby mode. Cold salt circulation through the field piping is started to provide freeze protection for the piping and the steam generator. An adequate volume of hot salt remains in the storage tanks to provide freeze protection for the HTS, to provide overnight auxiliary steam for the EPGS, and to restart the steam and turbine-generators the following morning.

4.2 Heliostat Design

4.2.1 Heliostat Description

The baseline heliostat design is submodule 3 m (10 ft) wide by 12 m (40 ft) long with four 9.3 m^2 (100 ft^2) reflector panels on each, as shown in Figure 4-12. Four submodules give a total of 148.6 m^2 (1600 ft^2) of aperture. They are connected together on a common rotational axis, 1.82 m (6 ft) above the ground, and driven by a single mechanism located between the second and third submodules. The heliostat modules are supported on cast-in-place concrete caissons with 1.82-m (6-ft) pedestals. The end modules of adjacent groups share a common foundation, as shown in Figure 4-13.

The reflective panels are constructed of 0.71 mm (0.028 inch) flex glass, which is silvered on the second surface and laminated to a rigid structural backing made of plastic sheet molding compound. The thin glass reflector was chosen for its high reflectance, high specularity, durability, ease of cleaning, and moderate cost. These reflector panel assemblies are lightweight, but have the required stiffness and dimensional stability for keeping long-term surface accuracy. Each panel is supported (at approximately the one-quarter span points, in both directions) to minimize panel deflection due to dead weight and operating wind loads.

The heliostat module structure is a glass-fiber-reinforced concrete cast into a triangular cross section torque tube for strength and rigidity. The predominant loads are flexural, due to dead weight and wind drag, with a small torsional component from the wind. Considering the structural weight, the effect of deflections on optics, manufacturing and installation factors, and foundation costs, a 12.2-m (40-ft) module length appears to be near optimum. Grouping four modules to each drive results in a cost-optimum configuration, based on drive cost and the cost of structure torsional stiffness. Fiber-reinforced concrete significantly reduces material cost, which more than offsets the increase in structural weight.

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FIGURE 4-13 LINE-FOCUS CENTRAL RECEIVER HELIOSTAT MODULE

The drive unit consists of a four-bar linkage mechanism, driven by a recirculating ballscrew, with an ac stepper motor. The heliostat group stows inverted and inclined at a 7° angle to the horizon. At this attitude, in winds that vary $\pm 10^\circ$ from the horizon, the wind-induced pitching movement always causes the heliostats to bear against a mechanical stop. In this way the drive unit need not be sized for the full 90 mi/h stowed loads, but only the 50 mi/h operating torque. Therefore, a significantly less expensive drive is possible.

The foundations are poured-in-place concrete caissons, with a 6-ft pedestal to support the heliostat modules. This design has been compared with many other foundation concepts. Replacing structural steel members with the concrete pedestal results in a lower cost, and also allows the use of a simplified installation procedure.

Specifications for the heliostat appear in Table 4-3, followed by cost and weight distribution summaries in Table 4-4. The optical error budget for new and in-service conditions is given in Table 4-5.

4.2.2 Reflector Panel

The reflective panels are a composite of a 0.7 mm (0.028 in) second surface silvered glass mirror, with a sheet molding compound (SMC) supporting structure, as shown in Figure 4-14. Sheet molding compound is a composite material, made of polyester resin with chopped glass fiber reinforcement. Details of the panel design, attachment links, and alignment are discussed in the next three subsections.

4.2.2.1 Reflective Panel Design Objectives

It was concluded during the design effort that a cost effective heliostat design should utilize a high-performance, durable, reflecting medium in order to meet the 30-year lifetime requirement. To meet this design goal, the following performance and structural design objectives were identified and met, as indicated:

Table 4-3

HELIOSTAT SPECIFICATIONS

Heliostat submodule size	3.05 m (10 ft) aperture x 12.2 m (40 ft) length. 37.2 m ² (400 ft ²) net reflector area.
Heliostat size	Four 12.2-m (40-ft) heliostat submodules. 148.7 m ² (1,600 ft ²) total area.
Rotational axis, height above ground	1.82 m (6 ft)
Nominal assembled weight of heliostat (foundation included)	63.9 kg/m ² (13.1 lb/ft ²)
Heliostat materials	
Reflector panels	Thin glass on plastic sheet molding compound
Structure	Glass-fiber-reinforced concrete, triangular cross-section torque tube
Drive	Steel
Foundation	Reinforced concrete
Stow position	Inverted, tilted 7° to horizon
Tracking angle	195° range, elevation
Reflectance	0.91
Power requirements (per drive)	
Tracking (on/off)	0.6 kWh/d
Slew	0.04 kW
Emergency shutdown	0.23 kW
Heliostat operating module (track, stow)	
Normal tracking	Up to 56 km/h (35 mi/h)
Stow command	Given at 56 km/h (35 mi/h)
Stow position	81-145 km/h (50-90 mi/h)
Environmental exposure	Wind exposure limits, as above. Full operation of heliostat function can be resumed after exposure to any of the environmental conditions, as stated in the system requirements definition.

Table 4-4

HELIOSTAT COST AND WEIGHT DISTRIBUTION TABLE

Subassembly	Cost (\$/ft ²)	Weight (lb/ft ²)
Reflector	0.83	0.4
Reflector backing (SMC)	2.04	2.0
Structure (GRC)	0.50	7.5
Drive	0.65	0.3
Foundations	0.40	2.9
Installation	0.63	--
Total	5.05	13.1

Table 4-5

HELIOSTAT OPTICAL ERROR BUDGET*

Optical Error	Magnitude (mr)	
	New	After 5 years
Reflector panel slope errors	1.0	1.2
Reflector panel deflections	0.5	0.5
Reflector specularity	0.1	0.2
Structural deflections	0.5	0.5
Drive backlash	0.8	0.9
Foundation deflections	0.05	0.05
Convolved total error	2.9	3.3

*With respect to reflected rays

TYPICAL RIB CROSS SECTION

FIGURE 4-14 REFLECTIVE PANEL BACKING

- Performance

- High reflectance and low specularity are achieved by the use of a second-surface-silvered glass mirror
- Low slope error is provided by the press-molded sheet molding compound
- High stiffness is obtained with an isogrid backing structure, optimized for low cost through low weight
- Minimum angular deflection of the panel results from the selection of optimum support point locations

- Structural

- Hail impact survival is provided by the sheet molding compound face sheet
- Overstressing the glass mirror due to 44.7 m/s (100 mi/h) wind load deflection is prevented by the sheet molding compound's isogrid backing
- Thermal cycling -26°C to 60°C (-20°F to 140°F) effects are minimized by matching the sheet molding compound's coefficient of thermal expansion to that of the glass mirror.

4.2.2.2 Flex Glass Reflector

The flex glass reflector was chosen because of its high performance and durability. In terms of performance, the second-surface-silvered reflecting surface provides the highest practical solar hemispherical reflectance ($R_{2,2\pi} = 0.95$), while the glass itself has excellent specularity ($\sigma_m < 0.1 \text{ mr}$). Glass is durable because of its high abrasion resistance and its resistance to most common, natural degrading substances (e.g., bird droppings). The mirror edges can be sealed to protect the reflective coating by many methods, depending upon the panel fabrication process.

4.2.2.3 Sheet Molding Compound (SMC) Panel

The reflective panel structure is a molded composite material, generically referred to as sheet molding compound (SMC). SMC is a glass-fiber-reinforced polyester resin structural material to which filler material can be introduced to modify properties such as density, modulus of elasticity, coefficient of thermal expansion, and strength

(Table 4-6). The face sheet, which supports the mirror, and the isogrid backing are formed simultaneously in a molding process conducted at 149°C (300°F).

Table 4-6

SHEET-MOLDING COMPOUND PROPERTIES

Compressive strength	28,000 psi
Tensile strength	12,500 psi
Flexural strength	28,000 psi
Modulus of elasticity:	
tensile	1.30×10^6 psi
flexural	1.35×10^6 psi
Specific gravity	1.85
Mean coefficient of thermal expansion*	from -60°F to 115°F: $11.5 \times 10^{-6}/^{\circ}\text{F}$ from 115°F to 300°F: $3.0 \times 10^{-6}/^{\circ}\text{F}$

* Can be altered to match the coefficient of thermal expansion of the glass.

The SMC face sheet supports the glass mirror and provides resistance to hail impact 19 mm (.75 in) diameter hailstones traveling at 19.8 m/s (65 ft/s). Preliminary impact analysis indicates that a 0.25-mm (0.01-in) thickness is adequate.

The isogrid structure supports the face sheet, and provides a high degree of stiffness at a minimal cost in weight. The grid stiffening imparts isotropic macroscopic properties to the structure; hence the term, isogrid. The isogrid/face sheet structure serves to limit the glass stresses resulting from deflections induced by 40 m/s (90 mi/h)

winds. The effective wind velocity on the panel in the analysis was actually 31.3 m/s (70 mi/h) due to the boundary layer effect and the low stow elevation angle.

In designing the isogrid structure the Isogrid Design Handbook (Ref. 4-1) was consulted to define the theoretical optimum combination of grid parameters, such as rib thickness, grid depth, and grid spacing. Because the face sheet thickness is a function of the hail impact survival specification, the parameter dimensions indicated in Figure 4-14 deviate somewhat from the theoretical optimum combination. Another factor contributing to this deviation is the actual ratio of minimum rib thickness to grid depth that can reasonably be manufactured. The finite element analysis computer code ANSYS was used to analyze the grid design to verify conformance to the wind survival specification.

4.2.2.4 Module Structure

For the conceptual design, a glass-fiber-reinforced concrete (GRC) structure was chosen, primarily on the basis of low material costs. Recent studies (Refs. 4-1, 4-2, 4-3, 4-4) have shown that the material strength of Portland-type cements can be increased 2 to 5 times in tension, flexure, and impact loading by the addition of glass fibers, up to about 8% by volume. The increases in physical strength are dependent upon variables such as fiber length and diameter. The typical effect of glass fiber addition is shown in Figure 4-15. Recent developments have resulted in alkali resistant glass-fiber compositions to develop long-term, stable properties. The usual method of fabricating large structural shapes consists of a mechanized spray (gunnite) approach that simultaneously applies chopped glass fibers and cement with an aggregate material (sand). In order to maximize mechanical strength, the molded shape can be put under vacuum (or pressed) to remove excess water from the glass-fiber and concrete matrix. If the glass-fiber-reinforced concrete was formed as a flat sheet, further mold forming can be done while it is still in this "green" state. After water removal and any further shaping is finished, the CRC mold is air-cured (or steam-cured) to allow sufficient strength to develop

FIGURE 4-15 EFFECT OF FIBER ADDITION ON STRENGTH

for subsequent release and handling. The structure is allowed to cure fully (to near-maximum physical strength) in the normal 28-day period used for Portland-type cements.

For design purposes, the physical properties listed in Table 4-7 were used as typical full-strength values for the glass-fiber-reinforced concrete.

In determining the overall size and shape of the GRC main support structure, the following considerations are important:

- Combine with 3.05 m (10 ft) wide panel for minimum panel deflection
- Balanced torsional and flexural strength and stiffness
- Low stow wind drag profile to minimize wind loading

Table 4-7

PHYSICAL PROPERTIES OF GRC

	Maximum Values	Allowable Values
Compressive strength, psi	9570	3580
Tensile strength, psi	2320	870
Modulus of rupture	5800	2175
Shear strength, psi	--	108
Modulus of elasticity, psi	4.35×10^6	--
Density, lb/ft ³	136	--

- Lowest practical weight to reduce material cost and cost of support structure and foundation
- Support pivot location at the center of gravity to minimize tracking power requirements

During operation, in comparison to the wind forces, the weight of the GRC structure is the primary load. In the stow position for survival, the wind load predominates. Considering the full range of operation, the receiver module is exposed to moderate wind pressure against the full 3.05 m (10 ft) aperture width during operation and high wind pressures against the structure 1 m to 1.2 m (3 ft to 4 ft) only in the stow position. These are the two governing load conditions. This loading scheme requires a structure with comparable load-carrying capacities or strengths in all orientations and (preferably) with a closed section to carry the wind-induced pitching moments restrained by the drive unit. Tubular structures with round, square, or triangular cross-sectional shapes have good load carrying capacities; the triangular shape proved the simplest to use with the 3.05 m (10-ft) wide reflective panels having the minimum structural material requirement.

For minimum reflective panel deflection, longitudinal and lateral supports are placed at the one-quarter span locations. For a 3.05 m (10-ft) panel, the supports are on nominally 1.63 m (64.0 inch) centers,

requiring the triangular structure be a maximum of 1.68 m (66 inches) on a side. See Figure 4-16. The depth (altitude) of the triangular cross section was sized to carry the maximum loads on the reflective panel and to provide the minimum required flexural strength for a minimum amount of structural material. The last criterion for sizing the triangular torque tube was an assumed requirement for a minimum wall thickness of 12.7 mm (0.5 inch) for the proper fabrication of the GRC material. An allowance must be made for the aggregate (sand) size and normal imperfections in the glass-cement matrix. With these design guidelines, the torque tube dimensions that resulted are:

Base	1.68 m (66 inches)
Altitude	1 m (39 inches)
Wall thickness	0.01 m (0.5 inches)
Length	12.2 m (480 inches)
Weight	1409 kg (3100 lb)

4.2.2.5 Panel Attachment

For proper panel support and alignment, panel support links must meet the following criteria:

- Permit rapid panel installation and alignment
- Allow for thermal expansion and contraction between the panels and the module structure
- Provide vertical and lateral stability

Adjustable turnbuckles mounted to clevises near one panel edge were selected. The turnbuckles provide a simple elevation adjustment for the correct panel alignment. Rigid links with balljoints at one end and attached at the other end to the module structure near the other panel edge were also selected.

4.2.2.6 Drive Subassembly

The drive subassembly consists of the drive motor, power transmission linkage, and module support bearings. Initially, the design

FIGURE 4-16 CONCRETE HELIOSTAT MODULE ASSEMBLY WITH TRIANGULAR TORQUE TUBE

loads for drive component sizing were based on the pitching moment at the 145 km/h (90 mi/h) stow condition. It was subsequently found that by positioning the heliostat with its reflective panels inclined slightly from complete inversion, a significant decrease in the drive holding torque resulted. However, the lift and drag loads increased somewhat. A cost tradeoff, between the drive being unloaded, and the foundation loads increasing, was performed to determine the stow position for the minimum total cost. It has been determined that at a 7° incline, with the drive linkage driven against a rigid mechanical stop and a wind approach angle of $\pm 10^\circ$ from horizontal, the net wind angle of attack ranges from 3° to 17°. The drive linkage is sized for the 3° angle of attack (a 69% reduction) and the mechanical stop is sized for the higher loads at 17°. Table 4-8 summarizes these loads.

Table 4-8

STOW LOADS

<u>Stow Position</u>	<u>Drag (lb)</u>	<u>Pitch Moment (ft-lb)</u>
0°	733	30,400
7°	1,373	9,424

The recirculating ballscrew actuator uses a 1/8 hp, 110 Vac stepper motor with an incremental encoder for positional tracking control; all are current, standard products. The linkage members are designed for the 50 mi/h slewing load requirements and provide 195° of total travel. These members are steel forgings, with pressed bushings at all connecting hinge points. The mechanical stop is an integrally forged boss in the main drive support yoke.

4.2.2.7 Bearings

To support the heliostat modules, while at the same time allowing tracking rotation, some type of bearing must be selected. Appropriate bearing types for this high static load, low and intermittent speed, and outside exposure application included radial ball bearings, bushings, wire race bearings, and cam followers. Both radial ball bearings (pillowblocks) and cam followers were previously used by Acurex/Aerotherm for single-axis tracking collectors, where they performed very well.

Wire race bearings, used for very large, slow-moving equipment, such as gun turrets and antenna mounts, were also considered. Suppliers were contacted, and the indications were that in the 0.3 m (12-inch) diameter size range of interest, this type of bearing would not be competitive with ball bearings.

Each bearing was evaluated on the basis of initial cost for material and assembly labor, parasitic friction losses, and precision. For this application pillowblock radial ball bearings were selected.

4.2.3 Heliostat Design

4.2.3.1 Heliostat Foundation

The heliostat foundation design selected for the conceptual plant design from the parametric analysis is an augered, reinforced-concrete caisson 0.38 m (1.25 ft) in diameter and 3.66 m (12 ft) deep. A reinforced concrete pedestal, 203 mm (8 inches) wide by 330 mm (13 inches) deep by 1.83 m (6 ft) high, is cast with each caisson. The pedestals support the heliostat bearings and drive mechanisms. The center-to-center spacing of the foundations is 12.2 m (40 ft). The 100 MW_e plant of the conceptual design requires 24,240 foundations.

4.2.3.2 Heliostat Drives

The electric power distribution system for single lines of heliostat drive motors is shown in Figure 4-17. Each heliostat drive is powered

4160 V FEEDERS
(CONT'D. FROM ESK-1)

NOTES:

- 1) EACH COLLECTOR DRIVE CONTROL TO BE PROVIDED WITH INTERNAL OVERCURRENT AND SURGE PROTECTION.

FIGURE 4-17 HELIOSTAT DRIVE, POWER DISTRIBUTION SINGLE LINE

by a stepping motor with an input requirement of approximately 500 volt-amperes. Each of the 24 heliostat rows has 30 drive motors, which results in 720 drive motors for each of the 10 receiver rows. Each of the selected 112.5 kVA, 3 phase, 60 Hz, 4160/240 volt transformers will feed half of the 12 rows. This results in 180 motors per transformer. Therefore a total of 40 transformers is required to power all the heliostat drive motors.

To provide reliability, four fields of ten transformers each will be supplied by three 4160 volt feeders, any two of which are adequate to carry the load.

4.2.3.3 Heliostat Performance

The specific energy collection of the heliostats is:

- Winter solstice day: $3.6 \frac{\text{kW}_{th}}{\text{m}^2}$
- Equinox day: $4.1 \frac{\text{kW}_{th}}{\text{m}^2}$
- Summer solstice day: $4.6 \frac{\text{kW}_{th}}{\text{m}^2}$
- Average day: $4.1 \frac{\text{kW}_{th}}{\text{m}^2}$

Figures 4-18, 4-19, and 4-20 show the energy flux absorbed on the receiver for the winter solstice, equinox, and summer solstice, respectively. Each chart shows an hour-by-hour flux. The highest flux values correspond to solar noon. Each lower curve corresponds to 1-hour intervals before and after solar noon. The calculation was made on a discrete basis by dividing the receiver into 24 nodes. The curves result from a fourth-order polynomial fit to the data.

FIGURE 4-18 RECEIVER FLUX MAP: WINTER SOLSTICE

FIGURE 4-19 RECEIVER FLUX MAP: SPRING EQUINOX

FIGURE 4-20 RECEIVER FLUX MAP: SUMMER SOLSTICE

4.3 Receiver Conceptual Design

4.3.1 Configuration

Selection of the preferred receiver configuration was based on a review of the parametric analysis. Special consideration was given to receiver cost and development requirements.

In the receiver configuration that was selected, the molten salt enters the lower circuit at the beginning of zone 5 and leaves at the end of zone 13. It enters the upper circuit at the beginning of zone 24 and leaves at the end of zone 14. In this configuration, 42.8% (112.6 lb/h-ft) of the molten salt goes into the lower circuit and 57.2% (150.4 lb/h-ft) goes into the upper circuit.

4.3.1.1 Tube Diameter

Figure 4-21 shows the average Reynolds number as a function of tube O.D. and receiver pressure drop. Operation with Reynolds numbers of less than 10,000 (the transition region) is not recommended. In this region, the flow is sometimes laminar and sometimes turbulent. Fluctuations in pressure drop and heat transfer are characteristic. Because of the uncertainty about the basic heat transfer and flow friction performance, the receiver was designed to operate above this transition region.

Figure 4-22 illustrates the relation between tube wall thickness, tube O.D., and operational tube weight (tube material plus molten salt). The figure shows that this weight is a strong function of tube O.D. and a weak function of tube wall thickness. Receiver costs are closely related to the weight per unit length. Thus, as can be seen from the figure, small O.D. tubes offer the greatest potential for cost savings. However, as the tube diameters decrease, more tubes are required, increasing shop labor because of added tube bends and tube-to-header welds. A tube with 19-mm (0.75-inch) O.D. was selected as a reasonable compromise. This allows the receiver to operate in the turbulent region at full load and gives reasonably low receiver weights.

FIGURE 4-21 AVERAGE REYNOLDS NUMBER vs TUBE O.D.
(49-mil WALL THICKNESS)

FIGURE 4-22 RECEIVER WEIGHT vs TUBE O.D.

4.3.1.2 Receiver Module Length

The length of the receiver module and the average Reynolds number were calculated as a function of the receiver pressure drop. The results are shown in Figures 4-23 and 4-24. The allowable receiver module pressure drop of 100 psi will give an average Reynolds number of 18,000, and a receiver module length between 22.9 m and 25.9 m (75 ft and 85 ft).

4.3.1.3 Tube Wall Thickness

Since the design pressure of the receiver tubes is low (150 psi), the hoop stresses are very low, and ductile rupture should not be a predominant failure mode. The minimum wall thickness required by Section VIII of the ASME Boiler and Pressure Vessel Code is smaller than the minimum wall thickness of 89 mm (0.035 inch) required by manufacturing considerations. Therefore, if internal pressure were the only consideration, a 89-mm (0.035-inch) wall thickness could be used. However, an 89-mm (0.035-inch) wall is difficult to weld and will require more rework. There are also concerns about the feasibility of extruding fins on a tube with this wall thickness. Thus, a 19-mm (0.75-inch) O.D. finned tube with a wall thickness of 1.24 mm (0.049 inch) was finally selected for the receiver.

4.3.1.4 Materials

SA 178 carbon steel and Type 316 stainless steel were selected for the receiver tubes and headers. The selection was based on recommendations from Martin Marietta Corporation, which has conducted extensive material test programs for molten salt receivers. These materials have gained wide acceptance and have an extensive service record in fossil fuel power plants. Carbon steel will be used in low temperature service [less than 427°C (800°F)] and stainless steel at temperatures above 427°C (800°F).

Table 4-9 shows the heat transfer coefficients and average wall temperatures in each zone of the receiver. This table was used to

FIGURE 4-23 RECEIVER MODULE LENGTH vs PRESSURE DROP, 19 mm (0.75 in) O.D. TUBES

FIGURE 4-24 AVERAGE REYNOLDS NUMBER vs RECEIVER PRESSURE DROP, 19 mm (0.75 in) O.D., 49 mil WALL

Table 4-9

PERFORMANCE AND MATERIAL SELECTION

Zone Number	Salt Outlet Temperature (°F)	Average Salt Temperature (°F)	Reynolds Number	Heat Transfer Coefficient (Btu/h-ft ² -°F)	Average Wall Temperature (°F)	Material
65T ↓ LOWER CIRCUIT	6	557.9	9,015	263	556.1	Carbon steel
	7	577.6	9,461	273	572.9	Carbon steel
	8	613.3	10,425	295	604.2	Carbon steel
	9	666.7	12,187	334	651.6	Carbon steel
	10	740.0	14,628	387	717.2	Carbon steel
	11	831.1	17,457	446	800.5	Carbon steel
	12	936.6	22,000	536	899.1	Stainless steel
	13	1,050.0	29,161	672	1,006.8	Stainless steel
	14	1,050.0	31,465	714	1,020.7	Stainless steel
	15	966.7	25,462	603	940.7	Stainless steel
	16	887.2	21,003	517	865.1	Stainless steel
	17	813.6	17,844	453	794.2	Carbon steel
	18	745.4	15,481	405	729.0	Carbon steel
↑ UPPER CIRCUIT	19	681.9	13,252	358	670.5	Carbon steel
	20	632.2	11,552	320	625.3	Carbon steel
	21	597.3	10,450	295	593.4	Carbon steel
	22	574.9	9,819	281	572.8	Carbon steel
	23	561.7	9,421	272	559.1	Carbon steel
	24	554.3	9,278	269	553.6	Carbon steel

select the tube materials used in each zone. Zones 6 through 11 and 17 through 24 will be made of SA 178 carbon steel, and zones 12 through 16 will be made of Type 316 stainless steel. Thus, about 74% of the tubes will be carbon steel, and 26% stainless steel.

4.3.1.5 Design Concept

Each module consists of the tube bundle, support structure, and insulation. The tube bundle contains 112 tubes, 19 mm (0.75 inch) O.D. by 1.2 mm (0.049 inch) wall. There are 84 carbon steel tubes and 28 stainless steel tubes. As shown in Figure 4-25, each module has two circuits of 16 panels each. The lower circuit has three tubes per panel, and the upper circuit has four tubes per panel. Details of the receiver modules are shown in Figure 4-26.

Figure 4-27 shows a typical inlet/outlet header. The headers are located behind the tube bundle, and are protected from the incident heat flux by the tube bundle and the inner casing (shell) of the receiver.

4.3.1.6 Tube Supports

The provision made for thermal expansion in the receiver module is shown in Figure 4-28. Sliding mounting brackets allow the tubes to expand horizontally by sliding along two angle brackets attached to the receiver shell. The tube bundles will be anchored at the inlet/outlet end, and will be free to expand towards the other end of the receiver module. This tube support arrangement will minimize thermal stresses in the tube bundles and in the inlet and outlet piping during steady loads and transients. A gap between receiver modules is required to accommodate thermal expansion of the tube bundle. Since the hotter tubes near the outlets will expand more than the colder tubes near the inlets, the tube lengths will vary to account for this difference in thermal expansion. Thus, during normal operation at rated load the tubes will expand to about the same length, minimizing the space between receiver modules. This method of allowing for thermal expansion was

FIGURE 4-25 DEVELOPED VIEW OF A TUBE BUNDLE

selected because of cost. The subject of expansion accommodation deserves further consideration during the detail design of the receiver.

4.3.1.7 Aperture Door

A movable insulated "eyelid" closes the aperture when there is no solar input, so that thermal cycling and heat losses are minimized. The receiver can be placed in a hot standby mode overnight or during cloud cover with minimum heat losses and without the danger of freezing the salt in the tubes. The hot standby mode eliminates the need to drain the receiver during relatively short periods of no insolation.

Each 20-ft door is opened to a horizontal position by two pneumatic cylinders, which provide a lightweight, inexpensive way to operate the receiver doors. The pneumatic cylinders also allow the doors partially to close by gravity in case of system failure.

4.3.1.8 Drainage

A drainage scheme was developed to provide drainage of the receiver modules in the event that salt is left in the receiver tubes long enough to freeze (approximately 430°F). Drainage will be accomplished by installing each receiver module in a tilted position 3° to 5° from horizontal so that the molten salt drains by gravity to the headers after the pumps are shut down. Each header has a drain line equipped with automatic drain valves that discharge into the tower downcomer through a check valve. This drainage system is proposed to show that it is feasible and relatively inexpensive. A detailed drainage system is usually part of the detail engineering phase of the project, but was beyond the scope of work of the conceptual design.

4.3.1.9 Structure

The support structure was designed to meet the environmental requirements defined in the requirements definition document. The primary concern was to minimize cost and weight. The detail design effort was limited to the support brackets which attach the receiver shell module

FIGURE 4-26 LINE-FOCUS RECEIVER, SECTIONAL VIEW

FIGURE 4-27 TYPICAL BUNDLE ARRANGEMENT AT AN INLET/OUTLET HEADER

FIGURE 4-28 TYPICAL EXPANSION BRACKET

to the supporting structure from which the receiver is suspended between the towers.

4.3.1.10 Insulation

The design requirements established for the receiver insulation are as follows:

- Low thermal conductivity
- Minimum sag, separation, and settling
- Availability in blanket form
- Low density
- Operation up to 1100°F
- Low cost.

Based on previous experience, ETR insulation made by Forty-Eight Insulations, Inc. was selected. This material is a mineral wool insulation that is manufactured in blanket form, has low thermal conductivity and low density, and will maintain bond strength up to 1200°F. A typical ETR blanket has a thermal conductivity less than 0.5 Btu-in/h-ft²-°F at a mean temperature of 700°F.

Determining the optimum insulation thickness involves a tradeoff between the incremental cost of insulation and heat losses. Thicker insulation results in more weight (increased tower and receiver costs) but less conduction losses. This tradeoff should be made at the system level during the detail design phase of the plant. For this preliminary conceptual design and cost estimate, an average insulation thickness of 6 inches was assumed.

4.3.1.11 Instrumentation and Control

The flow concept for the receiver module has the salt at a temperature of 550°F entering the receiver and flowing back and forth before exiting at 1050°F. The temperature at the receiver module outlet is controlled by a valve at the inlet that regulates the mass flow rate to keep the outlet temperature constant while the solar flux varies.

Temperature sensors in the salt downstream of each receiver module generate the control signals required by the receiver module feedback control loop.

It is recommended that the receiver operation be limited to power levels greater than 25% of the design power. Thus, when the duty of the receiver is too low to maintain the required molten salt at 25% of the design flow rate, the temperature control system is overridden by the minimum flow control system and the receiver is shut down.

4.3.1.12 Receiver Weights

A weight estimate has been prepared for the receiver. This estimate includes the weight of the shell, tube bundle, headers, insulation, support brackets and all other receiver components. The weight breakdown of an 80-ft receiver module is shown in Table 4-10.

Table 4-10

RECEIVER MODULE WEIGHT BREAKDOWN (80-ft Module Length)

	kg	lb
Support castings	116	256
Lugs for castings	149	327
Support brackets	1,877	4,130
Inner shell and structural members	4,818	10,600
Carbon steel tubes	1,736	3,820
Stainless steel tubes	568	1,250
Carbon steel headers	30	65
Stainless steel headers	17	37
Pneumatic cylinders	455	1,000
Insulation	2,509	5,520
Aluminum sheathing	759	1,670
Total per module	13,033	28,673
Linear weight of receiver	535 kg/m	359 lb/ft

4.3.2 Tower and Bridge Design

Data sheets for the receiver towers and bridges are presented in Tables 4-11, 4-12, and 4-13.

To identify the optimum tower spacing in the parametric analysis, conceptual level designs were prepared for the receiver towers and bridges. These were adopted for the conceptual design. Therefore, the tower and bridge design effort in Task 4 concentrated on a receiver wind deflection analysis.

The receiver tower design selected from the parametric analysis is a self-supporting steel structure 61 m (200 ft) high. The design, illustrated in Figure 4-29, is based on ASCE Manual 52 for electric transmission towers. The towers are constructed on the ground from galvanized steel sections. Once assembled, they are raised and placed on their foundations by a high lift crane. Four augered, belled caissons, one located at each corner of a 15.2 m (50 ft) by 15.2 m (50 ft) square, form the foundation for each tower. The tower bridges are parallel chord Warren trusses 45.6 m (150 ft) long. The 25.9 m (85 ft) receiver sections are supported on the south face of the bridges. The 100-MW_e plant of the conceptual design requires 310 receiver towers and 300 receiver bridges.

A survival wind speed of 40.3 m/s (90 mi/h) at an elevation of 61 m (200 ft) was used in the receiver deflection analysis. The wind speed was assumed to vary exponentially with elevation as outlined in the requirements definition document. Wind loads on the tower structure were based on the combination of wind speed and cross-sectional area at a number of elevations. Wind loads on the receiver were approximated as a point load on the receiver tower. In the deflection analysis, the tower was modeled as a cantilevered beam with varying moments of inertia. The results from the analysis indicate that the receiver deflection at the towers is approximately 102 mm (4 inches) and at the center of the bridge, approximately 152 mm (6 inches). Under assumed daily average wind conditions of 6.7 m/s (15 mi/h) at an elevation of 61 m (200 ft), the receiver deflection decreases to approximately 2.5 mm (0.1 inch).

Table 4-11

DESIGN CHARACTERISTICS FOR RECEIVER SUBSYSTEM RECEIVER PIPING AND INSULATION DATA

Receiver Headers	O.D.		I.D.		Wall		Total Length		Pipe Mass		Calcium Silicate Insulation			
											Thickness		Mass	
	mm	in	mm	in	mm	in	m	ft	kg	lb	mm	in	kg	lb
Cold	219.1	8.625	211.6	8.329	3.8	.148	10,576	34,700	210,713	464,542	57.15	2.25	101,080	222,843
	168.3	6.625	161.5	6.357	3.4	.134	3,974	13,040	55,008	121,272	50.8	2	26,711	58,889
	114.3	4.5	108.2	4.26	3.0	.12	4,094	13,432	34,058	75,085	50.8	2	20,733	45,709
	88.9	3.5	82.8	3.26	3.0	.12	369	1,210	2,365	5,215	50.8	2	1,581	3,485
Hot	219.1	8.625	211.6	8.329	3.8	.148	3,672	12,048	73,229	161,443	101.6	4	72,251	159,287
	168.3	6.625	161.5	6.357	3.4	.134	14,575	47,820	201,724	444,726	69.85	2.75	146,760	323,550
	114.3	4.5	108.2	4.26	3.0	.12	4,048	13,280	33,793	74,501	63.5	2.5	27,595	60,836
	88.9	3.5	82.8	3.26	3.0	.12	434	1,424	7,787	17,166	63.5	2.5	2,537	5,592

Table 4-12

RECEIVER SUBSYSTEM TOTAL MASS OF RECEIVER HEADER WORKING FLUID
(Excluding Riser/Downcomer)

Receiver Header	Nominal Pipe Diameter (in)	Salt Mass	
		lb	kg
Cold	8	1,430,902	649,046
	6	310,498	140,840
	4	157,828	71,589
	3	8,325	3,776
Subtotal		1,907,553	865,251
Hot	8	465,953	211,353
	6	1,131,945	513,442
	4	141,187	64,041
	3	8,864	4,021
Subtotal		1,747,949	792,857
Total		3,655,502	1,658,108

Table 4-13

RECEIVER SUBSYSTEM RISER/DOWNCOMER PIPING AND INSULATION DATA

	O.D.		I.D.		Wall		Total Length		Pipe Mass		Calcium Silicate Insulation			
											Thickness		Mass	
	mm	in	mm	in	mm	in	m	ft	kg	lb	mm	in	kg	lb
Cold riser*	273	10.75	260.4	10.25	6.4	.25	747	2,450	31,428	69,286	57.2	2.25	8,524	18,792
Hot downcomer†	273	10.75	265	10.42	4.2	.165	328	1,076	9,171	20,218	101.6	4	7,541	16,624
	168.3	6.625	161.5	6.357	3.4	.134	466	1,530	6,454	14,229	69.9	2.75	4,698	10,358

*Carbon steel A-106

†Stainless steel 316

- 1) ----- REDUNDANTS L 1 3/4 x 1 3/4 x 3/16
FOUR FACES
- 2) SECTION DIMENSIONS ARE IN INCHES

FIGURE 4-29 TOWER DESIGN

4.3.3 Receiver Conduction Losses

Conduction losses per linear foot of receiver at full load operating conditions were estimated as a function of the insulation thickness and air-side heat transfer coefficient for an ambient air temperature of 80°F. The results are given in Table 4-14. The resistance of the outside film was small compared to the insulation resistance. A review of the table shows that the estimated heat loss changes by less than 11% for the range of air-side heat transfer coefficients considered. Thus, the uncertainty associated with the air-side convective heat transfer coefficient acting on the outside surface of the receiver is not a critical factor, and the conservative assumption that the outside surface of the receiver is at ambient air temperature is a reasonable approximation. Using this assumption, the conduction losses per foot of receiver were calculated as a function of the ambient temperature (see Table 4-15).

4.3.4 Receiver Performance

The receiver is intended to be operated with an inlet temperature of about 550°F and an outlet temperature of 1050°F. The inlet temperatures may vary, but the flow is controlled to give the specified outlet temperature. The salt flow per foot of receiver is 95 lbm/h-ft at the design point, producing 100 MW_e. The salt flow is highest at summer noon (224 lbm/h-ft).

Figures 4-30 through 4-32 show the thermal power delivered on each of the three days studied. The areas under each curve are divided into four regions of three different kinds, designated as direct, optional, and stored. The direct power level indicates the thermal power level required by the EPGS to produce 100 MW_e. Power rates above that value must be stored. The optional energy may be stored or used directly by being supplemented from storage. The annualized receiver efficiency (including interception, reflection, radiation, and convection losses) is 63%.

Table 4-14

CONDUCTION HEAT LOSSES vs INSULATION THICKNESS.

Average Insulation Thickness (in)	Insulation Weight (lb/ft)	Insulation Cost (\$/ft)	Conduction Heat Losses					
			h = 2		h = 30		h = ∞	
			(kW/ft)	Percent of Incident Energy	(kW/ft)	Percent of Incident Energy	(kW/ft)	Percent of Incident Energy
3	40.8	16.60	0.622	3.902	0.683	4.290	0.689	4.321
6	69.0	28.54	0.340	2.133	0.356	2.236	0.358	2.244
9	93.3	40.46	0.239	1.499	0.246	1.547	0.246	1.550
12	125.6	52.38	0.187	1.174	0.191	1.200	0.191	1.202
15	154.0	64.30	0.155	0.975	0.158	0.992	0.158	0.993
18	182.3	76.22	0.134	0.840	0.135	0.853	0.136	0.853

Notes:

h = Air side heat transfer coefficient in $\text{Btu/h-ft}^2\text{-}^\circ\text{F}$

Ambient temperature = 80°F

All values per linear foot of receiver

Table 4-15

CONDUCTION HEAT LOSSES vs AMBIENT TEMPERATURE
(Average Insulation Thickness = 6 in)

Ambient Temperature		Conduction Heat Losses	
		kW/ft	Percent of Incident Energy
(°C)	(°F)		
-20	-4	0.404	2.534
0	32	0.384	2.410
20	68	0.364	2.286
40	104	0.344	2.160
60	140	0.325	2.036

FIGURE 4-30 DAILY PERFORMANCE OF RECEIVER: WINTER

FIGURE 4-31 DAILY PERFORMANCE OF RECEIVER: SPRING

FIGURE 4-32 DAILY PERFORMANCE OF RECEIVER: SUMMER

4.4 Heat Transport Subsystem

4.4.1 Subsystem Requirements

The heat transport subsystem (HTS) is designed to transfer thermal energy collected in the receivers to the energy storage subsystem (ESS) and to the electrical power generating subsystem (EPGS). The HTS of the conceptual design includes the field piping, primary salt pumps, and the steam generator.

4.4.2 Heat Transport Subsystem Design

The plot plan and field piping layout are illustrated in Figures 4-33 and 4-34, respectively. The HTS flow diagram is presented in Figure 4-35. The HTS equipment list is summarized in Table 4-16. Data sheets for the HTS are presented in Tables 4-17 through 4-23.

The conceptual design of the HTS includes the following items:

- Selection of the optimum insulation thickness for each hot and cold line size
- Selection of the optimum line diameter for each section of the field piping
- Details of the field piping including expansion provisions, schedules, supports, anchors, freeze protection, and drainage
- Hot and cold salt pumps
- Steam generator

Details of the conceptual design of the HTS are outlined in the following sections.

4.4.2.1 Piping System Design

4.4.2.1.1 Optimum Insulation Thickness

The selection of the optimum insulation thickness for each section of the field piping involves a tradeoff between the cost of the insulation and the incremental cost of the collector and receiver subsystems required to supply the heat loss through the insulation.

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FIGURE 4-34 PIPING LAYOUT

LINE NUMBER		①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩
NOON @ MGL	PRESSURE, MPa (psig) TEMPERATURE °C (°F) FLOW, 10 ³ kg/hr (10 ³ lb/hr.)	0.99 (143) 294 (561) 0.005 (0.01)	0.30 (43) 566 (1050) 0.005 (0.01)	2.32 (336) 294 (561) 1.62 (3.58)	0.32 (46) 566 (1050) 1.62 (3.58)	0	0				
EVENING @ MGL	PRESSURE, MPa (psig) TEMPERATURE °C (°F) FLOW, 10 ³ kg/hr (10 ³ lb/hr.)	0	0	0	0	0.32 (46) 294 (561) 1.09 (2.40)	1.00 (145) 566 (1050) 1.09 (2.40)				
25% GUARANTEED THROTTLE FLOW HEAT BALANCE	PRESSURE, MPa (psig) TEMPERATURE °C (°F) FLOW, 10 ³ kg/hr (10 ³ lb/hr.)							1.25 (181) 267 (513) 0.28 (0.62)	1.03 (147) 566 (1050) 0.28 (0.62)	1.24 (180) 267 (513) 0.56 (1.23)	0.79 (115) 566 (1050) 0.56 (1.23)
MAXIMUM GUARANTEED LOAD (MGL) HEAT BALANCE	PRESSURE, MPa (psig) TEMPERATURE °C (°F) FLOW, 10 ³ kg/hr (10 ³ lb/hr.)							2.28 (330) 294 (561) 1.09 (2.40)	0.80 (116) 566 (1050) 1.09 (2.40)	2.31 (335) 294 (561) 2.18 (4.80)	0.54 (78) 566 (1050) 2.18 (4.80)
VWO + 5% OVERPRESSURE HEAT BALANCE	PRESSURE, MPa (psig) TEMPERATURE °C (°F) FLOW, 10 ³ kg/hr (10 ³ lb/hr.)							2.34 (340) 296 (565) 1.20 (2.65)	0.76 (110) 566 (1050) 1.20 (2.65)	2.39 (346) 296 (565) 2.40 (5.29)	0.48 (69) 566 (1050) 2.40 (5.29)

MOTOR OPERATED BALL STOP VALVE
 CHECK VALVE

FIGURE 4-35 HTS/ESS FLOW DIAGRAM

Table 4-16

HEAT TRANSPORT SUBSYSTEM EQUIPMENT LIST

<u>Field Piping</u>	<u>Diameter (in)</u>	<u>Schedule</u>	<u>Length (m)</u>
Cold Salt Piping			
Main headers	10	20	490
	12	20	1,520
	14	20	520
Risers	10	20	1,740
Receiver headers	3	10S	730
	4	10S	4,100
	6	10S	3,980
	8	10S	10,570
Storage tanks		20	220
Steam generator		20	30
Hot Salt Piping			
Receiver headers	3	10S	830
	4	10S	4,050
	6	10S	14,580
	8	10S	3,670
Downcomers	6	10S	930
	10	10S	1,010
Main headers	8	10S	540
	10	10S	1,680
	12	10S	570
Storage tanks		10S	220
Steam generator		10S	30

	<u>Diameter (in)</u>	<u>Number</u>
Field Valves		
Cold Salt Piping		
Ball valves	2	300
	10	19
	14	2
Check valves	10	9
Hot Salt Piping		
Ball valves	2	280
	6	5
	8	6
	10	8
	12	11
Check valves	8	6

Table 4-16 (continued)

Primary Salt PumpsEPGS Cold Salt Pumps

Pumps	Quantity	3-50% capacity
	Type	Vertical, multi-stage centrifugal
	Materials	316 SS impellers and bowls, carbide bearings
	Capacity	0.159 m ³ /s, 111 m tdh (2525 gal/min, 364 ft tdh)
Variable speed drives	Quantity	3
	Type	Hydraulic pump and motor
	Turn down ratio	3 to 1
Motors	Quantity	3
	Type	Induction motor
	Rating	450 kW (600 bhp)

ESS Cold Salt Pumps

Pumps	Quantity	6-25% capacity
	Type	Vertical, multi-stage centrifugal
	Materials	316 SS impellers and bowls, carbide bearings
	Capacity	0.119 m ³ /s, 128 m tdh (1880 gal/min, 425 ft tdh)
Variable speed drives	Quantity	3
	Type	Hydraulic pump and motor
	Turn down ratio	3 to 1
Motors	Quantity	3
	Type	Induction motor
	Rating	375 kW (500 bhp)

ESS Hot Salt Pumps

Pumps	Quantity	6-25% capacity
	Type	Vertical, multi-stage centrifugal
	Materials	3.6 SS impellers and bowls, carbide bearings
	Capacity	0.088 m ³ /s, 61 m tdh (1390 gal/min, 215 ft tdh)
Variable speed drives	Quantity	6
	Type	Hydraulic pump and motor
	Turn down ratio	3 to 1
Motors	Quantity	6
	Type	Induction motor
	Rating	150 kW (200 bhp)

Steam GeneratorPreheaters

Quantity	2
Dimensions	1.09 m (43 in dia x 12.19 m (480 in) long
Shell (water)	Carbon steel
Tubes (salt)	732 U, 19.05 mm (3/4 in) O.D., 2.77 mm (0.109 in) wall, carbon steel
Duty	37.7 MW (128.5 x 10 ⁶ Btu/h)
Net area	1,110 m ² (11,950 ft ²)

Boilers

Quantity	4
Dimensions	1.09 m (43 in) dia x 6.1 m (240 in) high
Shell (salt)	347 SS
Tubes (water)	1,000 straight, 25.4 mm (1 in) O.D., 2.11 mm (0.083 in) wall, 347 SS
Duty	15.9 MW (54.2 x 10 ⁶ Btu/h)
Net area	410 m ² (4415 ft ²)

Steam Drum

Quantity	1
Dimensions	2.13 m (84 in) dia x 12.19 m (480 in)
Shell	Carbon steel
Internals	Moisture separators

Table 4-16 (concluded)

Steam Generator
(continued)

Superheaters	Quantity	2
	Dimensions	1.27 m (50 in)/x 8.53 mm (336 in) long
	Shell	347 SS
	Tubes (steam)	734 U, 19.05 mm (3/4 in) O.D., 1.65 mm (0.065 in) wall, 347 SS
	Duty	45.7 MW (156.0 x 10 ⁶ Btu/h)
	Net area	842 m ² (9065 ft ²)
Reheater	Quantity	1
	Dimensions	1.37 m (54 in) x 7.62 m (300 in) long
	Shell (salt)	347 SS
	Tubes (steam)	1000 U, 19.05 mm (3/4 in) O.D., 1.65 mm (0.065 in) wall, 347 SS
	Duty	46.7 MW (159.3 x 10 ⁶ Btu/h)
	Net area	916 m ² (9865 ft ²)

HTS Drain System

Cold Salt Drain Pumps	Cold salt circulation pump of salt maintenance system doubles as cold salt drain pump	
Hot Salt Drain Pumps	Hot salt circulation pump of salt maintenance system doubles as hot salt drain pump	
Cold Salt Drain Sump	Quantity	2
	Dimensions	2.44 m (8 ft) dia x 1.52 m (5 ft) high
	Materials	6.35 mm (1/4 in) carbon steel, 51 mm (2 in) calcium silicate insulation
	Comments	Salt level control Heat traced
Hot Salt Drain Sump	Quantity	2
	Dimensions	2.44 m (8 ft) dia x 1.52 m (5 ft) high
	Materials	4.75 mm (3/16 in) 316 SS, 102 mm (4 in) Calcium silicate insulation
	Comments	Salt level control Heat traced

Table 4-17

HEAT TRANSPORT SYSTEM HORIZONTAL PIPING AND INSULATION DATA

Main Header	C.D.		I.D.		Wall		Total Length		Pipe Mass		Calcium Silicate Insulation			
											Thickness		Mass	
	mm	in	mm	in	mm	in	m	ft	kg	lb	mm	in	kg	lb
ESS (cold)	273.1	10.75	260.4	10.25	6.4	0.25	460	1,508	19,344	42,646	57.15	2.25	5,246	11,565
	323.9	12.75	311.2	12.25	6.4	0.25	470	1,540	23,282	51,328	63.5	2.25	6,971	15,369
	355.6	14	339.8	13.376	7.9	0.312	476	1,560	32,240	71,078	63.5	2.25	7,062	15,569
ESS (hot)	219.1	8.625	211.6	8.329	3.8	0.148	534	1,754	10,662	23,504	101.6	4	10,518	23,188
	273.1	10.75	264.7	10.42	4.2	0.165	552	1,812	15,334	33,806	101.6	4	12,698	27,995
	323.9	12.75	314.7	12.39	4.6	0.18	568	1,862	20,412	45,002	101.6	4	14,814	32,659
EPGS (cold)	323.9	12.75	311.2	12.25	6.4	0.25	1,002	3,290	50,244	110,770	63.5	2.25	14,893	32,834
EPGS (hot)	273.1	10.75	264.7	10.42	4.2	0.165	1,286	4,222	35,734	78,784	101.6	4	29,588	65,230
Steam generator (cold)	406.4	16	373.1	14.688	16.7	.656	46	150	7,314	16,125	63.5	2.5	824	1,817
(hot)	406.4	16	381	15.0	12.7	0.5	85	280	10,511	23,173	101.6	4	2,660	5,863

Table 4-18

HEAT TRANSPORT SUBSYSTEM
PUMP CHARACTERISTICS

	ESS Hot	ESS Cold	EPGS Cold
Quantity	6 at 25% capacity	6 at 25% capacity	3 at 50% capacity
Type	Centrifugal, canned suction	Centrifugal, canned suction	Centrifugal, canned suction
Orientation	Vertical	Vertical	Vertical
Speed, rpm	1770	1770	1770
Number of stages	3	3	4
Head, m(ft)	61 (200)	128 (420)	111 (364)
NPSH required at impeller, m(ft)	5.94 (19.5)	5.79 (19)	8.38 (27.5)
Operating temperature, K(F°)	839 (1,050)	567 (561)	567 (561)
Pump power at MGL, kW _e (hp)	137 (169)	442 (526)	513 (611)
Capacity, m ³ /s (gal/min)	0.088 (1,395)	0.188 (1,880)	0.1593 (2,525)
Material:			
Impellers	316 S.S.	316 S.S.	316 S.S.
Bearings	Carbide	Carbide	Carbide
Sleeves	Colmonoy	Colmonoy	Colmonoy

Table 4-19

HEAT TRANSPORT SUBSYSTEM STEAM GENERATOR
HEAT EXCHANGER CHARACTERISTICS

Service	Preheater	Boiler	Superheater	Reheater
Number of shells	2	4	2	1
Type	U-tube and shell	Straight tube and shell	U-tube and shell	U-tube and shell
Orientation	Horizontal	Vertical	Horizontal	Horizontal
Number of tube/shell passes	2/2	1/1	2/2	2/2
Tube side fluid	Molten salt	Water	Steam	Steam
Shell side fluid	Water	Molten salt	Molten salt	Molten salt
Tube material	Carbon steel	347 S.S.	347 S.S.	347 S.S.
Shell material	Carbon steel	347 S.S.	347 S.S.	347 S.S.
Number of tubes	732	1,000	734	1,000
Tube OD/ID/wall, mm (in)	19.05/13.51/2.77 (0.75/.532/.109)	25.4/21.18/2.11 (1.0/.834/.083)	19.05/15.75/1.65 (0.75/.62/.065)	19.05/15.75/1.65 (0.75/.62/.065)
Tube pattern	Rotated triangular	Rotated triangular	Square	Square
Pitch dimension, mm (in)	20.64 (13/16)	31.75 (1-1/4)	25.4 (1)	25.4 (1)
Straight length, m (ft)	12.19 (40)	6.1 (20)	8.53 (28)	7.62 (25)
Shell diameter, m (ft)	1.09 (3.58)	1.09 (3.58)	1.27 (4.17)	1.37 (4.5)
Operating mass per shell, kg (lb)	36,287 (80,000)	15,876 (35,000)	27,216 (60,000)	27,216 (60,000)
Gross surface area per shell, m ² (ft ²)	1,110 (11,949)	410 (4,417)	842 (9,067)	916 (9,864)
Transfer rate, W/m ² -°C (Btu/h-ft ² -°F)	1,420 (250)	1,578 (278)	1,136 (200)	767 (135)
LMTD, K (°F)	22.4 (40.4)	24.6 (44.3)	44.4 (80)	63.6 (114.4)
Pinch point Δt , °C (°F)	5.6 (10)	5.6 (10)		
<u>Steam Drum</u>				
Quantity		1		
Shell ID, m (ft)		2.13 (7)		
Straight length, m (ft)		12.19 (40)		
Shell material		Carbon steel		
Internals		Moisture separator		

Table 4-20

HEAT TRANSPORT SUBSYSTEM OPERATIONAL
CHARACTERISTICSWORKING FLUID FLOWRATES, 10^3 kg/h (10^3 lb/h)

	Maximum	Design	Minimum
Total	5,400 (11,890)	5,400 (11,890)	1,620 (3,567)
Cold salt lines, mm (in)			
76 (3)	1.81 (3.99)	1.81 (3.99)	0.543 (1.197)
102 (4)	2.27 (4.99)	2.27 (4.99)	0.681 (1.497)
152 (6)	4.86 (10.68)	4.86 (10.68)	1.458 (3.204)
203 (8)	9.71 (21.37)	9.71 (21.37)	2.913 (6.411)
254 (10)	16.19 (35.61)	16.19 (35.61)	4.857 (10.68)
305 (12)	25.90 (56.98)	25.90 (56.98)	7.770 (17.09)
356 (14)	35.61 (78.34)	35.61 (78.34)	10.68 (23.50)
Hot salt lines, mm (in)			
76 (3)	1.81 (3.99)	1.81 (3.99)	0.543 (1.197)
102 (4)	3.28 (7.12)	3.28 (7.12)	0.984 (2.136)
152 (6)	8.09 (17.80)	8.09 (17.80)	2.427 (5.340)
203 (8)	13.92 (30.62)	13.92 (30.62)	4.176 (9.186)
254 (10)	22.66 (49.85)	22.66 (49.85)	6.798 (14.96)
305 (12)	42.08 (92.59)	42.08 (92.59)	12.62 (27.78)

PRESSURE, kPag (psig)

Cold salt lines			
Main headers	3,102 (450)	2,386 (346)	1,138 (165)
Risers	3,102 (450)	2,386 (346)	1,138 (165)
Receiver headers	1,625 (235)	1,250 (181)	0 (0)
Hot salt lines			
Main headers	1,338 (194)	1,027 (149)	1,027 (149)
Downcomers	1,338 (194)	1,027 (149)	0 (0)
Receiver headers	724 (105)	558 (81)	0 (0)

PRESSURE DROP, kPag/M (psig/100 ft)

Cold salt lines, mm (in)			
76 (3)	0.529 (2.340)	0.529 (2.340)	0.0476 (0.211)
102 (4)	0.217 (0.959)	0.217 (0.959)	0.0195 (0.086)
152 (6)	0.095 (0.421)	0.095 (0.421)	0.0086 (0.038)
203 (8)	0.100 (0.441)	0.100 (0.441)	0.0090 (0.040)
254 (10)	0.128 (0.565)	0.128 (0.565)	0.0115 (0.051)
305 (12)	0.223 (0.984)	0.223 (0.984)	0.0201 (0.089)
356 (14)	0.308 (1.361)	0.308 (1.361)	0.0277 (0.122)
Hot salt lines, mm (in)			
76 (3)	0.054 (0.237)	0.054 (0.237)	0.0049 (0.021)
102 (4)	0.187 (0.826)	0.187 (0.826)	0.0168 (0.074)
152 (6)	0.278 (1.229)	0.278 (1.229)	0.0250 (0.111)
203 (8)	0.100 (0.446)	0.100 (0.446)	0.0090 (0.040)
254 (10)	0.498 (2.201)	0.498 (2.201)	0.0448 (0.198)
305 (12)	0.155 (0.684)	0.155 (0.684)	0.0140 (0.062)

Table 4-20. (concluded)

**STEAM GENERATOR HEAT EXCHANGER DISCHARGE RATE,
PER UNIT (TOTAL), MW_{th}**

<u>Section</u>	<u>Maximum</u>	<u>Design</u>	<u>Minimum</u>
Preheater	37.7 (75.4)	37.7 (75.4)	10.7 (21.4)
Boiler	15.9 (63.6)	15.9 (63.6)	4.50 (18.0)
Superheater	45.7 (91.4)	45.7 (91.4)	12.9 (25.9)
Reheater	46.7 (46.7)	46.7 (46.7)	13.2 (13.2)

**WORKING FLUID OPERATING TEMPERATURES, IN/OUT,
K (°F)**

<u>Section</u>	<u>Temperature In</u>	<u>Temperature Out</u>
Preheater	638 (688)	567 (561)
Boiler	708 (814)	638 (688)
Superheater	839 (1,050)	708 (814)
Reheater	839 (1,050)	708 (814)

**WATER/STEAM OPERATING TEMPERATURES, IN/OUT,
K (°F)**

<u>Section</u>	<u>Temperature In</u>	<u>Temperature Out</u>
Preheater	506 (450)	632 (678)
Boiler	632 (678)	632 (678)
Superheater	632 (678)	811 (1,000)
Reheater	577 (579)	811 (1,000)

Table 4-21

HEAT TRANSPORT SUBSYSTEM WORKING FLUID AND WATER/STEAM
OPERATING PRESSURES AT MGL, MPag (psig)

	Salt				Water/Steam			
	In		Out		In		Out	
	MPag	psig	Mpag	psig	Mpag	psig	MPag	psig
Preheater	0.43	62	0.32	47	18.41	2670	18.20	2640
Boiler	0.40	58	0.48	69	18.24	2646	18.32	2657
Superheater	0.54	78	0.40	58	18.32	2657	17.75	2575
Reheater	0.54	78	0.40	58	2.9	421	2.6	377

Table 4-22

HEAT TRANSPORT SUBSYSTEM WORKING FLUID AND WATER/STEAM
FLOW RATES PER SHELL, 10^3 kg/h (10^3 lb/h)

	Salt				Water/Steam			
	Max (Design)		Minimum		Max (Design)		Minimum	
	kg/h	lb/h	kg/h	lb/h	kg/h	lb/h	kg/h	lb/h
Preheater	1200	2645	279	615	176	389	40	88
Boiler	600	1323	140	308	88	194	20	44
Superheater	1583	3490	335	738	351	773	80	176
Reheater	816	1800	223	492	314	693	68	151

Table 4-23

HEAT TRANSPORT SUBSYSTEM STEAM GENERATOR OPERATIONAL
CHARACTERISTICS AT DESIGN POWER

Service	Preheater	Boiler	Superheater	Reheater
Heat transfer coefficients, $\text{W/m}^2\text{-}^\circ\text{C}$ ($\text{Btu/h-ft}^2\text{-}^\circ\text{F}$)	1,420 (250)	1,578 (278)	1,136 (200)	767 (135)
Corrosion allowance, mm (in)				
Tube side	3.18 (1/8)	3.18 (1/8)	-	-
Shell side	3.18 (1/8)	-	-	-
Shell fouling, $\text{m}^2\text{-}^\circ\text{C/W}$ ($\text{h-ft}^2\text{-}^\circ\text{F/Btu}$)	0.00018 (0.001)	0.00018 (0.001)	0.00018 (0.001)	0.0018 (0.001)
Design power per shell, 10^6W (10^6 Btu/h)	37.7 (128.48)	15.9 (54.18)	45.7 (156.02)	46.7 (159.31)

Calcium silicate insulation with a weatherproof aluminum jacket used in conventional power plants has been selected for the hot and cold lines. Capital costs for the insulation were developed from historical data for each of the anticipated cold pipe diameters with insulation thicknesses of 38 mm (1.5 inches) to 76 mm (3.0 inches) and for each of the anticipated hot pipe diameters with thicknesses of 38 mm (1.5 inches) to 127 mm (5.0 inches).

Heat losses were calculated for each combination of pipe diameter and insulation thickness based on the following parameters:

- Cold pipe insulation thermal conductivity $0.059 \text{ W/m-}^{\circ}\text{C}$ ($0.034 \text{ Btu/hr-ft-}^{\circ}\text{F}$)
- Hot pipe insulation thermal conductivity $0.078 \text{ W/m-}^{\circ}\text{C}$ ($0.045 \text{ Btu/hr-ft-}^{\circ}\text{F}$)
- Jacket-to-air heat transfer coefficient $11.4 \text{ W/m}^2\text{-}^{\circ}\text{C}$ ($2.0 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$)
- Ambient temperature 292 K (65°F)

The incremental capital cost of the collector and receiver subsystems required to supply the heat loss through the insulation was estimated to be $\$250/\text{kW}_e$ or $\$0.0735/\text{Btu/h}$.

For each line size, the minimum total of the capital cost of the insulation and the corresponding capital cost of the heat loss defined the optimum insulation thickness. The optimum thicknesses are summarized in Table 4-24.

4.4.2.1.2 Optimum Line Diameters

The selection of the optimum line diameter for each section of the field piping involves a tradeoff between five components of the total cost of the piping. The components include the following:

- Material and installation cost of the pipe
- Material and installation cost of the pipe insulation
- Incremental cost of the collector and receiver subsystems required to supply the heat loss through the insulation

Table 4-24

OPTIMUM PIPE INSULATION THICKNESS

Pipe Temperature, K (°F)	Nominal Pipe Diameter (in)	Insulation Thickness, mm (in)
567 (561)	2,3,4,6,8	51 (2.0)
	10,12,14	64 (2.5)
839 (1050)	2,3,4	64 (2.5)
	6	76 (3.0)
	8	89 (3.5)
	10,12	102 (4.0)

- Incremental cost of the plant required to compensate for pressure drops in the field piping
- Electric heat tracing for freeze protection

Capital cost estimates for the stainless steel pipe, carbon steel pipe, and heat tracing were based on information obtained from the manufacturers. Estimates for the capital cost of the insulation and the incremental collector and receiver subsystems were taken directly from the optimization analysis for the insulation. The incremental cost of the plant required to compensate for pipe pressure drops was calculated as follows:

- Pressure drops for each of the anticipated pipe sizes were calculated for a number of flow rates
- Power requirements were calculated for each combination of line size, flow rate, and pressure drop
- Incremental cost of the plant to meet each power requirement was estimated at \$1390/kW_e
- Five capital cost components, for each anticipated line size, were then added and plotted as a function of salt flow rate.

Typical total pipe cost curves for the cold lines and hot lines are shown in Figure 4-36.

FIGURE 4-36 OPTIMUM HTS PIPE DIAMETERS

These curves define the optimum line size for each salt flow rate within the field. For example, the salt flow rate in the EPGS cold main header is approximately $0.11 \text{ m}^3/\text{s}$ ($4.0 \text{ ft}^3/\text{s}$); therefore, the line size that results in the lowest total pipe cost is a 254 mm (10 inch) diameter line. The other field piping line sizes in Figure 4-34 were determined in a similar manner. The only exceptions to this sizing criterion are the ESS receiver header downcomers and the ESS hot main headers. These line sizes were calculated to throttle the large receiver height salt static head as the salt enters the atmospheric storage tanks.

These curves also illustrate a number of design features in the line-focus power plant. First, for a particular pipe diameter, the hot salt line has a higher flow rate and, consequently, a greater pressure drop per foot than the cold salt line. The stainless steel of the hot line is more expensive than the carbon steel of the cold line. Therefore, a higher, and more costly, pressure drop may be afforded in the hot line before an increase in pipe diameter is justified. Second, optimum salt velocities in the large diameter lines are in the conventional range of 3.05 m/s (10 ft/s) to 3.65 m/s (12 ft/s). Optimum velocities in the small diameter lines, however, may approach 0.305 m/s (1 ft/s). Typical design velocities of about 1.22 to 1.52 m/s (4 to 5 ft/s) in the small lines result in moderate pressure drops. However, due to the high cost of supplying power to the salt pumps and to the relatively small incremental cost in increasing the pipe diameter, optimum pressure drops, and consequently salt velocities, are very low in the small diameter lines. Third, the optimum line sizes are influenced by the solar multiple of the plant. As the solar multiple is increased, the size of the plant and the peak salt flow rate are each increased. This, in turn, increases the auxiliary power requirements for the salt pumps and increases the incremental plant costs to supply this power. To compensate for this increase in pumping costs, line pressure drops will be decreased, thereby increasing optimum line diameters.

4.4.2.1.3 Field Piping Schedules

Pipe schedules for the HTS were selected on the basis of maximum anticipated pressures in each section of the field piping and the allowable stress levels published in ANSI B31.3 Standards. Schedule 20 has been selected for the EPGS cold main headers, EPGS receiver header risers, ESS cold main headers, and ESS receiver header risers. Schedule 10S has been selected for the balance of the hot and cold field piping.

4.4.2.1.4 Thermal Expansion Loops

To accommodate the thermal expansion in the salt lines, conventional expansion loops are provided in each section of the field piping. Expansion loops were favored over expansion bellows to achieve a high subsystem reliability. Allowable pipe stress levels, which determine the frequency and size of the expansion loops, are based on ANSI B31.3 Standards. An additional criterion was used in sizing the expansion loops for the receiver headers. The expansion loops are supported by the receiver trusses and the width of the trusses set a limit on the allowable length of the expansion loops. The length of the expansion loop, in turn, determined the distance between the loops. The expansion loops are specified in Figure 4-34.

4.4.2.1.5 Pipe Supports

The EPGS and ESS hot and cold main headers are located below grade to allow the mirror washing machines to move freely along the heliostat rows. The headers are located in concrete-lined trenches with vertical walls. The trenches are covered with 6.35-mm (0.25-inch) checkered steel plate to provide a bridge for the mirror cleaning machines and to limit the rainwater entering the trenches. The pipes are supported on steel I-beams which span the trench. The beams are located approximately 150 mm (6 inches) from the bottom of the trench to prevent water that enters the trench from flooding the pipe insulation. A steel T-beam, approximately 460 mm (18 inches) long and 152 mm (6 inches) deep, is welded to the bottom of the pipe. This T-beam,

which pierces the insulation, rides on the steel I-beam and provides the actual pipe support. This support arrangement is used for two reasons. First, the pipe support loads are carried directly by the pipe rather than being transferred to the pipe through the mildly compressive calcium silicate insulation. Second, rubbing wear, resulting from the daily thermal expansion and contraction of the salt lines, is considerably less for the still I- and T-beam combination than for the alternative arrangement, which sets the aluminum jacket of the pipe insulation directly on the I-beam. The principal disadvantage of the selected support arrangement is that the T-beam acts as a thermal fin and increases the pipe heat losses at each support. Nevertheless, the total increase in heat loss is estimated to be small compared to the total heat loss of the main headers.

The hot and cold receiver headers are supported by the receiver trusses with a steel I- and T-beam similar to that used with the main headers. The receiver header risers and downcomers are supported at the expansion loops by the receiver towers, again with an arrangement similar to that used with the main headers.

4.4.2.1.6 Pipe Anchors

The pipe anchors for the main headers, risers, downcomers, and receiver headers are identical to the respective pipe supports with one exception. Short steel I-beam sections are welded to the bottom of the T-beam section such that they straddle the main I-beam section. The short I-beam sections effectively limit the forward and reverse movement of the pipe. A clearance of approximately 1.6 mm (1/16 inch) is provided between the short and main I-beam sections to help reduce stresses in the pipe near the anchor.

4.4.2.1.7 Valves

Valve service required by the HTS includes the following:

- Small isolation valves for the heat receiver modules
- Isolation valves for HTS components

- Check valves for pump discharge lines
- Control valves for steam generator flow balancing and for steam generator flow control during startup and shutdown.

All shutoff valves should have a low pressure loss in the open position and should provide a reliable, tight seal. Simple internal valve geometries are preferred--those that minimize stagnant salt volumes, promote drainage, and that tend to scrape debris from sealing surfaces. For long packing life, rotating valve stems are preferred over translating valves.

A number of valve designs were considered. Globe valves were rejected because of their relatively large pressure loss and adverse effect upon pipe draining. Gate valves were rejected because of their translating valve stems. Two valve designs were selected as satisfactory for 567 K (561°F) and 839 K (1050°F) service in draw salt. One is the ball valve design, available from Kamyr Valves Inc. and Camaron Iron Works. The other, a tight-sealing butterfly design, available from Clow Corporation, incorporates a positively loaded conical valve seat.

The ball valve supplied by Kamyr incorporates a stellite seal ring with a graphite back seal. The stem packing is made of graphite or Grafoil. The stem and ball are of one-piece construction. The stem and bonnet are extended to accommodate operation at high temperature. This valve has been used in applications up to 1088 K (1500°F) with an abrasive media.

The tight-sealing butterfly valve supplied by Clow can be used as a stop valve, a combined stop-check valve, or as a stop-control valve. This valve has also been used in high-temperature applications, including a 1088 K (1500°F) wind tunnel.

Budgetary prices from valve suppliers indicated that the ball valve is somewhat less expensive than the butterfly for valves 356 mm (14 inches) and smaller. Therefore, ball valves were selected for the HTS conceptual design. A more extensive evaluation of both valves, as would be conducted in a preliminary design, could result in the selection

of the butterfly valve for HTS valves that are 254 mm (10 inches) or larger.

Power to drive the motor-operated valves in the field and in the turbine building comes from a number of sources. The 51-mm (2-inch) diameter isolation valves on the inlet and outlet to each pair of receivers are supplied with power from the electric distribution system for the receiver header heat tracing. The 152-mm (6-inch), 203-mm (8-inch), and 254-mm (10-inch) diameter isolation valves on the risers and downcomers are supplied with power from the electric distribution system for the main header heat tracing. Salt pump and steam generator isolation and control valves are supplied by local distribution panels.

4.4.2.1.8 Salt Freeze Protection

Three systems of freeze protection for the field piping have been included in the conceptual design. The first system employs electric heat tracing and is used during field piping shutdowns of 3 hours or less. The second system involves the circulation of cold salt and is used for shutdown periods of 3 hours to 10 days. The third system involves the draining of the field piping and the steam generator and is used only for extended shutdown periods greater than 10 days.

The first system uses electric heat tracing on all salt lines. Because of the high cost of supplying energy to this system, it is used only during the initial fill procedure for the field piping and for emergency freeze protection. Details of the electric equipment supplying power to the heat tracing are shown in the electrical single line diagram Figure 4-37. Ten groups of ten transformers (one group per receiver row), each rated at 112.5 kVA, 3 phase, 60 Hz, 4160-480Y/277 V, supply power for heat tracing the receiver and main headers. Six 4160-V feeders supply the transformers connected in a loop feed arrangement. One of the feeders serves as a spare, thereby providing an additional degree of circuit reliability.

The second freeze protection system circulates cold salt through the hot and cold field piping. During the parametric analysis, a brief economic study was conducted to determine the most economic method of freeze protection for field shutdown periods lasting 3 hours to 10 days. The methods included field draining to the ESS tanks, electric heat tracing, salt circulation with the thermal energy provided by an oil-fired heater, and salt circulation with the thermal energy provided by additional heliostat area and receiver length. Field draining, the most economic method, was rejected due to the operational problems in completely draining and filling the field piping each day. The heat tracing method was rejected because of its high annual cost. Assuming a levelized fuel oil cost for the 30-year period from 1985 to 2015, the salt circulation method using the oil-fired heater proved more expensive than the method based on additional heliostat area and receiver length. This final method, therefore, using additional heliostat area and receiver length was selected for the conceptual design.

Following each daily field shutdown, the 76-mm (3-inch) diameter cold receiver header sections approach freezing temperatures in approximately 3 hours. At this time, bypass valves, which connect the ends of the hot and cold receiver headers in each receiver row, are opened. Salt circulation pumps are then started, which pump cold salt from the ESS tanks into the cold main headers. Salt circulates from the cold main headers through the receiver header risers, cold receiver headers, bypass line, hot receiver headers, receiver header downcomers, and hot main headers; it then returns to the cold salt tank. The circulation flow rate is controlled such that the salt returning from the field is at least 28 K (50°F) above the freezing temperature of 494 K (430°F). Based on an overnight circulation heat loss of 9.78 MW (33.4×10^6 Btu/h), the required salt flow rate for the entire plant is approximately $0.0738 \text{ m}^3/\text{s}$ (1170 gal/min). Before it returns to the cold tank, the 522 K (480°F) salt from the field is reheated to the 567 K (561°F) cold tank temperature by mixing it with a $0.0063 \text{ m}^3/\text{s}$ (100 gal/min) salt flow from an adjacent hot tank. This energy in the salt flow from the hot tank must be supplied each day by the additional collector area and receiver length.

Because of the temperature limitations of the carbon steel piping, salt from the cold storage tank must be used in the circulation freeze protection system. However, the use of cold salt, as opposed to hot salt, presents three design advantages. First, the circulation heat loss is reduced, which reduces the additional collector area and receiver length requirements. Second, the salt temperature in the hot tanks remains constant at 811 K (1050°F). This minimizes the operational problems in restarting the steam generator and the turbine generator the following day. Third, 3 hours after the field shutdown, the salt in the hot salt lines is at temperatures ranging from 700 K (800°F) to 811 K (1000°F). As the circulation pumps are started, this intermediate temperature salt flows to the cold tanks. Stainless steel spargers are located in the tanks to carefully mix the intermediate temperature salt with the cold salt. This slightly raises the bulk salt temperature in the cold tanks, and therefore reduces the small salt flow from the hot tanks required to maintain the cold tank temperatures at 567 K (561°F).

During the morning field startup, cold salt is purged from the hot salt lines with hot salt from the receivers. The purge salt flow is directed to the spargers in the cold tanks until all hot salt lines reach an operating temperature of 839 K (1050°F). At this time, normal operation of the steam generator and EPGS may begin.

Directing the purge salt to the cold tanks has three thermodynamic benefits. First, the purge salt increases the bulk temperature of the cold storage system. This slightly reduces the energy the heliostats must deliver to the receivers to heat a given salt flow rate to 839 K (1050°F). Second, if the purge salt is directed to the hot, rather than the cold tank, the temperature and availability of the hot salt are decreased. (This would be considered as a system loss.) Third, if the purge salt is directed to the steam generator rather than the cold tank, the steam generated by the purge salt would be dumped to the condenser. This would also be a system loss.

4.4.2.1.9 Heat Transport and Receiver Subsystem Drainage

This, the third system of freeze protection, is used only during extended shutdowns. In this system, the salt in the field piping, steam generator, and receivers is drained to the salt storage tanks. Oil-fired salt heaters, adjacent to the tanks in a small circulation loop, prevent the salt from freezing in the tanks. This method of freeze protection is used because the instantaneous heat loss from the salt storage tanks is considerably less than the heat loss from the field piping and the receivers. During an extended shutdown, this significantly reduces the fuel oil consumption for freeze protection, which justifies the operational problems in draining and refilling the field piping and the receivers.

The receiver field is divided into two drain sections. The field north of the EPGS and the steam generator drains to the north salt tanks; the field south of the EPGS drains to the south salt tanks. The hot and cold receiver headers slope toward the risers and downcomers with an overall change in elevation of 0.92 m (3 ft). The hot and cold main headers slope toward the salt tanks. The overall change in elevation is also 0.92 m (3 ft).

When the field is to be drained, air vents are opened at the end of each receiver header and on each receiver section. The cold receiver headers, risers, and cold main headers drain to a sump adjacent to the cold salt tanks. The sump, approximately 3.7 m (12 ft) below grade, is designed as an atmospheric tank to maintain a constant suction head for the pumps that transfer salt from the sump to the storage tanks. Therefore, drain flows from the field are throttled as they enter the sump to dissipate the large static head from the receivers and to maintain a constant elevation in the sump. The receivers, hot receiver headers, downcomers, and hot main headers drain to a comparable sump adjacent to the hot salt tanks. The decision to drain the field piping and the receivers generally occurs after cold salt circulation for freeze protection has been started.

Under these conditions, all of the field piping is at approximately 538 K (510°F) and there is little need to differentiate between the salt drains from the cold and hot lines. During an emergency condition, however, the decision to drain the field piping and the receivers could occur when the cold and hot lines are at the normal operating temperatures. Under these conditions, there is a need to separate the cold and hot drains to ensure that high-temperature salt does not touch the carbon steel of the cold salt system. For this reason, separate cold and hot drain systems are provided.

4.4.2.1.10 Heat Transport and Receiver Subsystem Filling

The field piping, steam generator, and receivers must be filled with molten salt following the plant construction or an extended field shutdown. To ensure that the salt does not freeze during the filling process, the field components must be preheated to approximately 533 K (500°F). The field piping is preheated by the electric heat tracing. The steam generator is preheated by electric heat tracing of the exchanger shells and by a small, pressurized water circulation system with an integral electric heater. The receivers are preheated by focusing a limited number of collectors on the receiver tubes.

Once the heat transport and receiver subsystems reach 533 K (500°F), vents are opened on each receiver section; salt from the cold storage tanks is then pumped into the field piping. As the salt reaches each vent, the vent closes automatically. This continues until all air is expelled from the field subsystems, at which time the salt can be circulated in the freeze protection mode (outlined earlier) or it can be brought to operating temperature for electric power generation and thermal storage.

4.4.2.2 Primary Salt Pumps

During the parametric analysis, the following criteria for the selection of the hot and cold salt pumps were outlined:

- High reliability at the operating temperatures
- Continuous operation at flow rates ranging from one-third capacity to full capacity
- High efficiency at flow rates ranging from one-third capacity to full capacity

Salt pumps with high reliability are desirable in a solar thermal power plant because such pumps ensure that the collector and receiver subsystems deliver the maximum thermal energy to the steam generator. Continuous pump operation at high efficiency and over a wide range of flow rates is also desirable for two reasons. First, due to the solar insolation curve, the required salt flow rate varies widely each day. Second, the cost of supplying auxiliary power to the salt pumps is high.

These criteria were used to evaluate a number of pump and drive combinations in the conceptual design, including horizontal multistage, vertical cantilever, and vertical multistage. Horizontal multistage pumps, in which the seals and bearings are exposed to the salt, are suitable for cold salt. Pump seals, however, have not yet been developed for hot salt. Horizontal pumps that produce a refined flow rate are commercially available; they produce the required flow rate and head and were tentatively selected as the EPGS and ESS cold salt pumps.

Vertical cantilever pumps are suitable for use with hot salt and are commercially available. In these pumps, the bearing and seals are separated from the impeller by a cantilever arrangement. The pumps, however, are limited to single-stage designs. Preliminary total developed head requirements for the hot salt pumps showed that single-stage pumps could not be used. Therefore, multiple-stage pumps were evaluated for the conceptual design.

Vertical multistage pumps were also found to be suitable for use with hot salt, and they, too, are available commercially. In these pumps, the bearings run directly in the salt, but the seals are separated from the salt by an air cover gas system. Pumps that meet the flow rate and head requirements were selected for the ESS hot salt pumps. A brief economic study was also conducted comparing the use of horizontal

and vertical multistage designs for the EPGS and ESS cold salt pumps. The analysis involved a tradeoff between a lower-cost, lower-efficiency horizontal design with a higher-cost, higher-efficiency vertical design. The vertical pumps were the most economic choice and were thus selected for the conceptual design.

The pump-drive combinations that were considered included multiple small pumps, variable speed drives using fluid couplings, variable speed drives using motor frequency control, and variable speed drives using a hydraulic pump and motor. Multiple small pumps, such as those having five units at 25% capacity, could provide continuous operation at high efficiency by bringing the pumps on- or off-line as required. However, this arrangement was not used for two reasons. First, the small pumps and corresponding piping, valves, and switchgear would be expensive. Second, frequent starting and stopping of the pump motors could shorten their life. Variable speed drives using fluid couplings would reduce the number of pumps to perhaps three units having 50% capacity and would allow continuous pump operation. However, the part load efficiency of fluid couplings is very low. For this reason, these drives were omitted from further consideration.

The variable speed drives using motor frequency control or a hydraulic pump and motor satisfy the criteria for continuous operation at part loads with high efficiencies. The selection of the variable speed drive was based on a brief economic comparison among drive efficiencies and costs. The frequency control drive not only includes the frequency control unit but it also requires a more expensive wound rotor motor. An overall drive efficiency, between the power input to the motor and shaft horsepower to the pump, of 90% over a significant portion of the operating range may be achieved, however. The hydraulic pump and motor drive includes a conventional induction motor, a variable displacement piston hydraulic pump, a fixed displacement piston hydraulic motor, interconnecting lines, and an accumulator. Salt pump speed control is achieved by changing the displacement of

the hydraulic pump. With an induction motor efficiency of 93% and a hydraulic pump and motor efficiency of 82%, an overall drive efficiency of 76% over a significant portion of the operating range may be expected. The performance and economic tradeoff showed the two drive systems were approximately equal. On a somewhat arbitrary basis, the hydraulic pump and motor system was selected for the conceptual design.

The primary salt pump characteristics developed for the calculation of the auxiliary power requirements are summarized in Table 4-25. Specifications for the primary salt pumps are shown in Table 4-18.

Table 4-25

PRIMARY SALT PUMP CHARACTERISTICS

Characteristic	Primary Salt Pump		
	EPGS Cold	ESS Cold	ESS Hot
Number	3	6	6
Flow rate, m ³ /s (gal/min)	0.159 (2525)	0.119 (1880)	0.088 (1390)
Total developed head, m (ft)	111 (364)	128 (420)	66 (215)
Pump efficiency, %	84	82	85
Pump power, kW (shp)	392 (526)	346 (463)	114 (153)
Drive efficiency, %	82	82	82
Motor power, kW (bhp)	478 (641)	421 (564)	140 (187)
Rated motor power, kW (BHP)	450 (600)	375 (500)	150 (200)
Motor power consumption (kW _e)	519	457	152

Specifications for the primary salt pumps are shown in Table 4-18.

4.4.2.3 Steam Generator

The conceptual design of the steam generator is based on separate shell and tube heat exchangers for the preheaters, boilers, superheaters, and reheater. The heat exchangers are similar in construction to high quality process equipment. The general arrangement of the steam generator is shown in Figure 4-38. Detailed design information for each section of the steam generator is summarized in the heat exchanger specification sheets, Tables 4-26 through 4-29.

Design considerations included the selection of the materials, tubes, tubesheet welds, baffles, and method of freeze protection. The tube, tubesheet, and shell materials were selected to minimize corrosion of the respective metals. The steam generator design assumes that carbon steel is suitable for use at water and salt temperatures below 672 K (750°F) and that the stabilized grades of austenitic stainless steel are satisfactory for use at steam and salt temperatures between 672 K (750°F) and 839 K (1050°F). The final selection of the heat exchanger materials, however, will depend on further research in the corrosion rates of carbon and stainless steels in draw salt.

U-tube heat exchanger construction has been selected in preference to straight tube construction wherever possible. The advantages of the U-tube design include lower tube and shell stresses due to thermal expansion, the requirement for only one tubesheet, no expansion joints, and a lower capital cost. A potential disadvantage of the U-tube design is that it is more difficult to clean than the straight tube design. The water and salt, however, are both clean fluids and frequent tube cleaning should not be required.

Longitudinal baffles were selected for those heat exchangers in which fluid temperature crosses occur. Temperature crosses occur when the outlet temperature of the hot fluid is lower than the outlet temperature of the cold fluid. By designing the heat exchangers with a temperature cross, the number and capital cost of the heat exchangers may be reduced. The longitudinal baffles are multiple leaf sealing

FIGURE 4-38 ARRANGEMENT OF STEAM GENERATOR HEAT EXCHANGERS

Table 4-26

HEAT EXCHANGER DATA SHEET,
PREHEATER

SERVICE: PREHEATER		EQUIPMENT NO.	
		CONSISTS OF: 2 Parallel SHELLS	
PERFORMANCE: (1 SHELL)		SHELL SIDE	
FLUID		Water	
FLUID QUANT. ENTERING (1,000)		176 kg/hr (389 LB/HR)	
		1,200 kg/hr (2,645 LB/HR)	
SPECIFIC HEAT J/kg-°K (BTU/LB-°F)		1,552.3 (0.371)	
VISCOSITY 10 ³ Pa-sec		0.036 (0.024) AT 686 °F	
(10 ³ lb/ft-sec)		1.197 (0.806) AT 460 °F	
GRAVITY		AT °F	
NUMBER OF PASSES		2	
VELOCITY FT/SEC.		AT °F	
PRESSURE AT INLET kPag (PSIG)		18,409 (2,670)	
PRESSURE DROP kPa (PSI)		48.3 (7)	
FOULING FACTOR		0.00018 (0.001)*	
TEMPERATURE IN °C (°F)		238 (460)	
TEMPERATURE OUT °C (°F)		363 (686)	
HEAT EXCHANGED - BTU/HR: 128.48x10 ⁶		37,700kW LOG. MTD (°F 40.4) 22.4°C CORRECTED MTD °F	
TRANSFER RATE: SERVICE- 1420 (250)**		CLEAN- FOULING FACTOR-OVERALL:	
TRANSFER AREA: GROSS - 1,110 m ²		(11,949SQ.FT.): EFFECTIVE SQ.FT.:	
CONSTRUCTION			
DESIGN PRESSURE - kPag (PSIG)		19,305 (2,800)	
TEST PRESSURE - (PSIG)		1,034 (150)	
DESIGN TEMPERATURE - °C (°F)		388 (730)	
CORROSION ALLOWANCE mm (in.)		3.18 (1/8)	
TUBES - QUANT. 732 U		O.D. 19.05mm BxG 12 LENGTH 12.19mm (40") PITCH 20.64mm (13/16")	
SHELL - 1,092.2mm (43") I.D. (I.O.D.)			
CONNECTIONS: SHELL INLET -		203.2 (8 in.)	
(SIZE & RATING) CHANNEL INLET -		355.6 (14 in.)	
OUTLET -		203.2 (8 in.)	
		355.6 (14 in.)	
CODES AND STANDARDS APPLIED ASME I & TEMA Class R			
WEIGHT: DRY:		FLOODED: kg (lb)	
		OPERATING: 36,287(80,000) BUNDLE ONLY:	
PART	MATERIAL	PART	MATERIAL
TUBES	Carbon Steel	FLAT TUBE SHEET	-
SHELL	"	FIXED TUBE SHEET	Carbon Steel
SHELL COVER	"	TUBE SUPPORTS	
CHANNEL	"	CROSS BAFFLE	Carbon Steel
CHANNEL COVER		LONG BAFFLE	"
FLAT HEAD COVER	-	GASKETS	
MANUFACTURER		FOREIGN PRINT NO.	
MODEL			
VENDOR		LOCATION DWG. NO.	
PURCHASE ORDER NO.		P. & I. DIAGRAM NO.	
REMARKS: * m ² /°C/W (HR-FT ² -°F/BTU)			
** W/m ² -°C (BTU/HR-FT ² -°F)			

Source: Bechtel, Power and Industrial Division.

Table 4-27

HEAT EXCHANGER DATA SHEET,
BOILER

1 SERVICE: BOILER		EQUIPMENT NO.	
		CONSISTS OF: 4 Parallel SHELLS	
3 PERFORMANCE: (1 SHELL)		SHELL SIDE	
4 FLUID		TUBE SIDE	
5 FLUID QUANT. ENTERING (1,000)		600 kg/hr (1,323 LB/HR)	
		88 kg/hr (194 LB/HR)	
10 SPECIFIC HEAT J/kg-°K (BTU/LB-°F)		1,552.3 (0.371)	
11 VISCOSITY 10 ⁻³ Pa-sec		1.65 (1.11) AT 806 °F	
(10 ⁻³ lb/ft-sec)		2.05 (1.38) AT 696 °F	
13 GRAVITY		AT °F	
14 NUMBER OF PASSES		AT °F	
15 VELOCITY FT/SEC.		AT °F	
17 PRESSURE AT INLET kPag (PSIG)		399.9 (58)	
18 PRESSURE DROP kPa (PSI)		103.4 (15)	
19 FOULING FACTOR		0.00018 (0.001)*	
20 TEMPERATURE IN °C (°F)		369 (696)	
21 TEMPERATURE OUT °C (°F)		430 (806)	
22 HEAT EXCHANGED - BTU/HR: 54.18x10 ⁶ 15,900kW LOG. MTD (°F 44.3) 24.6°C CORRECTED MTD °F			
23 TRANSFER RATE: SERVICE: 1578 (278)** CLEAN: FOULING FACTOR-OVERALL:			
24 TRANSFER AREA: GROSS - 410 m ² (4,417 SQ. FT.) EFFECTIVE SQ. FT.:			
25			
26 CONSTRUCTION			
27 DESIGN PRESSURE - kPag (PSIG.)		1,034 (150)	
28 TEST PRESSURE - PSIG.		19,306 (2,800)	
29 DESIGN TEMPERATURE - °C (°F)		466 (870)	
30 CORROSION ALLOWANCE mm. (in.)		3.18 (1/8)	
31 TUBES - QUANT. 1000 Str. 0.0 25.4mm (1") BWG 14 LENGTH 6.1m (20') PITCH 31.75mm (1 1/4")			
32 SHELL 3,092.2mm (43") (I.O.) (O.O.)			
33 CONNECTIONS: -SHELL: INLET - 304.8 mm (12 in.) OUTLET - 304.8 mm (12 in.)			
34 (SIZE & RATING) -CHANNEL: INLET - 508.0 mm (20 in.) OUTLET - 609.6 mm (24 in.)			
35			
36			
37			
38 CODES AND STANDARDS APPLIED ASME I & TEMA Class R			
39 WEIGHT: DRY: FLOODED: kg (lb) OPERATING: 15,876 (35,000) BUNDLE ONLY:			
40 PART	MATERIAL	PART	MATERIAL
41 TUBES	347 SS	FLOAT TUBE SHEET	
42 SHELL	" "	FIXED TUBE SHEET	347 SS
43 SHELL COVER		TUBE SUPPORTS	
44 CHANNEL	Carbon Steel	CROSS BAFFLE	316 SS
45 CHANNEL COVER		LONG BAFFLE	
46 FLOAT HEAD COVER		GASKETS	
47 MANUFACTURER		FOREIGN PRINT NO.	
48 MODEL			
49 VENDOR		LOCATION DWG. NO:	
50 PURCHASE ORDER NO.		P. & I. DIAGRAM NO:	
51 REMARKS: * m ² -°C/M ² HR-FT ² -°F (BTU)			
52 ** W/m ² -°C (BTU/HR-FT ² -°F)			
53			

Source: Bechtel, Power and Industrial Division

Table 4-28

HEAT EXCHANGER DATA SHEET,
SUPERHEATER

1 SERVICE: SUPERHEATER		EQUIPMENT NO.	
2		CONSISTS OF: 2 Series SHELLS 1	
3 PERFORMANCE: (1 SHELL)		SHELL SIDE	
4 FLUID		TUBE SIDE	
5 FLUID QUANT. ENTERING (1,000)		Steam	
6		Salt	
7		351 kg/hr (773 LB/HR)	
8		(3,490 LB/HR)	
9			
10 SPECIFIC HEAT J/kg-°C (BTU/LB-°F)		1552.3 (0.371)	
11 VISCOSITY 10 ³ Pa-sec		1.65 (1.11) AT 806 °F	
12 (10 ³ lb/ft-sec)		0.049 (0.033) AT 686 °F	
13 GRAVITY		0.034 (0.023) AT 1,000 °F	
14 NUMBER OF PASSES		AT °F	
15 VELOCITY FT/SEC.		AT °F	
16		2	
17 PRESSURE AT INLET kPag (PSIG)		537.8 (78)	
18 PRESSURE DROP kPa (PSI)		18,319 (2,657)	
19 FOULING FACTOR		124 (18)	
20 TEMPERATURE IN °C (°F)		0.00018 (0.0001)*	
21 TEMPERATURE OUT °C (°F)		566 (1050)	
22 HEAT EXCHANGED - BTU/HR: 156.02x10 ⁶ 45,700kW LOG. MTD (°F 44.4) 80°C		CORRECTED MTD °F	
23 TRANSFER RATE: SERVICE 1136 (200)*		CLEAN FOULING FACTOR-OVERALL:	
24 TRANSFER AREA: GROSS 842 m ²		(9,067 SQ. FT.) EFFECTIVE SQ. FT.:	
25			
26 CONSTRUCTION			
27 DESIGN PRESSURE - kPag (PSIG)		1,034 (150)	
28 TEST PRESSURE - PSIG		19,306 (2800)	
29 DESIGN TEMPERATURE - °C (°F)		593 (1,100)	
30 CORROSION ALLOWANCE		566 (1050)	
31 TUBES - QUANT. 734 U		0.0 19.05mm (.75") BWG 16 LENGTH 8.53mm (28') PITCH 25.4mm (1 in.)	
32 SHELL - 1,270mm (50") (I.O.) (O.D.)			
33 CONNECTIONS: -SHELL: INLET -		457.2 mm (18 in.)	
34 (SIZE & RATING) -CHANNEL: INLET -		203.2 mm (8 in.)	
35		OUTLET - 457.2 (18 in.)	
36		OUTLET - 203.2 mm (8 in.)	
37			
38			
39 CODES AND STANDARDS APPLIED		ASME I & TEMA Class R	
40 WEIGHT: DRY:		FLOODED: kg (lb)	
41 PART		OPERATING: 27,216 (60,000) BUNDLE ONLY:	
42 TUBES		347 S.S.	
43 SHELL		"	
44 SHELL COVER		"	
45 CHANNEL		"	
46 CHANNEL COVER		"	
47 FLOAT HEAD COVER		"	
48 MANUFACTURER		FOREIGN PRINT NO.	
49 MODEL		LOCATION DWG. NO:	
50 VENDOR		P. & I. DIAGRAM NO:	
51 PURCHASE ORDER NO.			
52 REMARKS:		Longitudinal Flow: ** W/m ² -°C (BTU/HR-FT ² -°F)	
53		* m ² -°C/W (HR-FT ² -°F/BTU)	

Source: Bechtel, Power and Industrial Division

Table 4-29

HEAT EXCHANGER DATA SHEET,
REHEATER

1 SERVICE: REHEATER		EQUIPMENT NO.	
2		CONSISTS OF: 1 SHELL	
3 PERFORMANCE: (1 SHELL)		SHELL SIDE	
4 FLUID		TUBE SIDE	
5 FLUID QUANT. ENTERING (1,000)		816 kg/hr (1,800 LB/HR)	
6		314 kg/hr. (693 LB/HR)	
7			
8			
9			
10 SPECIFIC HEAT $J/kg \cdot ^\circ K$ (BTU/LB $^\circ F$)		1,552.3 (0.371)	
11 VISCOSITY 10^{-3} Pa-sec.		1.65 (1.11) AT 806 $^\circ F$	
12 $(10^{-3}$ lb/ft-sec)		0.78 (0.524) AT 1,050 $^\circ F$	
13 GRAVITY		AT $^\circ F$	
14 NUMBER OF PASSES		2	
15 VELOCITY FT/SEC.		AT $^\circ F$	
16			
17 PRESSURE AT INLET kPag (PSIG)		537.8 (78)	
18 PRESSURE DROP. kPa (PSI)		48 (7)	
19 FOULING FACTOR		0.00018 (0.001)*	
20 TEMPERATURE IN $^\circ C$ ($^\circ F$)		566 (1,050)	
21 TEMPERATURE OUT $^\circ C$ ($^\circ F$)		430 (806)	
22 HEAT EXCHANGED - BTU/HR: 159.31×10^6 46,700kW LOG. MTD ($^\circ F$ 114.4) 45.8 $^\circ C$ CORRECTED MTD $^\circ F$			
23 TRANSFER RATE: SERVICE 767 (135)**		CLEAN- FOULING FACTOR-OVERALL:	
24 TRANSFER AREA: GROSS - 916 m ²		(9,800 SQ.FT.): EFFECTIVE SQ.FT.:	
25			
26 CONSTRUCTION			
27 DESIGN PRESSURE - kPag (PSIG)		1,034 (150)	
28 TEST PRESSURE - PSIG		3,447 (500)	
29 DESIGN TEMPERATURE - $^\circ C$ ($^\circ F$)		593 (1,100)	
30 CORROSION ALLOWANCE		566 (1,050)	
31 TUBES - QUANT. 1,000 U		0.019,05mm(.75") BWG 16 LENGTH 7.62m(25') PITCH 25.4mm(1 in.)	
32 SHELL 1,371.6mm (54") (1.01(0.01))			
33 CONNECTIONS: -SHELL: INLET - 304.8mm (12 in.)		OUTLET - 304.8mm (12 in.)	
34 (SIZE & RATING) -CHANNEL: INLET - 609.6mm (24 in.)		OUTLET - 609.6mm (24 in.)	
35			
36			
37			
38			
39 CODES AND STANDARDS APPLIED		ASME I & TEMA Class R	
40 WEIGHT: DRY:		FLOODED: kg (lb)	
41		OPERATING: 27,216 (60,000) BUNDLE ONLY:	
42 PART		MATERIAL	
43 TUBES		347 S.S.	
44 SHELL		347 S.S.	
45 SHELL COVER		347 S.S.	
46 CHANNEL		347 S.S.	
47 CHANNEL COVER		347 S.S.	
48 FLOAT HEAD COVER		347 S.S.	
49 MANUFACTURER		FOREIGN PRINT NO.	
50 MODEL		LOCATION DWG. NO:	
51 VENDOR		P. & I. DIAGRAM NO:	
52 PURCHASE ORDER NO.			
53 REMARKS:		* $W/m^2 \cdot ^\circ C$ (BTU/HR-FT ² - $^\circ F$)	
		** $m^2 \cdot ^\circ C/W$ (HR-FT ² - $^\circ F$ /BTU)	

Source: Bechtel, Power and Industrial Division

units commonly found in process equipment. A possible alternative to the baffle design is a shroud which surrounds one-half of the tube bundle. Although it would provide a more effective seal, it would also be more expensive. For this reason, the shroud was not used in the conceptual design.

In the preheaters, in which the high-pressure water is in the shell, special tube-to-tubesheet welding has been specified to minimize crevice corrosion where the tube meets the tubesheet. Heat exchanger salt freeze protection is provided by heat tracing the shell and the channel.

4.4.2.3.1 Preheater Design

The preheater is a U-tube design with longitudinal baffles. A major design problem centered around the large difference in water and salt flow rates. A trial design with the low-pressure salt on the shell side resulted in a baffle spacing too large for a sound structural design. The salt, therefore, is placed on the tubeside and the high-pressure water on the shellside. Although the shell thickness is greater than normal for this size of heat exchanger, the shell design is not a significant problem. The preheaters are located on the mezzanine of the steam generator building, directly above the EPGS cold salt pumps.

4.4.2.3.2 Boiler Design

The boiler configuration selected for this study has a vertical shell, straight tubing, and natural circulation. An economic analysis compared the high capital and low operating cost of a natural circulation boiler with the low capital and high operating cost of a pump recirculation boiler. The study indicated that the natural circulation design is the economic choice. The natural circulation requirement results in a vertical-shell, straight-tube design. Expansion joints are provided to accommodate the differential expansion in the tubes and shell. As the result of two space constraints, four boiler shells in parallel have been selected for the conceptual design. The first constraint, the size

of the steam generator building, places a limit on the height of the boiler. The second constraint, the length of the steam drum, places a limit on the maximum number of risers and downcomers. With the straight tube boiler design, the high-pressure water and steam may be located on the tubeside of the heat exchanger. The boilers are located on the mezzanine directly above the preheaters.

4.4.2.3.3 Steam Drum Design

The steam and water from the boilers enter a conventional steam drum located approximately 1.2 m (4 ft) above the boilers on the main operating floor.

4.4.2.3.4 Superheater and Reheater Design

The superheaters and reheater are U-tube designs with longitudinal baffles. The main and reheat steams are on the tubeside to minimize the cost of the heat exchangers. The superheaters and reheater are adjacent to the steam drum on the main operating floor. Sufficient space has been provided on the operating floor and mezzanine of the steam generator building for tube pulling of the superheaters, reheater, and preheaters.

4.4.3 HTS Performance, Operation, and Control

This section briefly discusses the performance, startup, normal operation, shutdown, and control of the field piping and steam generator.

4.4.3.1 HTS Performance

The performance of the HTS addresses the thermal losses of the field piping and the auxiliary power requirements of the primary salt pumps.

4.4.3.1.1 Field Piping Heat Losses

During normal operation, the heat loss from the field piping is approximately 15.3 MW (52.3×10^6 Btu/h). This loss is approximately 2.4% of the thermal energy delivered to the HTS by the receiver

subsystem. During the overnight circulation of cold salt for HTS freeze protection, the field piping heat loss is approximately 9.78 MW (33.4×10^6 Btu/h). This loss is approximately 3.9% of the thermal energy delivered to the steam generator during evening EPGS operation.

4.4.3.1.2 Auxiliary Power Requirements

Table 4-30 summarizes the auxiliary power requirements of the primary salt pumps for various combinations of time-of-day and turbine load conditions. Also shown are the percentages of the gross turbine output that are represented by the auxiliary loads.

Table 4-30

PRIMARY SALT PUMP AUXILIARY POWER REQUIREMENTS

<u>Time of Day</u>	<u>Turbine Load Condition</u>	<u>Salt Pump Power (kW_e)</u>	<u>Percent of Gross Turbine Output (%)</u>
Morning	25% MGL	652	2.5
Noon	MGL	2,793	2.6
	VWO, 5% OP	3,041	2.6
Evening	25% MGL	56	0.2
	MGL	567	0.5

MGL = maximum guaranteed load

VWO = valve wide open

OP = overpressure

Note that the auxiliary power requirements decrease significantly during evening operation.

4.4.3.2 HTS Operation

This section outlines the procedures to fill, start up, shut down and drain the steam generator. The fill, operation, freeze protection,

and drain procedures for the field piping are addressed in Section 4.4.2. Operational considerations for the primary salt pumps are outlined in the following section on HTS control.

4.4.3.2.1 Steam Generator Fill

Following plant construction or an extended shutdown due to cloudy weather, the steam generator is at ambient temperature. Before molten salt can be admitted to the steam generator, the temperature of the unit must be increased to approximately 561 K (550°F). This ensures that the salt does not freeze as it enters the heat exchangers.

Electric heat tracing is used to preheat the shells and heads of the superheaters, reheater, steam drum, boilers and preheaters. To preheat the tubes of the preheaters and boilers and the internals of the steam drum, a small pressurized water recirculation loop with an integral electric heater, shown in Figure 4-39, has been included in the conceptual design.

The molten salt fill procedure begins by filling the preheater shells, boiler tubes, and steam drum with treated feedwater; the superheaters and reheater remain dry. The water loop is then pressurized to 7.1 MPag (1030 psig) and heating with the electric and trace heaters is started. Once the steam generator reaches a uniform temperature of 550°F, salt from the cold storage tank is admitted to the unit. Finally, salt is circulated through the steam generator and the electric and trace heaters are turned off. The steam generator is now in standby.

4.4.3.2.2 Steam Generator Startup

To bring the unit from standby to startup, the preheaters, boilers, and steam drum are pressurized to 14.8 MPag (2140 psig) by the recirculation loop pump. The steam drum only is heated to 617 K (650°F) by the integral electric heater. The salt temperature supplied to the steam generator is raised to 672 K (750°F) by blending cold and hot salt from the storage tanks. The main steam bypass valve to the condenser is opened and steam flows through the superheater. Raising the drum

FIGURE 4-39. AUXILIARY HEATER LOOP

temperature to 617 K (650°F) ensures that water in the preheater does not flash as the bypass line is opened. Once the superheater tubes, headers, and main steamline are free of condensate, main steam is admitted to the turbine and the turbine-generator is synchronized. The salt supplied to the steam generator is raised in temperature at a rate corresponding to the allowable rate of increase in the temperature of the HP turbine. This continues until the salt temperature reaches 839 K (1050°F) and the main steam temperature reaches 811 K (1000°F). At this time, the salt flow rate can be tailored to meet the turbine load conditions.

Salt temperature profiles for the minimum, design, and maximum turbine load conditions are illustrated in Figure 4-40. The superheater and reheater salt inlet temperatures are independent of turbine load. Note that the final feedwater temperature is sensitive to turbine load

FIGURE 4-40 STEAM GENERATOR TEMPERATURE PROFILES

while the temperature of the salt at the outlet of the preheater is not. This operating characteristic guarantees that the salt at the outlet of the preheater does not freeze under low turbine load conditions. This would not be the case if variable pressure steam generator operation had been selected for the conceptual design.

4.4.3.2.3 Steam Generator Shutdown

As the turbine load is reduced from the design level of 100 MW_e to the minimum operating level of 3 to 5 MW_e, the salt flow to the steam generator is also reduced. The temperature of the salt flow, however, remains constant at 839 K (1050°F). Once the turbine reaches the 3- to 5-MW_e load condition, it is immediately tripped. The main steam bypass line to the condenser is then briefly opened to prevent a transient overpressure condition in the steam generator. The steam generator is then cooled to a uniform temperature of 567 K (560°F) and placed in standby. The unit is cooled by ramping down the temperature of the salt supply from 839 K (1050°F) to 567 K (560°F). The intermediate temperature salt supplied for the cooling is produced by blending salt from the hot and cold storage tanks. The intermediate temperature leaving the steam generator during cooldown is directed to the cold storage tank. This stores the residual heat from the steam generator by slightly increasing the temperature of the cold storage tank salt inventory.

4.4.3.2.4 Steam Generator Drain

The steam generator can be maintained in standby for an indefinite period by transferring salt from the cold storage tanks to the unit or by operating the pressurized water recirculation loop. If an extended plant shutdown is anticipated, however, it may be necessary to drain the steam generator.

The drain procedure first involves operating the trace heating and the pressurized water loop. This maintains a steam generator temperature of at least 533 K (500°F) during the drain operation. Vents on the superheater shells, and a bypass line around the EPGS cold salt

pumps, are then opened. Through a suitable arrangement of the heat exchanger elevations, the salt drains from the superheater shells, through the reheater shell, boiler shells, and preheater tubes to the bypass line. From the bypass line, the salt flows through the EPGS cold salt main header to the drain sump adjacent to the northern ESS tanks. The salt within the can portion of the EPGS cold salt pumps remains in the pump and is protected from freezing by electric heat tracing.

Once the salt is removed from the steam generator, the electric heater in the pressurized water loop and the electric heat tracing may be turned off. The water loop remains pressurized, however, to ensure that the water inventory in the preheaters, boilers, and steam drum does not flash. To bring the water inventory temperature to ambient, a small flow of cold water is introduced into the preheater shells. Simultaneously, an equal quantity of high-temperature water from the steam drum is blown down to the condenser. The water injection and blowdown process continues until the steam generator reaches ambient pressure and temperature.

4.4.3.3 HTS Control

The control of the HTS entails the control of the primary salt pumps, the steam generator, and the plant heat tracing.

4.4.3.3.1 Primary Salt Pump Control

The salt flow during plant startup, normal operation, shutdown, and transients is controlled by modulating pump speed in response to speed demand signals generated by the master control subsystem. The MCS computes the demand signals for the primary salt pumps from the following parameters:

<u>Primary Salt Pump</u>	<u>Pump Speed Parameters</u>
EPS cold salt	Turbine load, steam drum pressure
ESS cold salt	Receiver outlet temperature, solar insolation
ESS hot salt	Turbine load, steam drum pressure

The salt pumps are stopped and started in response to changing solar insolation and turbine load conditions. Table 4-31 summarizes the numbers of primary salt pumps in operation for certain combinations of time of day and turbine loads.

Table 4-31

PRIMARY SALT PUMP OPERATION

Time of Day	Turbine Load Conditions	Number of Pumps in Operation		
		EPCS Cold	ESS Cold	ESS Hot
Morning	25% MGL	1	2	0
Noon	MGL	2	4	0
	VWO, 5% OP	3	4	1
Evening	25% MGL	0	0	1
	MGL	0	0	2

The salt pump capacities were calculated such that the design flow rate at each station is provided by two of the three pumps. When the turbine is operating under VWO, 5% OP load conditions, an additional EPCS cold and ESS hot pump must be put into service. Note that during the evening, only the ESS hot salt pumps are in use. They were designed with a sufficient head to transfer the salt from the hot storage tanks, through the steam generator, and return it to the cold storage tanks.

Each of the primary salt pumps has a minimum flow recirculation loop. The system includes a throttle valve and a line returning to the salt storage tank. It is designed to protect the pump by ensuring that the salt flow rate does not fall below a specified value during morning, afternoon, or transient conditions.

4.4.3.3.2 Steam Generator Control

As outlined above, the salt flow rate through the steam generator is controlled by changing the speed of the EPGS cold or ESS hot salt pumps, and the salt temperature is controlled by blending salt from the hot and cold storage tanks. Three additional control systems for the steam generator have been included in the conceptual design.

The first system balances the parallel salt flows through the superheaters and reheater to achieve equal heat exchanger outlet temperatures. The purpose of the control is to avoid mixing salt flows at different temperatures and, therefore, decreasing the availability of the combined salt stream. The pressure drop through the superheaters is greater than the pressure drop of the reheater. The majority of the flow control, therefore, occurs by throttling the reheater salt outlet.

The second system balances the parallel salt flows through the four boiler sections to achieve equal salt outlet temperatures. The third system also balances parallel flows through the two preheaters to achieve equal salt outlet temperatures. Flow controls on parallel heat exchangers have been provided to compensate for differences in manufacturing and for variations in performance due to scaling or tube plugging.

4.5 Energy Storage Subsystem

4.5.1 Subsystem Requirements

The energy storage subsystem is designed to store the thermal energy from the receiver subsystem when this energy is not used directly for electrical power generation. The stored energy is delivered to the EPGS during cloudy periods or at night. In the conceptual design, the EPGS includes the salt storage tanks, ullage control system, salt maintenance system, salt charging system, and salt makeup system.

4.5.2 Energy Storage Subsystem Design

The storage tank and support equipment layout is illustrated in Figure 4-41. The HTS/ESS flow diagram is shown in Figure 4-35. The ESS equipment list is summarized in Table 4-32. Data sheets for the ESS are presented in Tables 4-33 and 4-34.

The conceptual design of the ESS includes the following items:

- Internal or external tank insulation
- Insulation thickness optimization
- Storage volume requirements
- Tank design
- Foundation design
- Salt freeze protection
- Cover gas system
- Salt maintenance, charging, and makeup system

Details in the conceptual design of the energy storage subsystem are outlined in the following sections.

4.5.2.1 Internal or External Tank Insulation

Due to the large thermal storage requirements of the conceptual design, there is a need to develop an inexpensive means of salt storage. Large, field-fabricated carbon steel tanks, similar to petroleum storage tanks, would fill this need. The most stringent design requirement of carbon steel tanks, however, is a maximum tank material

FIGURE 4-41 ENERGY STORAGE SUBSYSTEM

Table 4-32

ENERGY STORAGE SUBSYSTEM EQUIPMENT LIST

Salt Storage Tanks

Cold Salt Tanks

Quantity	2
Dimensions	17.7 m (58 ft) dia x 10.7 m (35 ft) high, hemispheric roof
Material	SA-516-Gr 70 carbon steel
Plate thickness	
Wall	19.1 mm (3/4 in) bottom 3.66 m (12 ft) 14.3 mm (9/16 in) middle 3.66 m (12 ft) 7.9 mm (5/16 in) top 3.35 m (11 ft)
Floor	7.9 mm (5/16 in) 15.9 mm (5/8 in) outer 0.61 m (2 ft)
Roof	6.3 mm (1/4 in) with stiffeners
Connections	1-305 mm (12 in) dia and 1-356 mm (14 in) each with sparger-diffusers
Insulation	76 mm (3 in) calcium silicate with aluminum jacket
Foundation	Compacted dry soil

Hot Salt Tanks

Quantity	4
Dimensions	18.3 mm (60 ft) dia x 11.0 m (36 ft) high, hemispherical roof
Material	SA-240-316 stainless steel
Plate thickness	
Wall	30.2 mm (1-3/16 in) bottom 3.66 m (12 ft) 20.6 mm (13/16 in) middle 3.66 m (12 ft) 11.1 mm (7/16 in) top 3.66 m (12 ft)
Floor	6.3 mm (1/4 in) 19.1 mm (3/4 in) outer 0.46 m (1.5 ft)
Roof	6.3 mm (1/4 in) with stiffeners
Connections	1-305 mm (12 in) dia and 1-356 mm (14 in) 2 tanks 1-254 mm (10 in) dia and 1-305 mm (12 in) 2 tanks each with sparger-diffusers
Insulation	127 mm (5 in) calcium silicate with aluminum jacket
Foundation	Compacted dry soil

Cover Gas System

Quantity	2
Capacity	0.028 m ³ /s (60 ft ³ /min)
H ₂ O desiccant	2-100% trains, silica gel desiccant with electric heat regeneration
CO ₂ desiccant	1-100% train, mono- and diethanol amine desiccant with electric heat regeneration
Interconnecting vents	43 m (140 ft) 102 mm (4 in) dia. Schedule 10S, 316 SS
Valves	1.38 kPag (+0.2 psig) pressure relief -1.38 kPag (-0.2 psig) vacuum relief

Table 4-32 (continued)

Cold Salt Tank Heater

Quantity	2
Type	Oil-fired natural convection with intermediate heat transfer salt
Duty	0.73 MW _{th} (2.5 x 10 ⁶ Btu/h)
Exchange area	23.2 m ² (250 ft ²)
Tubes	256 m (840 ft) of 25.4 mm (1 in) dia, Schedule 40, carbon steel

Hot Salt Tank Heater

Quantity	2
Type	Oil-fired, natural convection with intermediate heat transfer salt
Duty	1.2 MW _{th} (4.0 x 10 ⁶ Btu/h)
Exchange area	37.2 m ² (400 ft ²)
Tubes	4.2 m (1,350 ft) of 25.4 mm (1 in) dia, Schedule 10S, 316 stainless steel

Salt Maintenance System

Cold Salt Circulation

Pumps

Quantity	4
Type	Vertical, multi-stage centrifugal
Materials	316 SS impellers and bowls, carbide bearings
Capacity	0.023 m ³ /s, 23 m t _{dh} (360 gal/min, 75 ft t _{dh})
Drive	15 kW (20 bhp) induction motor

Hot Salt Circulation

Pumps

Quantity	4
Type	Vertical, multi-stage, centrifugal
Materials	316 SS impellers and bowls, carbide bearings
Capacity	0.027 m ³ /s, 17 m t _{dh} (420 gal/min, 55 ft t _{dh})
Drive	11 kW _e (15 bhp) induction motor

NO₂ Bubble Columns

Quantity	2
Material	6.35 mm (1/4 in) carbon steel, 51 mm calcium silicate insulation
Dimensions	0.30 m (1 in) dia x 1.83 m (6 ft) high
Internals	Rings and perforated plates
Comments	Gas vent chamber with salt level control

Cold Salt

Precipitate

Plate and Frame

Quantity	4-50% capacity
Type	Plate and frame

Table 4-32 (continued)

Filters

Material	Carbon steel, 51 mm (2 in) calcium silicate insulation
Flow rate	0.023 m ³ /s (360 gal/min)

Salt Charging and Makeup System**NaNO₃ Storage**

Quantity	2
Construction	203 mm (8 in) thick concrete walls and floor, 4.8 mm (3/16 in). CS gable roof
Dimensions	3.0 m (10 ft) wide x 3.66 m (12 ft) high x 19.8 m (65 ft) long
Comments	1.5 m (5 ft) deep walls below grade, sloped at 45°, to tunnel conveyor

KNO₃ Storage

Quantity	2
Construction	203 mm (8 in) thick concrete walls and floor, 4.8 mm (3/16 in) CS gable roof
Dimensions	3.0 m (10 ft) wide x 3.66 m (12 ft) high x 13.7 m (45 ft) long
Comments	1.5 m (5 ft) deep walls below grade, sloped at 45°, to tunnel conveyor

NaNO₃ Hoppers

Quantity	6
Construction	6.35 mm (0.25 in) carbon steel
Diameter	1.83 m (6 ft)
Height	5.8 m (19 ft)

KNO₃ Hoppers

Quantity	6
Construction	6.35 mm (0.25 in) carbon steel
Diameter	1.52 m (5 ft)
Height	5.18 m (17 ft)

Mixed Salt Hopper

Quantity	2
Construction	6.35 mm (0.25 in) carbon steel
Diameter	1.83 m (6 ft)
Height	5.18 m (17 ft)

Salt Melt Tank

Quantity	2
Construction	6.35 mm (0.25 in) carbon steel
Dimensions	2.44 m (8 ft) high x 3.05 m (10 ft) diameter
Insulation	76 mm (3 in) calcium silicate with aluminum jacket

Table 4-32 (continued)

Molten Salt
Transfer Pump

Quantity	2
Type	Horizontal, centrifugal
Material	316 SS
Capacity	0.00032 m ³ /s, 30 m head (5 gal/min, 100 ft tdh)
Drive	0.4 kW (0.5 bhp) induction motor

Dowtherm - Salt

Quantity	2
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Heat Exchanger

Construction	25.4 mm (1 in) dia, 316 SS tube
Tube Length	134 m (440 ft) coiled in melt tank

Dowtherm Vaporizer

Quantity	2
Duty	0.29 MW _{th} (1.0 x 10 ⁶ Btu/h)
Fuel	No. 2 fuel oil
Capacity	2270 kg/h (5000 lb/h) Dowtherm A at 561 K (550°F)
Comments	Package unit

Dowtherm Flash Tank

Quantity	2
Construction	A106 CS, 6.35 mm (0.25 in) wall
Dimensions	0.76 m (2.5 in) dia x 1.52 m (5 ft) high
Insulation	51 mm (2 in) calcium silicate with aluminum jacket

Dowtherm Condensate
Tank

Quantity	2
Construction	A106, CS, 6.35 mm (0.25 in) wall
Dimensions	0.61 m (2 ft) dia x 1.22 m (4 ft) high
Insulation	51 mm (2 in) calcium silicate with aluminum jacket

Dowtherm Feed Pumps

Quantity	2
Type	Horizontal, centrifugal
Material	Cast steel
Capacity	0.0011 m ³ /s, 81 m tdh (18 gal/min, 265 ft tdh)
Drive	1.5 kW (2 bhp) induction motor

Table 4-32 (concluded)

Salt Conveyors

	<u>Tonnes/h</u> <u>(tons/h)</u>	<u>Length,</u> <u>m (ft)</u>
NaNO ₃ unloading conveyor	55 (60)	55 (180)
KNO ₃ unloading conveyor	36 (40)	43 (140)
NaNO ₃ tunnel conveyor	7 (8)	43 (140)
KNO ₃ tunnel conveyor	5 (6)	30 (100)
NaNO ₃ transfer conveyor	6 (8)	30 (100)
KNO ₃ transfer conveyor	5 (6)	30 (100)
NaNO ₃ weigh - feeder	5 (6)	24 (80)
KNO ₃ weigh - feeder	4 (4)	24 (80)

Dowtherm Piping

Quantity	30 m (100 ft).
Material	A106 CS, with 25 mm (1 in) calcium silicate insulation
Size	25.4 mm (1 in) dia , Schedule 40

Molten Salt Piping

Quantity	152 m (500 ft)
Material	A106 CS, with 51 mm (2 in) calcium silicate insulation
Size	25.4 mm (1 in) dia, Schedule 40
Comments	Heat traced

Table 4-33

ENERGY STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

Storage media

Specific heat, J/kg-K (Btu/lb-°F)	1552.3 (0.371)
Viscosity, 10 ³ Pa-s (10 ³ lb/ft-s)	3.27 (2.2) at 561°F 0.78 (0.524) at 1,050°F
Density, kg/m ³ (lb/ft ³)	1,903.4 (118.7) at 561°F 1,722.2 (107.4) at 1,050°F

Storage media mass, kg (lb)	19,283,572 (42,513,000)
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Tank characteristics

	<u>Hot</u>	<u>Cold</u>
Number		
Dimensions x H, m (ft)	18.3 x 11 (60 x 36)	17.7 x 10.7 (58 x 35)
Volume, m ³ (ft ³)	2,883 (101,800)	2,620 (92,500)
Material	SA-240-316	SA-516-Gr. 70
Design temperature K (°F)	839 (1,050)	567 (561)
Tank mass, kg (lb)	129,365 (285,200)	93,349 (205,800)
Surface area/tank, m ² (ft ²):		
Top	288.9 (3,110)	269.9 (2,906)
Side	630.4 (6,786)	592.4 (6,377)
Bottom	262.6 (2,827)	245.4 (2,642)
Total	1,181.9 (12,723)	1,107.7 (11,925)
Insulation thickness, mm (in)	76.2 (3)	127.0 (5)
Insulation mass, kg (lb)	1,539 (3,393)	2,410 (5,313)

Pump characteristics for purification and drain loops only:

Material	316 S.S.	316 S.S.
Number	4	4
Type	Centrifugal, vertical, suction canned	Centrifugal, vertical, suction canned
Speed (single), rpm	1,760	1,760
Head, m (ft)	16.8 (55)	22.9 (75)
NPSH, m (ft)	3 (10)	3 (10)
Operating temperature, K (°F)	839 (1,050)	567 (561)
Power, kW _e (bhp)	12 (15)	16 (20)
Capacity, m ³ /s (gal/min)	0.03 (420)	0.02 (360)

Table 4-34

ENERGY STORAGE SUBSYSTEM OPERATING CHARACTERISTICS

Extractable capacity MWHR (10^9 Btu): 754 (6,495)

Charging rate, MW:

Maximum	378.4
Design	378.4
Minimum	0

Discharging rate, MW:

Maximum	276.7
Design	253.8
Minimum	0

Discharge rate duration, h:

Maximum	6.84
Design	7.50
Minimum	0

Storage media operating temperature, K (°F):

Cold	567 (561)
Hot	839 (1,050)

Operating pressures, MPa (psia):

	<u>Hot</u>	<u>Cold</u>
Fully charged - top	0.10 (14.7)	0.10 (14.7)
bottom	0.28 (40.8)	0.29 (42.7)
Fully discharged - top	0.10 (14.7)	0.10 (14.7)
bottom	0.11 (16.2)	0.11 (16.3)

Heat loss rate, MW:

Fully charged storage	1.01
Fully discharged storage	0.51

Storage ramp rate, percentage of maximum discharge power/minute	100%
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temperature of approximately 644 K (700°F). The conceptual design cold salt storage temperature of 567 K (561°F) is well below the maximum design temperature for carbon steel of 644 K (700°F). Carbon steel, therefore, has been selected for the cold salt storage tank material. Conventional calcium silicate insulation with a weatherproof aluminum jacket is installed on the outside of the tank to reduce the thermal losses.

The hot salt storage temperature of 839 K (1050°F) implies that a carbon steel tank cannot be used. If an internal tank insulation is used, however, the tank wall temperature can be reduced to 644 K (700°F) or below. The difficulty is in finding a proven insulating material, or combination of materials, which will survive 30 years of pressure and temperature cycling in the presence of 839 K (1050°F) salt. Three internal insulating schemes were considered. The first was based on the Martin Marietta concept, which used ceramic fiber insulation and Incoloy 800 shingles for erosion protection. The second involved castable refractory insulation, with a stainless steel liner for corrosion protection. The third was based on brick refractory insulation, with an acid-resistant brick-facing for corrosion protection.

In their final report, Martin Marietta stated that the ceramic fibers tended to separate in molten salt and that the long-term stability of this insulation required additional tests. For this reason, this method of internal insulation was not considered for the conceptual design.

The principal design problem with the second insulation method is the stainless steel liner. It would have to not only provide a hermetic seal between the salt and the refractory, but it would also have to accommodate its thermal expansion while it remains in contact with the refractory. The liner must remain in contact with the refractory because the thin shell is not designed to withstand the hydrostatic force of the salt. Hydrostatic forces are transferred through the liner and the refractory to the structural tank wall. Because of the potential design and fabrication problems with the liner, this method of internal insulation was rejected.

The acid-resistant brick of the third insulation option avoids the problems of liner fabrication and thermal expansion. The bricks, however, must be set in place with mortar. Since the mortar is anticipated to have a lifetime of less than 30 years, periodic maintenance of the mortar is required. This entails the shutdown of at least a portion of the plant and the draining of the tank. This problem, together with the difficulty in spanning the approximately 18.3 m (60 ft) diameter roof with bricks and mortar, argues strongly against the third method of internal insulation. Because of the problems associated with each of these tank designs, the internal insulation concept was dropped in favor of a conventional tank design using 316 stainless steel as the tank material. The hot salt tank is similar in design to the cold salt tank. Conventional calcium silicate with a weatherproof aluminum jacket is installed on the outside of the tank to reduce thermal losses. It was felt that the simplified design and reduced maintenance requirements of the stainless steel tank justified the additional capital cost of the alloy steel.

4.5.2.2 Insulation Thickness Optimization

The selection of the optimum insulation thickness for the hot and cold tanks involves a tradeoff between the cost of the insulation and the incremental cost of the collector and receiver subsystems required to supply the heat loss through the insulation.

Overall tank dimensions were based on preliminary thermal storage requirements for the hot and cold tanks. Capital costs for the insulation and aluminum jacket were developed for cold tanks having insulation thicknesses of 25 mm (1.0 inch) to 152 mm (6.0 inches) and for hot tanks having thicknesses of 51 mm (2.0 inches) to 254 mm (10.0 inches). Daily heat losses were calculated for each combination of salt tank and insulation thickness based on the following parameters:

- Cold tank insulation thermal conductivity $0.059 \text{ W/m}^\circ\text{C}$ ($0.034 \text{ Btu/h-ft}^\circ\text{F}$)
- Hot tank insulation thermal conductivity $0.078 \text{ W/m}^\circ\text{C}$ ($0.045 \text{ Btu/h-ft}^\circ\text{F}$)
- Molten salt to tank wall heat transfer coefficient $>56.8 \text{ W/m}^2^\circ\text{C}$ ($10 \text{ Btu/h-ft}^2^\circ\text{F}$)
- Salt vapor to tank roof heat transfer coefficient $9.1 \text{ W/m}^2^\circ\text{C}$ ($1.6 \text{ Btu/h-ft}^2^\circ\text{F}$)
- Jacket to air heat transfer coefficient $22.8 \text{ W/m}^2^\circ\text{C}$ ($4.0 \text{ Btu/h-ft}^2^\circ\text{F}$)
- Average ambient temperature 291 K (65°F)

The incremental capital cost of the collector and receiver subsystems required to supply the heat loss through the insulation is approximately $\$250/\text{kWh}$ or $\$0.0735/\text{Btu/h}$. For each tank, the minimum total of the capital cost of the insulation and the corresponding capital cost of the heat loss defined the optimum insulation thickness. The optimum insulation thicknesses were found to be 76 mm (3 inches) for the cold tank and 127 mm (5 inches) for the hot tank. The optimum thicknesses apply to both the tank roof and the tank wall. Assuming that two storage tanks are filled with cold salt, two with hot salt, and two are empty, the ESS instantaneous heat loss is approximately 0.96 MW_t . Although heat losses through the tank foundations were not included in this analysis, estimates show that the losses are one to two orders of magnitude less than the wall and roof losses.

A transient heat transfer analysis was performed to determine the time required for the salt in the storage tanks to reach a freezing temperature of 506 K (450°F). Based on the optimum insulation thicknesses, an initial hot tank temperature of 839 K (1050°F), and an initial cold tank temperature of 567 K (561°F), the salt in the hot tank will freeze in approximately 3800 hours (5.3 months) and the salt in the cold tank in approximately 1460 hours (2.0 months).

4.5.2.3 Storage Volume Requirements

The design storage volume was calculated from the thermal energy storage requirements for the summer solstice and an overall change in

temperature from the hot salt to the cold salt of 272 K (489°F). The storage requirements include the following:

- 7.5 hours of EPGS evening operation at the rated capacity of 100 MW_e (net). Based on a net cycle efficiency of 39.4%, the required storage volume is 9,400 m³ (332,600 ft³).
- 7 hours of HTS evening freeze protection. The 7 hours assumes that the plant operates directly from solar insolation for 14 hours on the summer solstice and that the smallest cold salt line will freeze in 3 hours. Based on heat loss calculations for the hot and cold salt lines at a temperature of 544 K (520°F), the required salt storage volume is 320 m³ (11,200 ft³).
- 0.5 hour of EPGS morning startup energy. Assuming the EPGS will operate at the 25% heat balance conditions for 0.5 hour with a net cycle heat rate of 10,500 kJ/kWh (10,000 Btu/kWh), the required storage volume is 180 m³ (6,400 ft³).
- 24 hours of ESS tank heat losses. Based on an average tank heat loss of 0.96 MW, the required storage volume is 260 m³ (9,100 ft³).
- 3 hours of auxiliary steam generator operation. The 3 hours assumes that the plant operates directly from solar insolation for 14 hours and from thermal storage for 7.5 hours on the summer solstice. Assuming that the steam generator is operating in a hot standby mode and that auxiliary steam is bled from the steam drum, the required salt storage volume is 85 m³ (3,000 ft³).

The total storage requirement, therefore, is 10,245 m³ (362,300 ft³) divided among four hot salt tanks.

4.5.2.4 Tank Designs

In the requirements definition document, the Barstow site is classified as a seismic zone 3. Seismic considerations, therefore, had a strong influence on the tank design. These considerations include the ratio of the tank height to the diameter and also the tank floor design.

A tank-height-to-diameter ratio of 0.6 was selected for the conceptual design for three reasons. First, the low ratio reduces the tank overturning moment due to seismic accelerations. Second, an earthquake would produce waves in the salt inventory of the tank. If these waves were to become large enough, they might buckle the tank. A

low tank height reduces the size of these waves. Third, tank overturning moments due to earthquakes increase the soil bearing stress in certain locations of the tank foundation. A low tank height reduces the maximum bearing stresses.

In normal tank designs, stresses in the floor material are considerably less than in the wall material. As a result, the floor thickness is less than the wall thickness. If a large overturning moment is applied to such a tank, one edge of the tank may lift from the foundation. In this situation, the stress at the floor-to-wall weld may rupture the thin floor. To minimize this occurrence, the floor thickness of the tanks in the conceptual design is doubled for the outer segment of the tank radius 0.61 m (2 ft).

Two additional design parameters, the location of the piping penetrations and the minimum tank freeboard, have an influence on the design dimensions of the salt tanks. The centerlines of the piping penetrations are located 0.61 m (2 ft) above the tank floor to reduce the wall thermal and pressure stresses near the penetration. Salt inlet lines continue past the wall penetration to a pipe that follows the perimeter of the tank floor. Mixing spargers have been included in the perimeter lines for two reasons. First, as outlined in Section 4.4.2.1.8, the spargers mix intermediate temperature salt from the field piping with the salt inventory of the cold tanks during morning startups and evening shutdowns. Second, the spargers ensure that any precipitates in the hot and cold tanks remain suspended in the salt. Once suspended in the salt, they may be removed by the salt maintenance system. The salt outlet lines have a similar perimeter sparger arrangement to collect as many suspended solids as possible for the salt maintenance system. The minimum allowable salt level above the sparger centerline is 0.46 m (1.5 ft). This ensures that salt is always available at the pump suction. A salt freeboard of 0.30 m (1 ft) at the top of the tank has been assumed for the volume of the cover gas. Therefore, the total tank height that is unavailable for thermal storage is 0.92 m (3 ft).

Based on the above discussions, conceptual designs were prepared for the hot and cold salt tanks. The design data are summarized in Figures 4-42 and 4-43. The cold tanks have slightly smaller dimensions than the hot tanks because of the greater density of the cold salt. A flat bottom was selected for the conceptual design rather than a shallow cone. The static friction between the tank and the foundation is sufficient to prevent the tank from sliding during an earthquake. The wall course thicknesses were based on the maximum hydrostatic forces and the allowable material stresses. The tank designs include a corrosion allowance of 3.2 mm (1/8 inch) on the carbon steel and 1.6 mm (1/16 inch) on the stainless steel. The hemispherical section roof includes radial stiffeners to accommodate the snow and ice loads. An access manhole is also included in the roof.

Of the three storage tanks in the northern or southern heliostat fields, only two are constructed of stainless steel. This is made possible by sequencing the tank charging and discharging such that hot salt is stored in only two tanks. An example of this sequencing is shown in Table 4-35.

Table 4-35

TANK CHARGING AND DISCHARGING SEQUENCE

Storage Mode	Storage Tank Number		
	1	2	3
Charged	Hot	Hot	Empty
Discharging	Empty	Hot	Cold
Discharged	Cold	Empty	Cold
Charging	Empty	Hot	Cold
Charged	Hot	Hot	Empty

FIGURE 4-42 COLD SALT STORAGE TANK

FIGURE 4-43 HOT SALT STORAGE TANK

4.5.2.5 Foundation Design

As outlined in the parametric analysis, the design soil for the Barstow location is UBC Class 3 (sandy gravel). This soil, when dry, is a suitable foundation for the cold and hot salt tanks.

The foundations for the conceptual design are constructed by excavating the tank area to a depth of 3.1 m (10 ft). At this depth, the soil has a bearing strength of 474 kPa (10,000 lb/ft²). The excavation volume is backfilled with compacted soil to achieve a comparable bearing strength, and the tanks are then placed directly on the compacted soil. Heat transfer through the bottom of the tank will dry the foundation soil. The soil will not shrink from the heat, thus reducing the possibility of tank settling during the first years of plant operation. With a tank filled with cold salt, the soil bearing strength of 474 kPa (10,000 lb/ft²) has been specified to minimize the possibility of vertical tank movement following an earthquake.

If the site soils contain moderate to large amounts of clay, they may not be suitable as the tank foundation. Alternative foundation designs for these soils are shown in Figure 4-44. The design shown in figure 4-44a is suitable for soils with a moderate amount of clay. The foam glass insulation and compacted sandy soil provide thermal insulation for the undisturbed clay. The sand acts as a buffer, preventing thermal expansion tank movements from abrading the fragile foam glass. The rock and gravel perimeter ring contains the sand and foam glass. The design shown in Figure 4-44b has been developed for soils with predominant amounts of clay. The expanded shale acts as thermal insulation for the undisturbed clay.

4.5.2.6 Salt Freeze Protection

Two methods of salt freeze protection for the ESS have been included in the conceptual design; i.e., electric trace heating of the tank floor and an oil-fired salt heater.

(a) FOR SOILS WITH MODERATE AMOUNT OF CLAY

(b) FOR SOILS WITH PREDOMINANT AMOUNTS OF CLAY

FIGURE 4-44 ALTERNATE FOUNDATION DESIGNS

Electric heat tracing is designed to preheat the tank floor during plant construction. This ensures that molten salt from the salt charging system does not freeze as it enters the tank. Once the salt level reaches the elevation of the piping spargers, an oil-fired salt heater maintains the temperature of the salt and the heat tracing is turned off.

An oil-fired salt heater is located in each ESS pump shelter. The cold tank heaters are designed to maintain a salt temperature of 533 K (500°F) during construction and 567 K (561°F) during an extended shutdown. The hot tank heaters maintain a temperature of 839 K (1050°F) during an extended shutdown. The heater design duties selected are twice as large as the estimated tank thermal losses. This permits reheating of the salt should it fall below 567 K (561°F) or 839 K (1050°F). The natural convection heater includes a fire tube at the bottom of the heater, an intermediate heat transfer salt, and heat exchanger tubes containing the salt from the storage tanks at the top of the heater. This design ensures that local, high-temperature areas of the fire tube do not cause thermal decomposition of the storage salt. When the heater is in operation, salt flows from the storage tanks to the drain sump, through the salt circulation pumps, through the heater, and returns to the storage tank. Fuel oil for the heaters is stored in the central fuel oil storage tank adjacent to the turbine building.

4.5.2.7 Ullage Maintenance Design

The cover gas system provides H₂O- and CO₂-free air to the storage tanks and minimizes tank ventilation heat losses. The use of H₂O- and CO₂-free air minimizes the formation of hydroxides and carbonates, respectively, in the storage salt. Minimizing the ventilation heat losses also reduces the size of the heliostat and receiver subsystems required to charge the storage subsystem. The cover gas system includes a silica gel water vapor desiccant, a carbon dioxide absorbent, and interconnecting tank vent lines.

The water vapor desiccant is a commercial silica gel unit that includes two 100% capacity desiccant trains and an electric heater for regenerating the silica gel. The carbon dioxide absorbent system, also a commercial unit, includes an absorber column, stripper column, heat exchanger, pump, and controls. The absorbent is a combination of monoethanol and diethanol amines. Heat for regeneration of the absorbent is provided by an electric heater. The salt tanks at each storage location are interconnected by a 0.102 m (4 inch) diameter vent line.

As a tank is filled with salt, air expelled from the tank flows through the vent to an adjacent, emptying tank. If air from a cold tank enters a hot tank, it expands as it is heated from 567 K (561°F) to 839 K (1050°F). A pressure relief valve, set to open at 0.69 kPag (0.1 psig), is located on the vent line to ensure that the tanks do not experience above-atmospheric pressure. If air from a hot tank enters a cold tank, it contracts as it cools from 839 K (1050°F) to 567 K (561°F). To ensure that the tanks do not operate at subatmospheric pressure, a vacuum relief valve, set to open at -0.1 psig, is also included in the vent line. The interconnecting vent system minimizes the amount of ambient air brought into the tanks during a charge-discharge cycle. This reduces the size of the H₂O and CO₂ removal systems and reduces the heat transferred from the salt in heating the ambient air to 567 K (561°F) or 839 K (1050°F).

4.5.2.8 Fluid Maintenance Design

For the draw salt used in this study, air is the preferred cover gas for the salt storage tanks and drain sumps. However, carbon dioxide and water vapor in the air react with the salt to form the following carbonates and hydroxides:

Decomposition

- Oxide formation

- Hydroxide formation

- Carbonate formation

The carbonates and hydroxides form precipitates in the cold salt which may foul heat transfer surfaces in the steam generator and the receiver tubes. To minimize the annual expense of salt blowdown to control carbonate and hydroxide concentrations, a salt maintenance system has been included in the conceptual design.

The maintenance system consists of a drain sump, an NO_2 bubble column, salt circulation pumps, and, for the cold tanks only, plate and frame precipitate filters in a small circulation loop adjacent to the salt tanks. During the evening, when the maintenance system is in use, salt flows from the storage tank to the drain sump. (This is the same sump used for the field piping and receiver drain system.) Salt flows from the sump through the NO_2 bubble column, which is a counter-flow reaction vessel designed to convert the carbonates and hydroxides to the original nitrates. The reactions in this conversion process are as follows:

- Oxide conversion

- Hydroxide conversion

- Carbonate conversion

- Recombination

The water vapor and carbon dioxide formed in the conversion reactions are vented from the atmospheric column.

The circulation pumps draw suction from the bottom of the column and discharge, in the case of the cold tank, to two 100% capacity plate and frame precipitate filters. The filters are designed to trap any carbonate or hydroxide precipitates that do not react in the NO_2 column. Two full capacity filters are provided to ensure continuous operation during filter cleaning or maintenance. Discharge from the filter, or in the case of the hot tanks, from the circulation pumps, flows to a salt heater and returns to the storage tank. The salt heater is the freeze protection heater discussed earlier.

4.5.2.8.1 Salt Charging System

During the final stages of plant construction, bulk quantities of the draw salt constituents, solid NaNO_3 and solid KNO_3 , are delivered to the site. The salt charging system converts these solids to molten draw salt and delivers it to the storage tanks. The reasons for installing a charging system are twofold. First, the system produces molten salt at a lower cost than delivered, premixed salt. Second, due to the NaNO_3 and KNO_3 inventories, the system can provide a makeup salt supply as required.

Individual charging systems are adjacent to each group of salt storage tanks. Each system is sized to supply the molten salt required for one-half of the plant within an 8-month period and includes the following equipment:

- 30-day concrete storage bunkers for solid NaNO_3 and KNO_3
- Tunnel and transfer conveyors for solid NaNO_3 and KNO_3
- 48 hour storage hoppers for solid NaNO_3 and KNO_3
- Weigher-feeders for solid NaNO_3 and KNO_3
- 8 hour mixed salt hopper
- Melt tank with an integral heat transfer coil
- Dowtherm A vaporizer, flash tank, condensate tank, and feed pump
- Molten salt transfer pump

Storage for 30 days is provided for the solid NaNO_3 and KNO_3 . The solids are transferred, as required, to the 48-hour storage hoppers by conveyors. Weigher-feeders, located under the storage hoppers, weigh and deliver NaNO_3 and KNO_3 in the correct proportion to the 8-hour mixed salt hopper. Mixed salt is then delivered to the liquid inventory in the melt tank. The integral heat exchanger within the melt tank provides the thermal energy to maintain the inventory temperature at 561 K (550°F) and to melt the solids entering the tank. Molten salt from the tank is delivered to the cold salt storage tanks by a small transfer pump.

The oil-fired Dowtherm vaporizer produces saturated vapor at approximately 172 kPa (25 psia) and 561 K (550°F). The vapor is delivered to a flash tank to separate any liquid that is carried over with the vapor. The vapor then flows to the heat transfer coil, where it condenses to form a saturated liquid at 561 K (550°F). The condensate flows to a condensate storage tank, through a feed pump, and returns to the vaporizer. This system, rather than a direct-fired salt heater, is used for two reasons. First, the salt has little opportunity to decompose from local high-temperature areas of the heater. Second, with a condensing heat exchanger, the melt tank inventory temperature can be closely controlled.

4.5.2.8.2 Salt Makeup System

Although a salt maintenance system has been included in the conceptual design, a small salt blowdown from the storage tanks or the steam generator may be required. Also, since salt spills and leaks will inevitably occur, a salt makeup supply is required. For the conceptual design, the salt charging system will run intermittently to provide the required makeup salt.

4.5.3 ESS Performance, Operation, and Control

This section addresses the performance, startup, normal operation, shutdown, and control of the salt storage tanks and auxiliary equipment.

4.5.3.1 ESS Performance

The performance of the ESS is affected by the thermal losses of the salt storage tanks. Based on the optimum tank insulation thicknesses outlined in Section 4.5.2, the cold and hot storage tank heat losses are approximately $0.185 \text{ MW}_{\text{th}}$ and $0.294 \text{ MW}_{\text{th}}$, respectively, per tank. Assuming that two cold and two hot storage tanks are filled, the ESS heat loss is approximately 9.8% of the noon field piping heat loss and 0.15% of the noon receiver subsystem output. Assuming that the turbine-generator operates during the evening from storage at the maximum guaranteed load condition, the ESS heat loss is approximately 0.38% of the thermal energy delivered to the EPGS. Therefore, the thermal losses from the ESS have only a small impact on the performance of the plant.

Also, as outlined in Section 4.5.2, if a filled storage tank is left unheated, the salt in the cold and hot tanks will freeze in approximately 2.0 and 5.3 months, respectively. The possibility of the salt in a storage tank freezing before corrective action could be taken is remote.

4.5.3.2 ESS Operation

This section outlines the charging and discharging of the salt storage tanks for the summer and winter solstice days. The operation of the storage tank freeze protection, ullage control, salt maintenance, salt charging, and salt makeup systems are discussed in Section 4.5.2.

4.5.3.2.1 Summer Solstice

The charging and discharging of the hot salt storage tanks on the summer solstice is illustrated in Figure 4-45. The curves show the percent of tank capacity occupied by the hot salt as a function of the time of day. For brevity, only the sequencing for the hot tanks is shown. The sequencing of the cold tanks is, of course, the opposite of that for the hot tanks. The solar insolation curve, as presented in the requirement definition document, is included for reference.

The tank sequence shown in the figure assumes that the turbine-generator was operating at the maximum guaranteed load condition the previous evening. In the predawn hours of the following morning, the hot tank is 10% filled and the hot/cold tank is 0% filled (100% filled with cold salt).

At approximately 3:00 a.m., hot salt is transferred from the storage tank to the EPGS to start the steam generator and to bring the turbine-generator to the 3- to 5-MW_e load condition. The steam generator and turbine-generator startup procedure, which requires approximately 1.5 hours, is represented by the dip in the thermal capacity of the hot tank. By 4:30 a.m., the steam generator and main steam lines are at operating temperature and the load is increased on the turbine-generator. The RS supplies all of the thermal energy during this period. By 6:00 a.m., the turbine-generator has increased to the maximum guaranteed load. The RS is supplying thermal energy in excess of that required to drive the turbine-generator. The additional energy is directed to the hot tank and appears as an increase in capacity between 6:00 a.m. and 11:00 a.m. The salt to fill the hot tank is obtained by draining and heating the cold salt from the hot/cold tank.

FIGURE 4-45 HOT SALT TANK CHARGE AND DISCHARGE SEQUENCE

By 11:00 a.m., the hot tank is full and the hot/cold tank is empty. The RS continues to deliver thermal energy in excess of that required by the turbine-generator. Therefore, cold salt from the cold storage tank is heated in the RS and is transferred to the hot/cold tank. By approximately 6:00 p.m., the hot/cold tank is also filled with hot salt.

The solar insolation levels decline rapidly after 6:00 p.m. To maintain the maximum guaranteed load condition for the turbine-generator, hot salt is transferred from the hot/cold tank to the EPGS. This appears in the figure as a decrease in the capacity of the hot/cold tank. Cold salt from the EPGS is directed to the cold storage tank. The transfer of hot salt continues for approximately 4 hours until the hot/cold tank is empty.

At this time, salt is transferred from the hot storage tank to maintain the turbine-generator load. This appears in the figure as a decrease in the capacity of the hot tank. Cold salt from the EPGS is directed to the hot/cold tank. The transfer of hot salt continues for approximately 3 hours with the turbine-generator at the maximum guaranteed load. At this time, the programmed shutdown of the turbine-generator starts. Over a period of approximately 1 hour, the load on the turbine-generator is reduced from 100 MW_e to 3 to 5 MW_e, after which the turbine-generator is immediately tripped. The shutdown procedure appears in the figure as a taper in the capacity-time curve for the hot tank. The capacity of the hot tank at the end of the programmed shutdown is sufficient to prevent freezing of the HTS during the evening and to restart the steam generator and turbine-generator the following morning.

4.5.3.2.2 Winter Solstice

As shown in Figure 4-45, the charge and discharge sequence for the hot tanks on the winter and summer solstices is qualitatively identical. However, the two procedures differ in three quantitative areas.

The first difference involves the sequence periods. During the winter solstice the sun rises later and sets earlier. Therefore, thermal

energy is delivered to the storage tanks later in the morning and is transferred from the tanks earlier in the afternoon and evening.

The second difference deals with the quantity of energy collected during the winter solstice. During the winter, the quantity is considerably less than during the summer. This difference is reflected in the areas under the capacity-time curves for the hot and hot/cold tanks. During the winter solstice, the hot tank may fill to 100% capacity while the hot/cold tank fills to only 30% capacity. Under these conditions, 70% of the capacity of the cold storage tank is not heated to 1050°F. This, in turn, reduces the number of hours the turbine-generator can operate from storage during the evening.

The third difference involves the amount of thermal energy required in the hot tank to provide freeze protection for the HTS during the evening. During the winter, ambient temperatures are lower and the period of direct solar operation is shorter. Consequently, more thermal energy must be available in the hot tanks following a turbine shutdown on the winter solstice. This is reflected in a higher percentage of required hot tank capacity at the end of a winter day.

4.5.3.3 ESS Control

Instrumentation for the salt storage tanks includes level sensors and thermocouples along the tank wall. Signals from this instrumentation are fed to the master control subsystem. The MCS, in turn, controls the charging and discharging of the tanks according to the turbine-generator load, steam drum pressure, and available insulation. The charging and discharging are controlled through tank valve positions and the operation of the primary salt pumps.

4.6 Electrical Power Generating Subsystem (EPGS)

4.6.1 Subsystem Requirements

The electrical power generation subsystem (EPGS) converts the steam flows from the steam generator to electric power. In the conceptual design, the EPGS includes the following equipment and systems:

- Turbine-generator
- Feedwater heaters
- Condenser
- Cooling towers
- Boiler feed, condensate, and circulating water pumps
- Auxiliary mechanical and electrical systems
- Turbine building

4.6.2 EPGS Design

The turbine equipment and building layout are illustrated in Figure 4-46. The EPGS equipment list is summarized in Table 4-36. Data sheets for the EPGS are presented in Tables 4-37 through 4-39.

4.6.2.1 Turbine Design

The turbine is a 3600-rpm tandem-compound unit rated at 107.2 MW_e gross electrical output. At maximum guaranteed load, the throttle steam conditions are 16.5 MPag/811 K (2400 psig/1000°F) and the reheat steam conditions are 2.9 MPag/811 K (421 psig/1000°F). The low-pressure turbine exhausts to a 10.1 kPa (3-inch HgA) condenser back-pressure. The high pressure (HP), intermediate pressure (IP), and low pressure (LP) sections supply extraction steam for seven stages of feedwater heating as well as steam for the plant auxiliaries. The turbine has electrohydraulic controls and operates with constant pressure over the load range. It is assumed to be a GE two-casing machine with a combined high-pressure and intermediate-pressure casing. The LP section is a two-flow casing with 419 mm (16.5 inch) last stage blades.

FIGURE 4-46 TURBINE GENERATOR AND STEAM GENERATOR BUILDING

Table 4-36

EPGS EQUIPMENT LIST

Mechanical Equipment

Main Turbine:

Quantity	1
Type	Tandem compound two-flow with 419 mm (16.5 in) last stage blades Single reheat--two casing
Main steam conditions	16.5 MPag, 811 K (2,400 psig, 1,000°F)
Reheat steam temperature	811 K (1,000°F)
Condenser pressure	10.1 kPa (3 in HgA)
Design speed	3,600 rpm
Maximum guaranteed load gross unit output	107,205 KW _e
HP turbine efficiency	82.90%
Reheat turbine efficiency	91.39%
Includes	Lube oil system; all turbine inlet valves and turbine-generator control system package; steam seal equipment with regulating system, steam packing exhaustor, valves and piping; turbine exhaust spray system, motor driven turning gear; insulation and lagging

Generator:

Quantity	1
Type	Hydrogen-cooled
Maximum guaranteed load output	123,875 kVA max
RPM	3,600
Power factor	0.9
H ₂ pressure	414 kPag (60 psig)
Frequency	60 Hz
Phase	3 phase
Includes	Voltage and excitation controls; hydrogen cooling system; seal oil system; stator conductor cooling system; all pertinent coolers

Feedwater Heater No. 1:

Type	Horizontal, U-tube, 2-pass
Duty	6.24 MW _{th} (21.31 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D.; 13 BWG
Tube material	304 L
Gross surface	434 m ² (4,673 ft ²)

Feedwater Heater No. 2:

Type	Horizontal, U-tube, 2-pass
Duty	12.24 MW _{th} (41.78 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D. 13 BWG
Tube material	304 L
Gross surface	309 m ² (3,323 ft ²)

Table 4-36 (continued)

Feedwater Heater No. 3:

Type	Horizontal, U-tube, 2-pass
Duty	14.17 MW _{th} (48.38 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D., 13 BWG
Tube material	304 L
Gross surface	388 m ² (4,175 ft ²)

Feedwater Heater No. 5:

Type	Horizontal, straight tube
Duty	11.53 MW _{th} (39.35 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D. 22 BWG
Tube material	304
Gross surface	262 m ² (2,822 ft ²)

Feedwater Heater No. 6:

Type	Horizontal, straight tube
Duty	6.85 MW _{th} (23.39 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D. 22 BWG
Tube material	304
Gross surface	214 m ² (2,300 ft ²)

Feedwater Heater No. 7:

Type	Horizontal, straight tube
Duty	5.86 MW _{th} (19.99 x 10 ⁶ Btu/h)
Tube size	15.9 mm (5/8 in) O.D., 18 BWG
Tube material	304
Gross surface	272 m ² (2,923 ft ²)

Deaerating Heater:

Type	Direct contact tray, vertical
Material	285 Gr C
Outlet capacity	353,100 kg/h (776,760 lb/h) max.
Operating pressure	414 kPa (60 psia)
Temperatures	Extraction steam, 569 K (565°F)
	Saturation steam 418 K (292°F)
	Condensate inlet, 400 K (260°F)
	Drain inlet, 427 K (308°F)

Deaerator Storage Tank:

Capacity	28.8 m ³ (7,600 gal)
Diameter	2.74 m (9 ft - 0 in)
Overall length	5.97 m (19 ft - 7 in)
Shell thickness	17.5 mm (11/16 in)
Head	20.6 mm (13/16 in)
Type heads	Flanged and dished
Design press	517 kPag (75 psig)

Table 4-36 (continued)

Steam Generator Feedwater Pump:

Quantity	2
Type	Multistage, horizontal, split
Capacity	0.057 m ³ /s at 19.0 MPa head (905 gal/min at 2,760 psi head)
Motor hp	1300 kW (1,750 bhp)
RPM	3,500
NPSH	14.3 m (47 ft)

Condensate Pumps:

Quantity	3
Type	Multistage, vertical can
Capacity	0.044 m ³ /s at 3.68 MPa head (700 GPM at 2,760 psi head)
Motor hp	56 kW (75 hp)
NPSH	4.9 m (16 ft)

Condenser (less tubes):

Quantity	1
Type	Single pressure and shell, divided waterbox
Duty	145.3 MW _{th} (495.8 x 10 ⁶ Btu/h)
Turbine exhaust flow	230,500 kg/h (507,045 lb/h)
Cooling water flow	2.65 m ³ /s (42,000 gal/min)
Inlet/outlet CW temperature	302/316 K (84°F/108°F)
Supply includes	Circulating water line expansion joints; main turbine exhaust duct and expansion joints; main turbine exhaust transition pieces with heater mounting flanges

Condenser Tubes:

Quantity	7,981
Number of passes	2
Length/pass	8.23 m (27 ft) effective
Tube O.D./BWG	22.2 mm (7/8 in)/18
Material	Admiralty
Surface	4,600 m ² (49,500 ft ²)
Cleanliness	85%
Design velocity	2.13 m/s (7 ft/s)
Heat transfer coefficient	3,520 W/m ² -°C (620 Btu/h-ft ² -°F)
Impingement and air removal tubes	22.2 mm (7/8 in), 20 BWG, 304 SS

Steam Jet Air Ejector Package:

Number of stages	2
Includes	Intermediate and after condensers
Capacity	0.0094-0.014 m ³ /s (20-30 SCFM)

Hogging Ejector:

Number of stages	1
Capacity	0.165 m ³ /s (350 SCFM)

Table 4-36 (continued)

Circulating Water Treatment Package:

Quantity	1
Includes	Storage tanks, pumps and injectors for sodium chlorides and carbonates

Cooling Tower:

Quantity	1
Type	Mechanical draft
Flow rate	2.65 m ³ /s (42,000 gal/min)
Number of cells and fans	4 (rectangular)
Cell dimensions	15.5 m x 15.5 m (51 ft x 51 ft) (rectangular)
Fan drivers	Induction motors, 4 x 100 kW (135 hp)
Approach	5.6 K (10°F)
Range	13.3 K (24°F)

Instrument and Service Compressed Air Package:

Quantity	1
Includes	2-0.353 m ³ /s/862 kPag (2 to 750 SCFM/125 psig) compressors, 3 after-coolers, 1 air receiver, 3 moisture separators, 3 intercoolers, 6 oil coolers/filters, 3 inlet filter-silencers, 2 dryers, 2 prefilters/after filters, 2 instrument air dryer coolers

Fire Water Pump and Diesel Drive:

Quantity	1
Type	Vertical, centrifugal
Capacity	0.126 m ³ /s at 107 m head (2,000 gal/min at 350 ft head)
RPM	1,700
Efficiency	77%
Driver	Diesel engine, 172 kW (230 bhp)

Fire Water Pump and Motor:

Quantity	1
Type	Vertical, centrifugal
Capacity	0.126 m ³ /s at 107 m head (2,000 gal/min at 350 ft head)
RPM	1,700
Efficiency	82%
Driver	Induction motor, 182 kW (245 bhp)

Fire Water Jockey Pump and Motor:

Quantity	1
Type	Vertical, centrifugal
Capacity	0.0032 m ³ /s at 49 m head (50 gal/min at 160 ft head)
RPM	1,700
Efficiency	40%
Driver	Induction motor, 4 kW (5 bhp)

Table 4-36 (continued)

Demineralized Water Storage Tank/Condensate Storage Tank:

Quantity	1 each
Capacity	190 m ³ (50,000 gal)
Dimensions	6.7 m (22 ft) dia x 5.5 m (18 ft) high
Material	Stainless steel

Circulating Water Pumps:

Quantity	2
Capacity	1.33 m ³ /s at 14.3 m head (21,000 gal/min at 47 ft head)
Type	Vertical, wet pit, multistage
Motor	225 kW (300 hp)

Condensate Polisher Package:

Quantity	3 to 50% capacity trains
Capacity	0.082 m ³ /s (1,300 gal/min)

Demineralized Water Makeup Storage Transfer Package:

Quantity	2 trains
Capacity	0.0035 m ³ /s (55 gal/min)

Lube Oil Storage Transfer and Purification:

Includes	2 lube oil storage tanks and pumps 1 lube oil centrifuge
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Startup Steam Generator Feedwater Pump:

Quantity	1
Type	Multistage, horizontal
Capacity	0.025 m ³ /s at 24.6 MPag (400 gal/min at 3,565 psi head)
Motor	750 kW (1,000 hp)

Emergency Diesel Generator:

Quantity	1
Includes	Starter motor and battery rack
Output	2,000 kW

No. 2 Oil Storage Tank:

Quantity	1
Type	Above ground
Dimensions	6.1 m (20 ft) dia x 5.5 m (18 ft) high
Capacity	159 m ³ (1,000 barrels)
Plate material	Carbon steel
Plate thickness	6.35 mm (1/4 in) to 12.7 mm (1/2 in)

Makeup Water Pump:

Quantity	2
Type	Deepwell, centrifugal
Capacity	0.038 m ³ /s at 50 m head (600 gal/min at 165 ft)
Motor	37 kW (50 bhp)

Table 4-36 (concluded)

Auxiliary Closed Cooling Water Heat Exchangers:

Quantity	2
Effective surface	930 m ² (10,000 ft ²)
Material	Stainless steel
Duty	3.59 MW _{th} (12.25 x 10 ⁶ Btu/h)
Type	Horizontal, U-tube

Auxiliary Closed Cooling Water Pumps:

Quantity	2
Type	Horizontal, centrifugal
Capacity	0.158 m ³ /s at 17 m head (2,500 gal/min at 55 ft head)
Motor hp	19 kW (25 hp)

Service Water Pumps:

Quantity	2
Capacity	0.126 m ³ /s at 76 m head (2,000 gal/min at 250 ft head)
Type	Vertical, wet pit turbine
Motor	150 kW (200 bhp)

Critical Piping

Main Steam Piping

ASTM A335 P-11
 318 mm (12-1/2 in) O.D.
 65.06 mm (2.561 in) wall thickness
 Length, 61 m (200 ft)
 102 mm (4 in) insulation

Hot Reheat

ASTM A335 P-11
 521 mm (20-1/2 in) O.D.
 20.4 mm (0.803 in) wall thickness
 Length, 61 m (200 ft)
 102 mm (4 in) insulation

Cold Reheat

ASTM 106 Gr.B
 610 mm (24 in) O.D.
 Schedule 30
 Length, 6.1 m (20 ft)
 64 mm (2-1/2 in) insulation

Feedwater

ASTM 106 Gr.C
 168 mm (6.625 in) O.D.
 Schedule 120
 Length, 76 m (250 ft)
 51 mm (2 in) insulation

Circulating Water

Concrete
 1.37 m (54 in) I.D.
 Length, 1,460 m (4,800 ft)

Table 4-37

EPGS DESIGN CHARACTERISTICS

Turbine-generator type and nameplate rating, MW: Tandem compound, two-flow with 16.5 in L.S.B., single reheat, two casing, 3,600 rpm and rated at 107.2 MW gross unit output at maximum guaranteed load.

Hydrogen-cooled generator, 3,600 rpm, 0.9 power factor, 60 Hz, 3-phase with output of 123,875 kVA max at maximum guaranteed load.

Turbine inlet conditions:

Temperature, K (°F)	811 (1,000)
Pressure, MPag (psig)	16.5 (2,400)
Flow rate, 10 ³ kg/h (10 ³ lb/h):	
Maximum	350.4 (772.4)
Design	316.8 (698.4)
Minimum	79.2 (174.6)

Turbine reheat conditions:

Temperature, K (°F)	811 (1,000)
Pressure, MPag (psig)	2.6 (377)
Flow rate, 10 ³ kg/h (10 ³ lb/h):	
Maximum	314.5 (693.3)
Design	284.8 (627.8)
Minimum	68.3 (150.6)

Number and location of feedwater heating extractions: Seven stages--
1 HP, 2 IPs and 4 LPs

Condenser type and configuration: Single pressure/shell with divided water box, 2 pass

Water flow rate, 10 ⁶ kg/h (10 ⁶ lb/h)	9.5 (21)
Water temperature in, K (°F)	302 (84)
Water temperature out, K (°F)	315 (108)
TTD, K (°F)	17 (31)

Feedwater pump stages: 14

<u>Auxiliary steam supply,</u>	9.1
10 ³ kg/h (10 ³ lb/h)	(20)
at rated pressure MPag (psig)	1.72 (250)

Sealing steam requirements at 25% load, kg/h (lb/h): 1,395 (3,075)

EPGS performance, operation, and control: refer to Section 4.6.4.3.

Table 4-38
CHARACTERISTICS FOR EPGS PUMPS

Pump	Steam Generator Feedwater	Condensate	Circulating Water	Service Water	Auxiliary Closed Cooling Water	Makeup Water	Startup Feedwater	Diesel Drive	Firewater	
									Main	Jockey
Quantity	2	3	2	2	2	2	1	1	1	1
Type	Centrifugal Horizontal Multistage Split Case	Centrifugal Vertical Canned Suction	Centrifugal Vertical Wet Pit	Centrifugal Vertical Wet Pit	Centrifugal Horizontal	Centrifugal Vertical Wet Pit (Deepwell)	Centrifugal Horizontal Multistage	Centrifugal Vertical Wet Pit	Centrifugal Vertical Wet Pit	Centrifugal Vertical Wet Pit
Speed, rpm	3,600	1,760	590	1,770	1,750	1,760	3,600	1,760	1,760	1,760
Head, m (ft)	2,207 (7,242)	156.7 (514)	14.3 (47)	76.2 (250)	16.8 (55)	50.3 (165)	2,642 (8,669)	107 (350)	107 (350)	49 (160)
NPSH, m (ft)	10.7 (35)	3 (10)	6.1 (20)	6.1 (20)	6.1 (20)	3 (10)	4.6 (15)	6.1 (20)	6.1 (20)	6.1 (20)
Operating temperature, K (°F)	421 (298)	319 (115)	302 (84)	302 (84)	304 (88)	289 (60)	387 (237)	302 (84)	302 (84)	302 (84)
Power, KW _e (bhp) at MGL	1,128 (1,390)	77 (95)	243 (300)	126 (155)	37 (45)	27 (33.3)	784 (967)	187 (230)	187 (230)	4 (5)
Capacity, m ³ /s (gal/min)	0.04 (703)	0.04 (588)	1.32 (21,000)	0.13 (2,000)	0.16 (2,500)	0.04 (600)	0.02 (372)	0.13 (2,000)	0.13 (2,000)	0.003 (50)

Table 4-39

EPGS CYCLE PARAMETERS

Load	VWO + 5% O.P.	100% (MGL)	100% (MGL)	25%
Time of day	Noon	Noon	Evening	Morning
Gross turbine cycle efficiency, %	42.4	42.3	42.3	36.5
Net turbine cycle efficiency, %	39.6	39.4	41.4	31.9
Gross cycle heat rate, kJ/kW-h (Btu/kW-h)	8,493 (8,050)	8,517 (8,073)	8,517 (8,073)	9,868 (9,354)
Net cycle heat rate, kJ/kW-h (Btu/kW-h)	9,089 (8,615)	9,131 (8,655)	8,705 (8,251)	11,278 (10,690)
Turbine exhaust pressure, mm (in) Hg	76.2 (3)	76.2 (3)	76.2 (3)	76.2 (3)
Feedwater temperature, K (°F)	511 (460)	505 (450)	505 (450)	439 (331)
Net unit output, kW _e	109,583	100,000	104,887	22,938
Auxiliary loads, kW _e	7,692	7,205	2,318	3,275
Gross unit output, kW _e	117,275	107,205	107,205	26,213

The turbine is provided with the necessary instrumentation connections for an ASME performance test.

4.6.2.2 EPGS Cycle Equipment Design

4.6.2.2.1 Feedwater Heaters

Feedwater is heated in seven stages, with one direct-contact deaerating heater and six stages of closed feedwater heating. Three stages of low pressure (LP) heating are located between the condenser and the deaerator. The three LP feedwater heaters are installed in the condenser steam dome. The first stage consists of a single shell provided with an individual bypass. Two single shell horizontal intermediate pressure (IP) heaters and a single shell high-pressure heater, complete with individual bypass, are located in the feedwater cycle between the steam generator feedwater pumps and the steam generator.

4.6.2.2.2 Steam Generator Feedwater Pumps

Two half-capacity steam generator feedwater pumps take suction from the deaerator storage tank and discharge through two intermediate feedwater heaters and one high-pressure feedwater heater. The feedwater pumps are horizontal, multistage, barrel-type, and motor driven. Since the unit is cycling because of daily startup and shutdown, a motor-driven horizontal multistage pump sized at approximately one-quarter the unit rated flow is included to provide feedwater during startup. This eliminates one of the half-capacity pumps from operating, thereby minimizing pump energy loss due to throttling.

4.6.2.2.3 Condensate Pumps

Three one-third capacity vertical multistage canned-type condensate pumps pump condensate from the condenser hotwell through the condensate polishers, the gland steam condensers, air ejector inter- and after-condensers, and the LP feedwater heaters to the deaerating heater. If one of the pumps fails, the other two run out to meet desired system requirements. The condenser hotwell level is controlled by reject and

makeup control valves in piping interconnected with the demineralized water storage tank. Minimum flow for the condensate pump and air ejector inter- and aftercondensers is maintained by recirculation control valves.

4.6.2.2.4 Condenser and Auxiliaries

The condenser is the single-shell pressure-type with divided waterboxes, admiralty tubes and deaerating hot well. The condenser handles the maximum condensing duty occurring during maximum continuous operation.

Air removal equipment consists of a full capacity steam hogging ejector for establishing a vacuum, and a two-stage, twin-element steam air ejector, complete with inter- and aftercondensers, for holding service.

4.6.2.2.5 Condensate Polishers

An inline mixed bed condensate polisher removes dissolved and suspended solids from the condensate and maintains feedwater chemistry within allowable limits in case of small condenser tube leaks. Three polisher vessels, each sized for 50% maximum continuous condensate flow, are provided, together with four resin charges (two in service, one on standby, one in regeneration). A complete system for automatic regeneration of exhausted resins is also provided.

4.6.2.2.6 Extraction Steam and Heater Drains

The HP feedwater heater receives extraction steam from the cold reheat lines while the IP feedwater heaters receive their extraction steam from the IP turbine. Normally, the drains from the HP and IP heaters cascade to the deaerator. Each heater is also provided with an individual dump line to the condenser.

The deaerator and LP feedwater heaters take extraction steam from the LP turbine. Three LP feedwater heaters are installed in the

condenser steam dome to eliminate extraction steam pipe runs and to conserve space. During 25% load operation, cold reheat steam is supplied to the deaerator to peg the pressure above atmospheric; only six feedwater heaters are active. The lowest LP feedwater heater is out of service.

Bleeder trip valves and block valves are provided in the extraction lines to prevent turbine overspeed and to protect the turbine from damage due to water induction per the requirements of the turbine manufacturer.

4.6.2.2.7 Auxiliary Steam

Auxiliary steam is supplied to the building heating system, to peg the deaerator, to seal turbine glands, and to hog the air ejector. During normal plant operation, this steam comes from the steam cycle. When this source is unavailable, steam to the process systems only is supplied from an auxiliary oil-fired boiler.

4.6.2.2.8 Emergency Power

On the plant site, a diesel generator provides power primarily for emergency shutdown of the unit during plant blackout. Power for startup after a blackout is provided from the grid to which the plant is connected.

4.6.2.2.9 Fuel Oil System

No. 2 fuel oil is assumed for the ESS tank oil heaters, auxiliary boiler, the fire pump diesel driver, and the emergency generator diesel. One rotary pump is provided to unload oil from tank trucks for storage in a single aboveground outdoor storage tank with a total capacity of 159 m³ (42,000 gal). Three half-capacity rotary pumps that take suction from the tank deliver oil to various services.

4.6.2.2.10 Lube Oil System

Lube oil is stored in two tanks, one for clean oil and one for dirty. Each tank has sufficient capacity to provide a complete oil change for the turbine generator. Two rotary pumps are provided to unload and transfer oil between tanks and the turbine reservoirs. A centrifuge purifies dirty oil stored in the storage tank and in the main turbine lube oil reservoir.

4.6.2.2.11 Compressed Air System

Instrument and service air are provided by two full-capacity centrifugal compressors, each with a moisture separator and aftercooler. The compressors discharge to a common receiver, from which separate headers supply the service and instrument air systems. The instrument air system includes two automatic desiccant-type air dryers operating on an 8-hour cycle.

4.6.2.2.12 Fire Protection System

The fire protection system generally meets the rules and regulations of the National Fire Protection Association. All hoses, nozzles, and cabinets are labeled or listed by the Underwriters Laboratories, Inc. A plant fire loop and a building fire loop are provided. Firefighting water is taken from the cooling tower basin by a motor-driven vertical turbine pump. A smaller vertical motor-driven jockey pump replenishes losses from leaks and maintains pressure when the system is not used. The motor-driven fire pump is backed up by a diesel-driven horizontal centrifugal pump.

Automatic fixed spray systems, hose connections and hydrants are installed in areas subject to fires. Portable extinguishers, including pressurized water, carbon dioxide and dry chemical, are also provided.

4.6.2.3 Cooling Water System Design

4.6.2.3.1 Raw Water

Raw water is assumed to be available from a deep well in the plant site. Two half-capacity vertical wet pit turbine pumps are provided to pump water through a single pipe to the cooling tower basin for makeup and to the plant makeup water treatment system. Either of the pumps can provide raw water during shutdown and during startup.

4.6.2.3.2 Circulating Water

Two half-size vertical, wet pit, mixed flow type pumps provide circulating water to the condenser. The pumps have water-lubricated bearings and removable elements. The circulating water pipes are concrete. Motor-operated butterfly valves are located in the circulating water piping at each pump discharge and at the condenser inlets and outlets. Provisions are made for future installation of a condenser tube cleaning system.

4.6.2.3.3 Service Water

Two full-size vertical, wet pit, turbine-type pumps provide service water to plant auxiliary components and to the auxiliary cooling water heat exchangers. A duplex basket strainer removes debris from the service water.

4.6.2.3.4 Cooling Tower

Hot water from the circulating water system and service water system discharges to a mechanical draft cooling tower where heat is ejected to the atmosphere. Provision is made for bypassing the tower at low temperatures and for freeze protection on shutdowns.

Equipment is provided to add sulfuric acid for pH adjustment, thus preventing calcium carbonate scaling. Chlorine solution is added for algae control. The chlorine content in tower blowdown discharged from the plant is kept within federal and state allowable limits.

Cold water from the cooling tower flows to a pump well containing the circulating water, service water pumps, and fire water pumps. Two fixed screens in series remove debris from the water before it reaches the pump suction.

4.6.2.3.5 Auxiliary Closed Cooling Water

An auxiliary closed loop cooling water system serves as an intermediate system to cool the hydrogen coolers, generator stator coolers, exciter coolers, turbine lube oil coolers, bearing lube oil coolers, main chillers, and other plant systems. Two full-capacity horizontal centrifugal pumps circulate buffered condensate in the loop. Heat is dissipated to one of two full-capacity shell-and-tube heat exchangers. A head tank accommodates thermal expansion, contraction of the coolant, and any surging in the system.

4.6.2.4 Electric Plant Equipment Design

The electric distribution system for the EPGS is shown in Figures 4-47 and 4-48.

4.6.2.4.1 General

The steam turbine-generator is rated at 107.2 MW, 0.9 PF, 13.8 kV, 3 phase, 60 Hz. A main power transformer steps up the voltage to tie in with the utility grid system. A main auxiliary transformer supplies indoor metal-clad switchgear rated 250 MVA, 4160 V, 3000 A to power the plant auxiliary equipment. Plant startup power is provided from a reserve auxiliary transformer. A double-ended 3000-kVA load center substation provides the necessary power for plant utilization at 480 V.

4.6.2.4.2 Emergency Diesel Generator

A diesel generator set rated at 2000 kW provides standby power for defocusing the heliostats under emergency conditions or during loss of normal power. Multiplexed control prevents operating all drives

1) DIESEL GENERATOR RATED AT 2000 KW HAS BEEN SELECTED AND BASED UPON ASSUMED USE OF MULTIPLEXING OF CONTROL UNDER EMERGENCY CONDITIONS .

FIGURE 4-48 480-VOLT POWER DISTRIBUTION,
SINGLE LINE

instantaneously, thus assuring that the capacity of the diesel generator set is adequate.

4.6.3 EPGS Performance, Operation, and Control

This section briefly discusses performance, startup, normal load operation, shutdown, and control of the EPGS.

4.6.3.1 EPGS Performance

The performance of the EPGS addresses the gross and net electric output of the plant at the minimum, design, and maximum operating conditions of the turbine-generator.

4.6.3.1.1 Gross Electric Power Output

Computer heat balances were prepared for the 25% of maximum guaranteed throttle flow load, maximum guaranteed load (MGL), and valves wide open, 5% overpressure (VWO, 5% OP) load. These heat balances, shown in Figures 4-49 through 4-51, correspond to the minimum, design, and maximum EPGS load conditions, respectively. All heat balances were based on a constant main steam pressure, "valve best point" operation, and a constant condenser backpressure. The gross turbine outputs for the three load conditions are summarized in Table 4-40.

The variation in final feedwater temperature and boiler temperature with turbine-generator load is illustrated in Figure 4-52.

4.6.3.1.2 Auxiliary Loads

The principal plant auxiliary loads include the electric motors of the EPGS and the primary salt pumps of the HTS. The auxiliary loads of the EPGS depend primarily on the turbine-generator load while the auxiliary loads of the HTS depend primarily on the insolation levels and time of day. The electric demand of the primary salt pumps is at a minimum in the morning and afternoon and at a maximum at noon.

LEGEND

ENGLISH	SI
W = LB/HR	= .4535 kg/HR
P = PSIA	= 6.895 kPa
F = °F	= (F-32) / 1.8 °C
H = BTU/LB	= 6.46 x 10 ⁴ KWHR/kg
HEAT RATE = BTU / KWHR	= 1.055 kJ / BTU
TURBINE TYPE: G.E. TC2F - 16.5" L.S.B. 3600 RPM	
STEAM GENERATOR DUTY	= 245.20 x 10 ⁶ BTU/HR
GROSS UNIT OUTPUT	= 26,213 KWe
GROSS TURBINE CYCLE HEAT RATE	= $\frac{245.20 \times 10^6 \text{ BTU/HR}}{26,213 \text{ KWe}} = 9,354 \frac{\text{BTU}}{\text{KWHR}}$
GROSS TURBINE CYCLE EFFICIENCY	= $\frac{3413}{9,354} \times 100 = 36.5 \%$
NET UNIT OUTPUT	= 22,938 KWe
NET CYCLE HEAT RATE	= $\frac{245.20 \times 10^6 \text{ BTU/HR}}{22,938 \text{ KWe}} = 10,690 \frac{\text{BTU}}{\text{KWHR}}$
NET CYCLE EFFICIENCY	= $\frac{3413}{10,690} \times 100 = 31.9 \%$

NOTE: STEAM GENERATOR DRUM BLOWDOWN TO WASTE IS MADE UP TO CONDENSER

FIGURE 4-49 100 MW_e LINE-FOCUS SOLAR CENTRAL POWER PLANT NET AT MGL (Heat balance at 25% of guaranteed throttle flow)

ENGLISH

LEGEND

SI

LEGEND

W = LB/HR = .4535 kg/HR
 P = PSIA = 6.895 kPa
 F = °F = (F - 32) / 1.8 °C
 H = BTU/LB = 6.46 x 10⁴ KWHR/kg
 HEAT RATE = BTU/KWHR = 1.055 kJ/BTU

TURBINE TYPE: G.E. TC2F - 16.5" L.S.B. 3600 RPM

STEAM GENERATOR DUTY = 865.47 x 10⁶ BTU/HR

GROSS UNIT OUTPUT = 107,205 KWe

GROSS TURBINE CYCLE HEAT RATE = $\frac{865.47 \times 10^6 \text{ BTU/HR}}{107,205 \text{ KWe}} = 8,073 \frac{\text{BTU}}{\text{KWHR}}$

GROSS TURBINE CYCLE EFFICIENCY = $\frac{3413}{8,073} \times 100 = 42.3 \%$

NET UNIT OUTPUT = 100,000 KWe

NET CYCLE HEAT RATE = $\frac{865.47 \times 10^6 \text{ BTU/HR}}{100,000 \text{ KWe}} = 8,655 \frac{\text{BTU}}{\text{KWHR}}$

NET CYCLE EFFICIENCY = $\frac{3413}{8,655} \times 100 = 39.4 \%$

NOTE: STEAM GENERATOR DRUM BLOWDOWN TO WASTE IS MADE UP TO CONDENSER

FIGURE 4-50 100 MW_e LINE-FOCUS SOLAR CENTRAL POWER PLANT NET AT MGL (Heat balance at maximum guaranteed load)

LEGEND

ENGLISH	SI
W - LB/HR	- .4535 kg/HR
P - PSIA	- 6.895 kPA
F - °F	- (F-32) / 1.8 °C
H - BTU/LB	- 6.46 x 10 ⁴ KWHR / kg
HEAT RATE - BTU / KWHR	- 1.055 kJ / BTU
TURBINE TYPE: G.E. TC2F - 16.5" L.S.B. 3600 RPM	
STEAM GENERATOR DUTY	- 944.02 x 10 ⁶ BTU/HR
GROSS UNIT OUTPUT	- 117,275 KWe
GROSS TURBINE CYCLE HEAT RATE	- $\frac{944.02 \times 10^6 \text{ BTU/HR}}{117,275 \text{ KWe}} = 8,050 \frac{\text{BTU}}{\text{KWHR}}$
GROSS TURBINE CYCLE EFFICIENCY	- $\frac{3413}{8,050} \times 100 = 42.4\%$
NET UNIT OUTPUT	- 109,583 KWe
NET CYCLE HEAT RATE	- $\frac{944.02 \times 10^6 \text{ BTU/HR}}{109,583 \text{ KWe}} = 8,615 \frac{\text{BTU}}{\text{KWHR}}$
NET CYCLE EFFICIENCY	- $\frac{3413}{8,615} \times 100 = 39.6\%$

NOTE: STEAM GENERATOR DRUM BLOWDOWN TO WASTE IS MADE UP TO CONDENSER

FIGURE 4-51 100 MWe LINE-FOCUS SOLAR CENTRAL POWER PLANT NET AT MGL (Heat balance at VWO + 5% O.P. load)

Table 4-40

GROSS TURBINE OUTPUT

<u>Load Condition</u>	<u>Gross Output (kW_e)</u>
25% MGL throttle flow	26,213
MGL	107,205
VWO, 5% OP	117,275

Plant auxiliary loads were calculated for a number of combinations of field and turbine-generator operating conditions. The loads are summarized in Table 4-41.

The change from day to evening operation affects the auxiliary power requirements for the salt pumps, receiver control valves, collector drives, cooling tower fans, and master control subsystem.

4.6.3.1.3 Net Electric Power Output

The net electric output of the plant is derived by subtracting the auxiliary power requirements from the gross electrical output. The auxiliary power requirements change during the day; therefore, for a given turbine load, the net electric output also changes. The net plant output is minimal at noon, increases in the morning or the afternoon, and is maximal in the evening. The net plant outputs and corresponding heat rates for the load conditions outlined in Table 4-41 are summarized in Table 4-42.

4.6.3.2 EPGS Operation

This section outlines the hot startup, warm startup, cold startup, and programmed shutdown of the EPGS.

FIGURE 4-52 STEAM GENERATOR TEMPERATURE PROFILES

Table 4-41

PLANT AUXILIARY LOADS (kW_e)

	25% MGL		MGL		VWO, 5% OP
	Morning	Evening	Noon	Evening	Noon
<u>EPGS</u>					
Feedwater pumps	1240	1240	2255	2255	2377
Condensate pumps	84	84	153	153	161
Cooling tower pans	240	120	437	220	461
Circulating water pumps	267	267	486	486	512
Auxiliary systems	275	275	500	500	527
Miscellaneous	106	106	192	192	202
<u>HTS</u>					
EPGS cold salt pumps	330	--	1026	--	1192
ESS cold salt pumps	193	--	1767	--	1767
ESS hot salt pumps	129	56	--	567	82
<u>RS</u>					
Receiver control valves	100	--	100	--	100
<u>CS</u>					
Collector drives	211	--	211	--	211
<u>MCS</u>					
Master control subsystem	100	50	100	50	100
Total	3275	2198	7227	4423	7692

4.6.3.2.1 Hot Startup

Under periods of clear weather, the EPGS operates approximately 20 hours per day during the summer and 10 hours per day during the winter. The EPGS is shut down each evening for 4 to 14 hours.

Table 4-42

NET EPGS CYCLE PERFORMANCE

	25% MGL		MGL	VWO, 5% OP	
	Morning	Evening	Noon	Evening	Noon
Gross electric output (kW _e)	26,213	26,213	107,205	107,205	117,275
Auxiliary power (kW _e)	3,275	2,198	7,227	4,423	7,692
Net electric output (kW _e)	22,938	24,015	99,978	102,782	109,583
Net heat rate (kJ/kWh)	11,278	10,772	9,133	8,883	9,089

Thermal fatigue limits for turbines exposed to many startup and shutdown cycles require restrictions on the time required to bring a turbine to full operating temperature, speed, and load. The extent of the startup restrictions varies considerably with the turbine design. Applications involving cycling of peaking units, where necessary to produce the rated capacity quickly, require turbines that accommodate rapid heating without overstressing the casing or rotor. A number of machines of this type can be brought to full capacity in less than 15 minutes; however, these designs involve an increase in turbine cost and a decrease in full load performance.

The turbine for the conceptual design does not require this rapid start capability. Although the turbine operates on a daily cycle, the plant is intended to meet intermediate load requirements of the grid. The plant delivers power to the grid when thermal energy is available from the solar energy system; it is not designed to respond quickly to rising peak loads.

The turbine cooldown and cyclic life information for double-shell turbines, such as the General Electric machine selected for the conceptual design, indicates that the turbine cooldown during load

reduction and shutdown permits startup times of 1.0 to 1.5 hours without impairing the 30-year turbine life. With respect to the time required to bring the HTS and RS to operating temperature in the morning, this startup time is considered acceptable.

As a hot startup is initiated, the high-pressure turbine is at temperature between 644 K (700°F) and 727 K (850°F). This assumes that all valves to the turbine are closed and that no forced cooling has occurred. The steam generator is in the standby mode. In this mode, 533 K (500°F) water at 7.1 MPag (1030 psig) circulates through the steam drum, boilers, and preheaters. Salt from the cold storage tanks circulates through the superheaters, reheater, boilers, and preheaters to maintain the pressurized water temperature.

The boiler, superheater, and reheater drain-and-vent valves are set in the startup position. The auxiliary boiler is brought to rated capacity to supply the auxiliary startup steam. The salt flow to the steam generator increases in temperature by blending salt flows from the cold and hot storage tanks. The deaerator is pegged with auxiliary steam and the startup feedwater pump is operated to establish a small feedwater flow. The full-flow condensate polisher minimizes the introduction of corrosion products into the steam generator and allows continued operation of the steam generator without load restriction during periods of minor condenser leakage. Before establishing the condenser vacuum, all reheater drains and vents are closed. Drains and vents connected to the condenser may be left open until the turbine is carrying a light load.

Once condenser vacuum and circulating water flow have been established, the main steam bypass line to the condenser opens, producing superheated steam in the steam generator from the small feedwater flow. The steam preheats the main steamline and turbine steam chest and then is dumped to the condenser. A sufficient steam flow is maintained to clear the superheater tubes and headers of all condensate. The salt flow to the steam generator increases in temperature until the superheated steam temperature is 56 K (100°F) greater than the

temperature of the HP turbine metal. At this time, the turbine control valves are opened and the main steam bypass line is closed.

The turbine is brought to 3600 rpm and the generator is synchronized with the grid. The salt temperature to the steam generator increases, at a rate of approximately 83 K (150°F) per hour, until the salt reaches 839 K (1050°F). At this time, the main steam conditions are 16.5 MPag (2400 psig) and 811 K (1000°F) and the turbine-generator is producing 3 to 5 MW_e. Because of main steam throttling, the HP turbine metal temperature is approximately 755 K (900°F). Note that the energy required to start the turbine and bring it to the 3- to 5-MW_e condition is supplied by the ESS. Depending on the duration of the overnight shutdown, the operation takes 1 to 2 hours. Since the energy is supplied by the ESS, the operation may be started in the morning before solar thermal energy is available from the field. This startup operation does not add to the 1- to 1.5-hour period in the morning that is required to bring the turbine from minimum load to design load.

Once the 3- to 5-MW_e load is established, the steam source for the deaerator is switched to the cold reheat line. The balance of the auxiliary steam requirements are switched to the turbine cycle and the auxiliary boiler is phased out. The feedwater pump duty is switched from the startup feedwater pump to the primary feedwater pumps.

At this time, the load on the turbine is increased such that the change in HP turbine metal temperature does not exceed 83 K (150°F) per hour. Under these conditions, the turbine should reach full load within 1 to 1.5 hours. As mentioned before, this is not a strong operating constraint. The morning solar insolation curves are such that approximately 1 hour is required for the receiver subsystem to deliver sufficient thermal energy to produce 100 MW_e.

4.6.3.2.2 Warm Startup

A warm startup occurs following a turbine shutdown of 18 to 72 hours. Under these conditions, the HP turbine metal temperature is between 422 K (300°F) and 644 K (700°F). The startup is similar to the hot

startup outlined above, but the turbine heating period using thermal energy from storage is somewhat longer.

4.6.3.2.3 Cold Startup

A cold startup occurs following plant construction, extended shutdowns during cloudy weather, or turbine overhaul. Preparatory work for the startup includes the following:

- Chemical cleaning of the equipment
- Blowing the steam lines
- Setting the safety valves
- Filling the feedwater train, preheaters, boilers, and steam drum with properly treated feedwater
- Checking all control systems, mechanical equipment, and instrumentation to assure availability
- Preparing the turbine-generator in accordance with the instructions of the manufacturers

The auxiliary boiler is started to preheat the turbine and feedwater train and to supply the auxiliary steam requirements. The steam generator is preheated as outlined in Section 4.4.3 and filled with salt from the cold storage tanks. The balance of the cold startup is similar to the warm startup. Cold startup requires approximately 2 hours to preheat the turbine to 422 K (300°F) with auxiliary steam and approximately 5 hours to bring the HP turbine to operating temperature, using superheated steam from the steam generator.

4.6.3.2.4 Programmed Shutdown

During a programmed shutdown, the load on the turbine is reduced such that the change in the HP turbine metal temperature does not exceed 83 K (150°F) per hour. As the turbine load is reduced, the low pressure feedwater heaters are removed from service. Once the load has been reduced to 3 to 5 MW_e, the turbine is tripped immediately. The superheater and reheater drains are opened to remove any condensate from the main and reheat steamlines.

During the programmed shutdown, the temperature of the salt entering the steam generator is maintained at 839 K (1050°F). Main steam conditions during the shutdown remain at 16.5 MPag (2400 psig) and 811 K (1000°F). Although considerable throttling of the main steam occurs at the low-load operations, the throttling is isenthalpic. Under these conditions, the minimum HP turbine inlet temperature is 750 K (890°F). When the turbine is tripped, the HP turbine metal temperature is 750 K (890°F). Since this temperature is not significantly lower than the operating metal temperature of 811 K (1000°F), the programmed shutdown of the turbine does not seriously affect the time required to restart the turbine the following day.

Once the turbine-generator is removed from the grid, the main steam bypass line to the condenser is briefly opened to prevent overpressurizing the steam generator. The steam generator is then cooled to 567 K (560°F) and placed in standby, as outlined in the hot startup procedure. The steam generator is cooled by ramping down the salt temperature supplied to the unit from 839 K (1050°F) to 567 K (560°F). The intermediate temperature salt supplied for the cooling is produced by blending salt from the hot and cold storage tanks. The intermediate temperature salt leaving the steam generator during cooldown is directed to the cold storage tank, which stores the residual heat from the steam generator by slightly increasing the temperature of the cold storage tank salt inventory.

4.6.3.3 EPGS Control

Conventional power plant controls are required to operate the EPGS. These consist of extensive local and control board instrumentation for monitoring critical equipment, and alarms and turbine trip provisions to guarantee operational safety and prevent damage to hardware. The major control affecting turbine-generator steam generator and feedwater system operation is the three-element feedwater control. Its function is to control feedwater flow to satisfy the turbine steam demand without permitting excessive fluctuations in steam drum water

level. EPGS load indicators, such as the turbine admission valve stem position and steam drum pressure, are monitored by the master control subsystem to control steam generator salt flow.

Controls dealing with the salt side of the steam generator are discussed in Section 4.4.3.

4.7 Master Control Subsystem

4.7.1 General

The master control subsystem (MCS) supplies all required operational and safety controls and monitors all subsystems. The primary requirements imposed upon the MCS are simplicity, reliability, and cost effectiveness of both design and operation. The MCS uses standard control practices and simple, well-defined interfaces with other subsystem controls. Primary operation is automatic, but manual override is also provided.

Reliability is assured by using proven designs and eliminating single-point failure through the use of redundant elements whenever it is cost-effective. Failure of the MCS does not prevent manual operation of the plant. The MCS contains independent control within each plant subsystem; coordination of the subsystem control is provided by the central MCS.

4.7.2 MCS Design

4.7.2.1 Introduction

The master control subsystem coordinates and controls all other subsystems, as indicated in Figure 4-53. The overall functions of the MCS for this line-focus power system are substantially the same as those for the point-focus-type of solar power system; however, MCS functions for the line-focus system are considerably simpler. Since every heliostat rotates on a single axis through the same angle, the line-focus MCS can use distributed controllers and modularity; this combination leads to a simple, low-cost and reliable control subsystem. The distributed controllers communicate with each other and with a central computer.

4.7.2.2 Central Computer

The central computer for the MCS is shown in Figure 4-53, which gives an overview of the entire control subsystem. Figure 4-54 shows

FIGURE 4-53 MASTER CONTROL SUBSYSTEM

FIGURE 4-54 CENTRAL COMPUTER COMPLEX

the central computer consists of several parts: an operator's console with displays and manual input controls; a plant operational control (POC) minicomputer; a peripheral control and data acquisition (PCDA) minicomputer; associated disc and backup tape storage; and a single-board computer called the array controller. Overall supervision of the line-focus power system is provided from the central computer by the POC. Under normal operations, the PCDA provides real-time data routing/processing, drives displays, and collects data for performance and maintenance analysis. If the POC fails, the PCDA functions as a backup and performs all the supervisory tasks essential to system operation. Normally, the POC controls plant operation and interfaces with the PCDA through a common memory; interrupt lines are provided for critical parameters. The PCDA also interfaces with a single-board computer (the array controller) to control the array of heliostats. This array

controller relieves the POC and PCDA of routine tasks associated with heliostat operation. The backup for the array controller is an identical single-board computer, which is held in reserve for replacement in the event of failure. The array controller interfaces the central computer to the block controllers shown in Figure 4-53.

4.7.2.3 Control of the Heliostats

The method for communicating between the central computer and the heliostats is an important feature of the line-focus central power system MCS. The MCS selectively transmits information from the central computer to individual heliostats, to a group of heliostats, and to all heliostats. The MCS also transmits routine status information and alarm conditions to the central computer from the heliostats. The entire array of heliostats in the system is divided into 10 identical modules or "blocks." All heliostats in one block are aimed at one row of receivers (there are 10 rows of receivers in the entire system). The configuration of a block is shown schematically in Figure 4-55; 24 rows and 25 columns of heliostats comprise the block, and 6000 heliostat drives are in the entire array. Each heliostat drive has its own local controller. Calibration of the heliostat aiming is by the reflection of a light on the tower to a photocell on the tower. The frequency of programmed aiming accuracy checks will be determined by operating experience.

FIGURE 4-55 ONE BLOCK IN HELIOSTAT ARRAY

The data paths for controlling the heliostat array are shown in Figure 4-56. Commands that originate in the central computer (normally from the array controller) are transmitted to the local controllers for the heliostats via a block controller. The data path from the central computer to the block controllers is a simple two-way serial link. Asynchronous information is simultaneously transmitted to all block controllers by a finite-length message packet (48 bits). Each block controller examines the address of the packet and responds accordingly.

For example, if the message were addressed to take a single heliostat out of service for routine maintenance, the corresponding block controller would relay the message to the individual heliostat local controller (after removing the superfluous bits and reestablishing the error checking bits). Since each block has 24 rows of heliostats, each block controller has 24 two-way serial data paths, and each of the 24 paths interconnects 25 local controllers. The 25 local controllers are addressed by column number within the block. Status information relating to each heliostat is passed from its local controller to the block controller upon command from the block controller; if appropriate, this information is then relayed to the central computer. In addition to relaying routine information to the central computer, alarm conditions are transmitted immediately by priority interrupt.

In the line-focus central power system, all data paths have a very low data rate, including the path between the central computer and the block controllers. Thus, since the bandwidth requirements of the connecting cable are minimal, other parameters govern the cable selection. Twisted pair cable, fiber optics, and power line multiplexing were considered in the tradeoff studies. Twisted pair cable cost less per foot than fiber optics, and the installation costs were similar. Equipment costs for a power line multiplexing system are significant, so direct burial of twisted pair cable was selected.

An important feature of the line-focus central power system is simplicity of tracking: only the elevation angle is changed during tracking. Once each row in a block has been aimed at the receivers

FIGURE 4-56 DATA PATHS FOR CONTROLLING HELIOSTAT ARRAY

during the morning startup, the angular increment remains the same for all heliostats in the field for each elevation update. In addition, the absolute angle of a row of heliostats is the same for similarly positioned rows in all blocks. (Each heliostat is individually calibrated during installation to provide a reference from which to measure elevation angle.) Since the angular update increment is the same for all heliostats, only a single command is needed from the array controller to specify the increment. The calculation of the value of the increment is done in the array controller for the entire field, and this information is transmitted by a single command addressed to all heliostats via the block controllers.

In principle, all heliostats could be rotated by the same angular increment simultaneously. However, to avoid large surges in the power bus feeding the heliostat drive motors, motors are turned on in sequence by the block controllers, which allows a cost-effective sizing of the power distribution system. Since a single command word containing the increment value is sufficient to change the elevation angle of all heliostats periodically, the primary and most frequently used heliostat control is greatly simplified.

The commands issued by the central computer are in the form of a 48-bit serial data stream--a message packet--as illustrated in Figure 4-57. Commands are issued asynchronously by the computer. The format for the address bits is shown in Figure 4-58; four address modes

FIGURE 4-57 48-BIT MESSAGE PACKET
(SEQUENTIAL BIT STREAM)

Operating Mode	Address Mode	Number of Heliostats Responding	Address Bits Required
Run	All	6000	2
Startup, shutdown, run, stow	Group	25 x 10 = 250 (one row in each block)	7
Calibrate, repair, test, startup, shutdown, stow	Individual	1	16

MODE	Bit Number															
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
INDIVIDUAL	0	1	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y
GROUP	1	0	Y	Y	Y	Y	Y	X	X	X	X	X	X	X	X	X
ALL	1	1	X	X	X	X	X	X	X	X	X	X	X	X	X	X
RECEIVER	0	0	X	X	X	X	X	Y	Y	Y	Y	X	X	X	X	X
SELECTION	Mode		Row				Block				Column					

X = don't care
Y = assigned value

FIGURE 4-58 ADDRESSING STRUCTURE

accommodate the various operating modes. The maximum number of bits required for addressing (16) is determined by the total number and configuration of the local controllers. The maximum number of bits required to specify the action to be taken is determined by the angular resolution of the heliostats.

The technique used to implement the local controllers is important because of the large quantity required--one per heliostat. The control functions provided by the local controller in the line-focus system are not demanding; only a simple microcomputer is required (a single-chip microcomputer, called a microcontroller, should be sufficient). The elevation angle for the heliostat is stored in the microcontroller because an incremental shaft encoder is used to measure elevation. For this reason, the random access memory (RAM) in the microcontroller should be nonvolatile; i.e., the information in the memory is retained when power is turned off. For example, a ferrite core RAM is nonvolatile; semiconductor RAMs are not. The microcontrollers currently manufactured do not have on-chip nonvolatile RAMs, but it is reasonable to expect that they will by the mid-1980s. The so-called EEROMs (electrically erasable read-only memories) are not nonvolatile RAMs and are not suitable for this application.

CMOS microcontrollers are desirable because they have good noise immunity, low power drain, and a very wide temperature range. If nonvolatile RAMs are not available in CMOS microcontrollers by the mid-1980s, a custom-designed hybrid circuit containing a CMOS microcontroller chip and a nonvolatile CMOS RAM should be cost effective.

4.7.2.4 Control of the Receivers

To this point we have considered in some detail the control needs for the heliostat array; a second part of the field control system is that which is needed for the receivers. Each row of receivers has 52 receivers associated with one block of heliostats, as shown in Figure 4-55. Each receiver module is 85 ft long and is arranged in mirror-image pairs, typically with one inlet valve regulating the flow

of the draw salt through the pair. For the two receivers (a pair) at the beginning and end of each long receiver row, a separate valve is required by each receiver to accommodate end effects. One local controller (a microcontroller) is associated with each pair of receivers (26 microcontrollers per row). Thermocouples provide temperature inputs to each controller as the basis for flow rate regulation during the normal run mode of the system.

Figure 4-59 shows the data path configuration for the local receiver controllers and block controllers. Each block controller communicates with a row of 26 local receiver controllers in addition to the 648 local heliostat controllers in each block. In the addressing structure discussed above, message packets can be addressed to local receiver controllers from the central computer.

4.7.2.5 Summary

The master control subsystem consists of a central computer complex and multiple distributed controllers. The central computer has a console with displays and manual input controls, and two minicomputers. The central computer coordinates all elements of the MCS. The distributed portion of the MCS has a hierarchy of controllers interconnected by two-way, low-data rate, serial paths. Since the data rate is low, the technical requirements on the connecting data paths (cables) are modest.

Modularity is an important characteristic of this system. For example, 10 identical modules comprise the heliostat array controls, and within these modules are submodules that are likewise identical. Modularity facilitates the initial design, installation, routine maintenance, and subsequent plant expansion. It also increases overall reliability of the system. Another characteristic of significance is the simplicity of the heliostat control during tracking. In the line-focus central power system, the incremental heliostat rotation is the same for all heliostats in the field for each elevation update. This means that for each update only a single calculation is made, and the resulting elevation increment is sequentially transmitted to all heliostat controllers.

FIGURE 4-59 DATA PATHS FOR RECEIVER CONTROL

4.7.3 Autonomous Subsystem Controls

As illustrated in Figure 4-53, the master control subsystem coordinates and controls all other subsystems in the line-focus power system. The MCS is configured in a distributed manner, but there are no autonomous subsystem controls outside the MCS.

REFERENCES, SECTION 4

- 4-1 Isogrid Design Handbook, McDonnell Douglas Astronautics Company, February 1973.
- 4-2 Ali, M. A., Majumbar, A. J. and Singh, B., Properties of Glass Fiber Cement - The Effect of Fiber Length and Content, Journal of Materials Science, Vol. 10, pp. 1732-1740, 1975.
- 4-3 Lankard, D. R., Fiber Reinforced Cement-Based Composites, Ceramic Bulletin, Vol. 54, No. 3, pp. 272-276, 1975.
- 4-4 Blackman, L. C. F. et al., Glass Fiber Reinforced Cement - A New Material for Construction and Engineering, Chartered Mechanical Engineer, pp. 45-51, January 1977.

5.0 PERFORMANCE AND COST ESTIMATES

5.1 Introduction

5.1.1 Strawman Comparison

The purpose of this Phase I study was to compare the cost and performance of the line-focus central power system with the first generation "power tower" point-focus central power system. Table 5-1 compares the "strawman" design in the RFP and the design selected. Two major differences exist: the use of salt as the receiver working fluid (which simplifies the storage system and allows full power operation from storage) and the increased storage capacity.

Table 5-1

DESIGN CHARACTERISTICS

80th Point-Focus Tower ("Strawman")	80th SRI Line-Focus Tower System
System	System
Water receiver	Molten salt receiver
100-MW electrical steam, Rankine cycle	100-MW electrical steam, Rankine cycle
Solar multiplier, 1.7	Solar multiplier, 2.46 (equinox noon) as a result of assumed load factor, 0.6 on 100-MW _e rating
6-h storage at 70 MW _e	7.5-h storage at 100 MW _e
Caloria	Salt
Physical	Physical
$0.78 \times 10^6 \text{ m}^2$ heliostat area	$0.9 \times 10^6 \text{ m}^2$ heliostat area
BBEC = 84 mil/kWh	BBEC = 61 mil/kWh

A cost comparison for the first and 80th plants in terms of $\$/kW_e$ is shown in Table 5-2. Parametric design cost estimates for the first plant are computer-calculated. Values for the 80th plant are the result of adjusting the receiver and tower costs according to a learning curve of 0.95. The indirect costs are obtained by using the same percentages as the strawman.

The conceptual design cost estimates are based on the costs from subcontractor estimates. These costs were adjusted by SRI to provide for mass production of the design, where applicable. The conceptual design was based on a capacity factor of 0.56 versus 0.60 for the parametric design.

The capital cost per rated power is greater for the 80th line-focus central power plant than for the point-focus central power strawman. This is because the line-focus design is larger and has a greater storage capacity, which increases the average daily capacity factor. The resulting cost of electricity is lower than for the strawman.

If the strawman capacity factor were increased, or if the strawman were rated at the off-storage power of $70 MW_e$, the line-focus design would also show superiority in cost per installed capacity. The line-focus central power plant is superior because the heliostats gather more energy in proportion to their cost. Again, the optical "efficiency" of the single-axis heliostat is less than the two-axis heliostat, but the reduced cost more than offsets the decreased efficiency. The most important criterion, however is the cost of electricity; in this respect, the 80th line-focus central power plant generates electricity at a cost of 61 mil/kWh, which is superior to the cost of the 80th strawman point-focus central power system (83 mil/kWh). Even further BBEC reductions would be possible with an incremental startup of the line-focus tower system.

5.1.2 Costing Strategy

The strategy followed during the performance and cost estimation process was:

Table 5-2

BASELINE CAPITAL COSTS
(\$/kW_e)

Item	Power Tower Strawman		Parametric Design (c.f.=0.6)		Conceptual Design (c.f.=0.6)	
	1st	80th	1st	80th	1st	80th
Land, yardwork	5	6	3.8	3.8	43.14	33.69
Structures	51	39	39.0	39.0	9.92	7.74
Turbine plant	242	187	263.1	263.1	228.57	178.48
Electric plant	88	68				
Collectors	695	504	525.5	525.5	439.30	343.02
Receiver	185	108	364.8	263.6	211.20	164.91
Tower and trusses	124	124	101.2	64.2	66.88	52.22
Heat transport sys.	0	0	135.3	135.3	286.67	223.84
Thermal storage	215	156	110.7	110.7	74.67	58.30
Distributables	86	36	36.0	36.0	84.77	66.19
BOP	53	38	38	38	138.08	107.82
Direct cost	1,744	1,262	1,617.4	1,479.2	1,583.20	1,236.22
Indirect cost	348	136	322.7	159.4	515.32	402.38
Total	2,092	1,398	1,940.1	1,638.6	2,098.52	1,638.60

*The plant cost shown above is based on \$5.05/ft² with an assumed production rate of 75,000 heliostats per year. Costs below this are projected for designs using glass fiber reinforced cement as a substrate for the reflective surface. Note that a learning curve reduction for the 80th plant is only used for the receiver. A learning curve reduction for other components of the 80th plant would reduce these costs even further.

- Estimate costs of subsystem parametrically
- Optimize the plant on the basis of the above costs
- Perform a conceptual design to the parameters selected
- Estimate the cost and performance of the conceptual design.

Ideally, the cost estimates from the last step would match those in the second step. In fact, the cost estimates have differed because of changes and refinements made during the conceptual design.

The line-focus power plant requires an advanced technology that is very different from conventional power plants, so much so that cost estimating procedures used for conventional power plants are inadequate. The solar power plant, for example, contains many duplicated components that can be automated, such as the 160-ft heliostat modules, which are repeated 6,000 times. In turn, each 160-ft length is made of four duplicate components. Further examples would be the 80-ft receiver module, which is repeated 550 times, and the basic tower and truss unit, which is duplicated 276 times. Such duplication makes feasible the use of mass production techniques, with resulting cost reductions.

When a trench is to be dug, for example, the cost should not be estimated using a one-of-a-kind "back hoe" formula. What must be considered is that many identical trenches will be dug throughout the plant, and that specialized equipment can result in substantial savings. The same applies when setting foundations, erecting towers, or even lagging lines. The site preparation costs should be carefully calculated, taking into consideration that the solar plant is roughly ten times the area of a conventional power plant.

The receiver cost estimators formulated their values for the receiver on the basis of a single 80-ft section. They multiplied their estimates by the number of such sections and took into account the benefits of mass production, assuming a learning curve of 97% (i.e., the cost per unit decreases to 97% of the value each time the cumulative volume doubles). Experience in aircraft, automobile, and other automated manufacturing operations indicates that learning curves of 90% are conservative for labor and materials. Learning curves for labor alone can be in the range of 70% to 80%.

The RS and HTS cost estimators considered the price reductions afforded by bulk orders of steel, valves, etc., but did not use learning curves. They added the indirect costs in proportion, costs which clearly would be reduced by a considerable degree of duplication in the architectural and engineering effort. Hence, these costs are also believed to be conservative.

These estimates, considered by SRI to be too conservative, are included in this report without alteration. SRI has modified these estimates with a conservative learning curve of 0.9. This approach gives results very much like those generated during the parametric study.

The principles of the learning curve as applied to solar energy are discussed in a separate publication (Ref. 5-1), which shows that the cost of production, y_x , of the xth unit is given by

$$y_x = ax^{-b} \quad (5-1)$$

where

a = cost of the first unit

b = measure of the rate of reduction

The parameter b is related to "the percent learning that occurs each time output is doubled," the progress index (PI), as

$$PI = 2^{-b} \quad (5-2)$$

For a progress index or learning curve of 0.9, $b = 0.152$.

To include the learning effect cost of the first plant, SRI first determined the basic unit that the estimator used in costing the plant. By knowing the basic unit, the number of units in the plant, and the total cost, the cost of the first unit can be determined. The learning curve costs can then be integrated to give the revised estimate. The

formula for the revised estimate, E_r , in terms of the original, E_o , is

$$E_r = E_o \frac{\left(\frac{N}{q}\right)^{-b}}{1 - b} \quad (5-3)$$

where

N = number of units in the plant

q = number of units that formed the basis for the estimate

Estimates for the 80th plant are based on estimates for the first plant with the learning curve of equation (5-1) applied with a progress index (PI) = 0.95 and $b = 0.074$ to all costs except the salt. The procedure is valid, although it does not distinguish those components for which cost will be reduced the most. Although labor costs are the ones emphasized in learning factor considerations, cost savings also result from the reduction and the substitution of materials of construction. One example is the use of glass-reinforced concrete (GRC) rather than plastic substrates proposed for the heliostat mirrors. A learning progress index of 0.95, which is historically quite conservative, was used for estimating the costs of the 80th plant.

Table 5-3 shows the cost estimates (without learning considerations), the basis of cost estimating, annotations, and finally the SRI learning factor adjusted cost estimate. The results are somewhat different than the parametric estimates, but the conformity is sufficiently close to validate the parametric plant design. The cost estimates include all overhead, profit, and contingencies.

The cost for the first plant, according to learning factor analysis, and taking into consideration the effects of automation, on certain subsystems is thus:

Table 5-3

CONCEPTUAL DESIGN COSTS
(millions of dollars)

Subject	Account	Reporter	Reported Value	Adjustment	N/q	c.f. = 0.56		c.f. = 0.60	c.f. = 0.40
						1st	80th	1st	1st
Land and preparation	5100	Bechtel	4.720		10	3.922	3.062	4.314	2.801
Administrative	5200	Bechtel	0.830		1	0.830	0.648	0.830	0.830
Collectors	5300	Accurex	39.936		1	39.936	31.183	43.930	28.526
		Bechtel	11.205	-11.205 duplication					
Receivers	5400	Foster- Wheeler	31.593	+0.75	520	19.200	14.992	21.120	13.714
Towers and trusses	5400	Bechtel	12.331		310	6.080	4.747	6.688	4.343
Master Control	5500	Bechtel	0.162		1	0.162	0.126	0.162	0.162
		SRI	1.101		1	1.101	0.860	1.101	1.101
Heat transport	5600	Bechtel	34.846		20	26.061	20.349	28.667	18.615
Thermal	5700	Bechtel	7.467		1	7.467	5.830	7.467	2.339
Electric power system	5800	Bechtel	22.857		1	22.857	17.848	22.857	22.857
Salt at 15¢/lb			7.376		1	7.376	5.759	7.376	7.376
Balance of plan and contingency			18.366			13.106	10.234	13.808	10.114
<hr/>									
Total Direct			192.790			148.098	115.638	158.320	112.828
Indirect:		Acurex	8.544			8.544	6.671	9.398	6.103
		Bechtel	45.467			32.446	25.335	34.183	25.039
		Foster-Wheeler	11.711			7.117	5.557	7.829	5.084
		SRI	0.122			0.112	0.087	0.112	0.112
Total Indirect			65.844			48.219	37.650	51.522	36.338
<hr/>									
TOTAL COST			258.634			196.317	153.288	209.842	149.166

\$196,327,000 for the 0.56 average daily capacity factor plant,

or

\$209,852,000 when adjusted to an 0.60 average daily capacity factor.

This is to be compared with \$194,010,000 for the first plant parametric analysis, including indirect costs.

The learning factor estimate for the 80th plant of 56% capacity factor is \$153,299,000 and \$163,859,000 for the 0.60 capacity factor, based upon the conservative performance index of 0.95 for the receiver subsystem only and 1.00 for all other subsystems.

Figure 5-1 shows the total investment conceptual design cost of the first 100-MW_e line-focus tower solar thermal power plant as a function of capacity factor. The economic analysis and market

FIGURE 5-1 LINE-FOCUS TOWER SOLAR THERMAL PLANT INVESTMENT COST

penetration study (Section 7.0) were based on the cost of 40% capacity factor plants in areas of lesser insolation.

5.2 System Performance and Cost Estimates

Figure 5-2 shows the system performance in terms of $\overline{\text{BBEC}}$ as a function of capacity factor for several different storage capacities. The 750-MW_{th} storage capacity indicates the performance of the plant selected for a capacity factor of 0.6. Figures 5-2 through 5-5 are "waterfall" charts showing the power losses at winter, spring-fall, and summer noon. Figure 5-6 shows the annualized total energy.

The efficiency during operation from thermal storage is the same as operating directly from solar energy since the draw salt is stored at the same temperature. The thermal losses from the storage system are

FIGURE 5-2 LEVELIZED BUSBAR ELECTRIC COST ($\overline{\text{BBEC}}$)

FIGURE 5-3 NOON POWER FLOW - WINTER

FIGURE 5-4 NOON POWER FLOW - SPRING

FIGURE 5-5 NOON POWER FLOW - SUMMER

FIGURE 5-6 ANNUAL ENERGY PERFORMANCE

accounted for as continuous energy losses. Waterfall charts for other days and times are shown in Figures 5-7 through 5-12.

5.3 Discussion

This performance discussion outlines the energy and mass balances for each plant operating mode, the performance for the key days of the year, and the integrated annual energy balance.

5.3.1 Energy and Mass Balances for Each Operating Mode

The energy and salt flows to and from the HTS, ESS, and EPGS for seven plant operating modes are summarized in Table 5-4. The receiver subsystem delivers thermal energy to the HTS. The energy is, in turn, delivered to the ESS or the EPGS. The ESS is the only subsystem which provides a significant amount of thermal storage. Therefore, energy is transferred to and from the ESS only.

The values shown in the table correspond to gross, rather than net, energy transferred. For example, the energy transferred to the HTS includes only the component received from the RS. The energy supplied to the HTS salt pumps from the EPGS is not included. Similarly, the energy supplied to the EPGS corresponds to the gross turbine-generator output.

The seven operating modes presented in the table have been specified as follows:

- Maximum Power--In this mode, the RS is supplying sufficient power to operate the turbine-generator in the VWO, 5% OP load condition. Morning or afternoon insolation conditions have been assumed, in that the RS is supplying sufficient energy to drive the turbine-generator only. The energy delivered to the ESS is assumed to be negligible.
- Normal Solar Power--Summer noon insolation conditions were assumed for this mode. The energy delivered by the RS is used to drive the turbine-generator at the maximum guaranteed load. The thermal energy not used by the EPGS is transferred to the ESS.

FIGURE 5-7 RATED POWER: WINTER AT 8:50 a.m.

FIGURE 5-8 RATED POWER: SPRING AT 8:32 a.m.

FIGURE 5-9 RATED POWER: SUMMER AT 8:07 a.m.

FIGURE 5-10 FULL DAY ENERGY - WINTER

FIGURE 5-11 FULL DAY ENERGY - SPRING

FIGURE 5-12 FULL DAY ENERGY - SUMMER

Table 5-4

ENERGY AND MASS BALANCES

Operating Mode	Flows to/ from Subsystem	Energy Transferred,		Total Salt Flow Rate (10 ⁶),	
		MW _t (10 ⁶ Btu/h)		kg/h (lb/h)	
Maximum power	To HTS	276.6	(944.1)	2.37	(5.21)
	To ESS	0.0	(0.0)	0.00	(0.00)
	From ESS	0.0	(0.0)	0.00	(0.00)
	To EPGS	276.6	(944.1)	2.37	(5.21)
Normal solar power	To HTS	631.5	(2,155.3)	5.40	(11.88)
	To ESS	377.9	(1,289.8)	3.23	(7.11)
	From ESS	0.0	(0.0)	0.00	(0.00)
	To EPGSS	253.6	(865.5)	2.17	(4.77)
Low solar power	To HTS	126.8	(432.8)	1.08	(2.38)
	To ESS	0.0	(0.0)	0.00	(0.00)
	From ESS	126.8	(432.8)	1.08	(2.38)
	To EPGS	253.6	(865.5)	2.17	(4.77)
Intermittent cloudiness	To HTS	126.8	(432.8)	1.08	(2.38)
	To ESS	0.0	(0.0)	0.00	(0.00)
	To ESS	126.8	(432.8)	1.08	(2.38)
	To EPGS	253.6	(865.5)	2.17	(4.77)
Charging thermal storage	To HTS	631.5	(2,155.3)	5.40	(11.88)
	To ESS	631.5	(2,155.3)	5.40	(11.88)
	From ESS	0.0	(0.0)	0.00	(0.00)
	To EPGS	0.0	(0.0)	0.00	(0.00)
Evening	To HTS	0.0	(0.0)	0.00	(0.00)
	To ESS	0.0	(0.0)	0.00	(0.00)
	From ESS	253.6	(865.5)	2.17	(4.77)
	To EPGS	253.6	(865.5)	2.17	(4.77)
Overnight shutdown	To HTS	0.0	(0.0)	0.00	(0.00)
	To ESS	0.0	(0.0)	0.00	(0.00)
	From ESS	9.8	(33.4)	0.08	(0.18)
	To EPGS	0.0	(0.0)	0.00	(0.00)

- Low Solar Power--The solar insolation conditions for this mode correspond to early morning or late afternoon. The turbine-generator is assumed to be operating at the maximum guaranteed load. The thermal energy required by the EPGS comes from a combination of RS and ESS inputs.
- Intermittent Clouds--For the system of the conceptual design, this operating mode is identical to the low solar power operating mode.
- Charging Thermal Storage--For this operating mode, summer noon insolation conditions were assumed and the entire thermal output of the RS is transferred to the ESS.
- Evening--In this mode, the thermal output of the RS is zero and all of the energy supplied to the EPGS is transferred from the ESS. The turbine-generator is assumed to be operating at the maximum guaranteed load condition.
- Overnight Shutdown--During an overnight plant shutdown, the turbine-generator is removed from the grid and thermal input to the EPGS is zero. A small amount of energy is transferred from the ESS to provide freeze protection for the field piping and the steam generator.

5.4 Cost Estimates

5.4.1 Cost Accounting Categories

5.4.1.1 General Requirements

The estimate is based on the conceptual design and engineering information prepared for the study in the form of engineering drawings, outline specifications, and equipment lists. The estimate assumes that the facility is the first commercial plant. Estimating methods consistent with the conceptual nature of the design information were employed and rely on informal vendor contact as well as information extrapolated from subcontractors' historical data. SRI applied the appropriate learning curves to the subcontractors' estimates, where applicable.

The construction cost estimate is composed of field costs, engineering services, contingency, and fee. The largest category, field costs, comprises the direct cost of permanent plant equipment and labor, and the indirect cost of temporary construction materials, supervision, etc., that are to be distributed across the entire

facility. The estimate anticipates an engineer-constructor direct-hire operation employing field construction labor forces.

5.4.1.2 Price Levels

The estimate is based on price and wage levels for the second quarter of 1979. Allowance has been made for future escalation in the economic evaluation (Section 7.0).

5.4.1.3 Field Construction Costs

5.4.1.3.1 Equipment

Budgetary suppliers' estimates based on the conceptual designs and specifications were obtained verbally and in writing for the following major equipment items:

- Turbine-generator (General Electric)
- Main condenser (Ecolaire Incorporated)
- Feedwater heaters (Southerwestern Engineering Company)
- Steam generator and accessories (Struthers-wells Corporation)
- Fired heaters (KTI)
- Pumps with variable speed drives (Linde Hydraulics Corporation and Von Roll Ltd.)

Other equipment was estimated using the SRI team members' data derived from recent purchase orders and vendor contacts.

5.4.1.3.2 Material

Quantity evaluations were made from drawings for the identifiable materials of concrete, steel, pipe, wire, cable, etc., and these quantities were priced using current unit prices from team members, sources. Other materials that would be delineated on the final drawings have been evaluated as a percentage of the installed cost of the identified materials.

The general level of pricing includes the following:

- Civil-Structural
 - Delivered concrete, \$40/yd³
 - Reinforcing steel, \$400/ton
 - Fabricated embedded metal, \$0.80/lb
 - Fabricated structural steel, \$1000/ton
- Electrical
 - Cable tray (24 in), \$11/ft (linear)
 - Cable (5kV, 3C, 250 mcm), \$8.70/ft (linear)
 - Cable (600V, 3C #10), \$0.78/ft (linear)
 - Rigid conduit (3 in), \$4.75/ft (linear)
- Piping, including fittings and freight
 - Carbon steel (Schedule 10S), \$0.38/lb
 - Stainless steel (Schedule 10S), \$2.30/lb
 - Stainless steel (Schedule 20S), \$2.19/lb
- Valves--Vendor quotes were obtained for all major valves in the ESS and HTS. The vendors were:
 - Valves, Inc. (ball valves)
 - Clow Corporation (check valves)

5.4.1.3.3 Subcontracts

Subcontract costs for equipment and materials commonly installed by subcontractors were estimated and priced in accordance with team members' experience from current sources. Major subcontracts for the project are assumed to be let for the fabrication of all on-site steel tanks, pipe insulation, tank insulation, and the augering of the collector foundations.

5.4.1.3.4 Construction Labor

The direct construction cost for the installation of the plant equipment and material was estimated using recent productivity experience and a wage rate based on labor contracts and fringe benefits typical for Barstow, California.

The wage rate of \$13/h used in the direct field cost estimate is based on a craft mix appropriate to this construction. Fringe benefits such as health and welfare, pension and vacation, F.I.C.A. taxes, workman's compensation, allowances for casual overtime, showup time, travel and subsistence are included in the indirect field costs. Sufficient manual labor to complete the project within the construction schedule was assumed to be available in the project vicinity, or to be attracted to the area by the travel allowance.

5.4.1.3.5 Indirect Field Costs

The indirect field costs are those items of construction costs that cannot be ascribed to the direct costs of the facility and thus are accounted separately. They were estimated from experience gained in modifying fossil-fired plants, resulting in an assessment of 163% of direct labor costs. The indirect field costs have been distributed over the installation of direct equipment and materials as a function of the installation costs.

The items covered by indirect field costs are as follows:

- Temporary Construction Facilities--Temporary buildings, working areas, parking areas, utility system and general purpose scaffolding (24% of direct labor).
- Construction Services, Supplies, and Expense--General job cleanup, maintenance of construction equipment and tools, material handling and surveying, construction equipment, small tools, consumable supplies and purchased utilities (28% of direct labor).
- Field Staff, Subsistence and Expense--Field craft supervision, engineering, procurement, scheduling, personal warehousing, first aid and the cost of operating the field office (23% of direct labor).
- Craft Benefits, Payroll Burdens and Insurances--Associated with the hiring of personnel are certain payroll additives. These include health and welfare, pension, vacation, FICA, workman's compensation taxes, subsistence to cover an allowance for travel, and insurance (64.5% of direct labor).
- Equipment Rental--An allowance to cover the rental of construction equipment such as cranes, earthmoving equipment, trucks, etc. (24% of direct labor).

5.4.1.4 Engineering Services

Engineering services include engineering and other home office costs. Engineering includes preliminary engineering and optimization studies, specifications, detailed engineering, vendor drawing review, site investigation, and support to vendors. Other home office costs comprise procurement, estimating and scheduling services, quality assurance, acceptance testing, and construction and project management.

The sum of these categories falls into historically consistent percentages in the range of 4 and 10%, depending on the complexity of the project. For this study a figure of 5% of field construction costs has been used as typical for a plant that, while new in concept, does not depart radically from basic engineering principles and involves highly repetitious construction features.

5.4.1.5 Contingency

Included in the estimate is an allowance for the uncertainty that exists within the conceptual design in material quantities, quantity pricing, and labor productivity. Implicitly, the allowance will be expended during the design and construction of the project and it cannot be considered as a source of funds for overruns or additions to the project scope. If the conceptual design of the plant components contains major uncertainties or if additional major components are ultimately necessary, then the scope of the project was inadequately defined and this would not be covered by the allowance. A 10% contingency level was used for land and site preparation, administration area, and EPGS. A 15% contingency was used for all other subsystems.

5.4.1.6 Fee

For the given capital cost level of this facility, the estimated fee for the engineer-constructor of 3.75% of the total capital cost. This represents the compensation given over and above the normal expenses experienced by the contractor.

5.4.1.7 Draw Salt

The 1979 cost of draw salt is taken to be 33.08¢/kg (15¢/lb). This is derived from bulk costs of 35.83¢/kg (16.25¢/lb) for KNO_3 and 16.98¢/kg (7.7¢/lb) for NaNO_3 ; or 27.34¢/kg (12.4¢/lb) for the draw salt mixture. The transportation cost of 5.51¢/kg (2.5¢/lb) brings the total cost at the plant site to approximately 33.08¢/kg (15¢/lb).

5.4.1.8 Cost by Categories

The major qualifications in the estimate are presented by cost category in the following pages.

5100 Land, General Site Preparation--The cost of the land was based on recent information issued by Sandia Laboratories (Ref. 5-2). The cost was based on a price of \$890/acre; 680 acres are required.

The cost of land clearing, rough grading, fencing, roads, firewater piping, sewage treatment, yard lighting, grounding and instrumentation are based on recent Bechtel experience. The costs have been appropriately factored to meet specific project requirements and land use. Table 5-5 summarizes the cost breakdown.

5200 Administration Areas--The costs of the administration building, maintenance building, and miscellaneous structures, have been estimated on the basis of Bechtel experience. The costs include HVAC systems and plant equipment, but exclude office equipment and furniture. Table 5-6 summarizes the cost breakdown.

5300 Collector Subsystem--The estimate for the collector subsystem includes the material and installation costs associated with the collector foundations and the electrical distribution equipment for the collector drive motors. The foundation costs are based on the use of a subcontractor for the augering of the foundation holes, with the main contractor placing the reinforcing steel and concrete. The cost has been developed from previous Bechtel construction experience as well as

CONSTRUCTION COST ESTIMATE, ADMINISTRATIVE AREAS

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CONSTRUCTION COST ESTIMATE, LAND AND GENERAL SITE PREPARATION

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verbal quotations from various contractors and material suppliers in the southwestern United States.

The electrical estimate is based on the single-line drawings prepared for this study. It includes the transformers, motor control centers, switchgear, cable tray, and the cable required to supply power to the collector drive motors. The costs of these items are based on takeoff quantities and recent vendor quotations.

Heliostat Cost Breakdown--The heliostat production costs include direct labor and materials, indirect costs, and equipment and tooling. Direct costs include actual labor and materials expended within the production plant for in-progress fabrication, assembly, and packaging operations. Indirect costs include variable and nonvariable direct labor (engineering, supervisory, quality control, material control, etc.), inventory, fringe benefits, facility, process energy, and indirect, consumable materials. Equipment and tooling costs include the amortized cost of all material-handling equipment, shop-aid tooling, fabrication and assembly fixtures, and packaging equipment. Each cost element has been determined for the following heliostat subassemblies: reflector, reflector backing, structure assembly, drive, and foundations. The installation costs include the direct labor and materials, equipment and overhead to assemble, erect, align and checkout heliostat modules. No cost estimates for site grading, field wiring and controls, or system interfaces have been included.

In performing the cost analysis, 1979 dollars are used for a production rate of 75,000 heliostats (300,000 modules) per year (120 million ft²/year). Further, no fee (profit) or G&A (corporate and selling expenses) costs were included, as fee and G&A depend on the type of corporation conducting the business and the marketplace, not on the cost of the particular concentrator design. Typical values of fee and G&A at large scale production range between 6 and 10%.

A detailed production plan and costing for a point-focus concentrator very similar in size and design has been previously generated by

Acurex (Ref. 5-3) for production rates of 100, 1000, 10,000 and 100,000 heliostats per year. This particular point-focus concentrator was designed with sheet-molded compound (SMC) substrates for the reflective glass panels, structural steel spaceframe structure, hydraulic drive, steel support structure and concrete foundation. Specific production plans and costing methodology of this previous study for the SMC reflective panels, drive fabrication and assembly, and field installation have been modified for this particular line-focus concentrator. The costing of the GRC structure was modeled somewhat after production facilities of concrete products. The GRC production plan included the labor and tooling required for bulk material storage, handling and mixing, gunnite molding, mold press, dewatering, steam cure, final assembly and storage. The bulk materials were priced at \$42/yd³ or 1.4¢/lb, which includes cement, glass fiber, sand aggregate and water.

Field erection costs were based upon shipping reflector panels, GRC torque tubes, drive units, and caisson support bearings to the jobsite for final assembly. The installation sequence is a simple one:

- Mount bearings to caisson pedestals
- Bolt GRC torque tubes to bearing flanges
- Install drive unit and connect to adjacent GRC torque tubes
- Mount and attach reflective panels
- Align panels to receiver aperture

A summary of the installed heliostat costs appears in Table 5-7. Costs are shown by subassembly of the heliostat for the categories of direct labor and material, tooling, and equipment and indirect costs. Note that the SMC substrate accounts for about 40% of the total cost. It is believed that a significant reduction in cost can be achieved by substituting GRC for the glass substrate.

Table 5-8 shows additional costs for heliostat foundations that duplicate the above foundation costs and were not used.

Table 5-7

HELIOSTAT PRODUCTION COST SUMMARY, \$/ft² (\$/module)
 (400 ft² aperture/module; 300,000 modules/yr)

Cost Element	Direct Labor	Direct Material	Tooling/ Equipment	Indirect	Total
Reflector	--	0.80 (320)	--	0.03 (12)	0.83 (332)
Reflector backing (SMC)	0.25 (100)	1.26 (504)	0.11 (44)	0.42 (168)	2.04 (816)
Module structure (GRC)	0.05 (20)	0.26 (104)	0.13 (52)	0.06 (24)	0.50 (200)
Drive	0.01 (4)	0.59 (236)	0.01 (4)	0.04 (16)	0.65 (260)
Foundations	0.10 (40)	0.22 (88)	--	0.08 (32)	0.40 (160)
Installation	0.30 (120)	0.03 (12)	0.04 (16)	0.26 (104)	0.63 (252)
Total	0.71 (284)	3.16 (1264)	0.29 (116)	0.89 (356)	5.05 (2020)

5400 Receiver Subsystem

Tower and Bridge Costs--The estimate for the receiver towers and bridges considered the tradeoff between shop or field assembly of the units. It was determined that the first alternative in terms of cost estimating the tower and bridges costs would be to order the steel components fabricated, galvanized and bundled in the shop. The final assembly of the units would then necessarily take place in the field. The costs estimated for this process were used for the plant capital cost of the towers and bridges. They are shown in Table 5-9.

Lower tower and bridges costs can be expected by using steel sections formed from flat stock rather than using the more expensive hot-rolled sections. Additional savings could then be realized by using oxide-protected Corten[®] steel for an additional \$5/hundredweight instead of using galvanizing for an additional \$17/hundredweight. Still further cost reductions can probably be effected by making some sub-assemblies in the shop, thereby reducing many field assembly operations.

Table 5-8

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil	85,210	1,108	495	2,023	3,626
B	Concrete					
C	Structural Steel					
D	Buildings					
E	Machinery & Equipment					
F	Piping					
G	Electrical	159,300	2,071		5,508	7,579
H	Instruments					
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	244,510	3,179	495	7,531	11,205
L	Temporary Construction Facilities					763
M	Construction Services, Supplies & Expense					890
N	Field Staff, Subsistence & Expense					731
P	Craft Benefits, Payroll Burdens & Insurances					2,050
Q	Equipment Rental					763
	INDIRECT FIELD COSTS					5,197
	TOTAL FIELD COSTS					16,402
R	Engineering					820
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					820
	TOTAL FIELD & OFFICE COSTS					17,222
V	Contingency					2,583
W	Fee					743
	TOTAL CONSTRUCTION COST					20,548

CONSTRUCTION COST ESTIMATE,
RECEIVER SUBSYSTEM, TOWERS AND BRIDGES

 

Receiver cost--Estimated capital and installation costs, using a conservative equivalent learning curve of 97%, for the receiver are given in Tables 5-10 and 5-11, respectively. They are based on 550 receiver modules, each 80-ft long, for a total receiver length of 44,000 ft.

Because the receiver is made of 550 identical modules, the repetitive construction and installation will allow mass production techniques. Using a realistic, yet conservative, learning curve of 90% for the first plant, the estimated total installed receiver cost per foot is \$750.47. Using a conservative learning curve of 95% for the 80th plant, the 80th plant installed receiver cost is estimated at \$542/ft.

After changing the working fluid from steam to draw salt during the parametric analysis, insufficient funds were available to change the receiver design from a tube "boiler" to panels of Roll-Bond[®] two-layer expanded-metal construction. Since the molten salt contracts when it freezes and has a very low vapor pressure, much lighter and much lower receiver weight and cost would be possible using this construction. Since the receiver for this system is relatively large, this would offer a significant opportunity to further reduce the BBEC for this system.

5500 Master Control Subsystem--The costing results of the master control subsystem gave a total cost of \$1.385 million. The capital cost of the control room structure with a HVAC system was \$162,000 and the rest of the system was \$1.223 million, as shown below:

<u>MCS Element</u>	<u>Cost</u>
Central computer	\$ 117,000
Block controllers	40,000
Heliostat controllers	610,000
Receiver controllers plus flow valves	273,000
Data paths	<u>183,500</u>
MCS Total	\$1,223,500

Table 5-10

RECEIVER FABRICATION COST ESTIMATE

Shop materials	\$ 6,600,000
Shop labor	12,271,000
Outgoing freight	678,000
Incoming freight	68,000
Insulation, sheeting and pneumatic cylinders	5,206,000
Instrumentation allowance	<u>1,100,000</u>
Total Equipment Cost	\$25,923,000
Design drafting	198,000
Blueprints	10,000
Miscellaneous costs	30,000
Contract control	93,000
Engineering	30,000
Estimating	8,000
Contract reserve	1,000,000
Travel and living expense	20,000
Contract administration	11,000
Quality Assurance and Manufacturing Engineering	<u>50,000</u>
Subtotal Cost	\$27,373,000
GS&A and overhead	1,916,000
Contingency	5,858,000
Fee	<u>2,812,000</u>
Total Cost*	\$37,959,000

* Does not include installation costs.

Table 5-11

RECEIVER INSTALLATION COST ESTIMATE

Receiver installation	\$2,970,000
Temporary construction facilities	150,000
Field staff, subsistence and expense	275,000
Equipment rental	700,000
Construction management	<u>125,000</u>
Total Field and Office Costs	\$4,220,000
Contingency	425,000
GS&A, 7%	325,000
Profit, 8%	<u>375,000</u>
Total Installation Construction Cost	\$5,345,000

The cost breakdown is shown in Table 5-12. The cost of the MCS building is shown in Table 5-13.

5600 Heat Transport Subsystem--The capital cost estimate for the HTS includes all the costs associated with the field piping, heat tracing, insulation, and associated equipment. The costs are based on the equipment lists and material quantity takeoffs. The material unit rates for the piping, insulation, and electrical equipment have been derived from vendor contact or recent Bechtel experience. Labor rates used are in line with recent Bechtel experience, reflecting the repetitive nature and relative complexity of the work. A detailed valve list was derived and vendor quotes were obtained to establish their cost contribution. An allowance has been included for the HTS instrumentation. Table 5-14 shows the cost breakdown for the HTS.

5700 Energy Storage Subsystem--The major capital cost within this subsystem is that of the molten salt storage tanks. These costs have been estimated on the basis of a subcontract installation. Vendor quotes were obtained regarding the current price of the required steel. Table 5-15 shows the cost breakdown for the ESS.

5800 Electrical Power Generating Subsystem--The major capital cost item in this subsystem is that of the turbine generator. The cost of this item was derived from conversations with General Electric. Other equipment costs, such as that for the condenser, feedwater heaters, and diesel generator, were obtained from vendor quotations. The remainder of the equipment was estimated on the basis of recent Bechtel experience.

The turbine building and associated equipment was factored from recent Bechtel projects. This included allowances for the earthwork, concrete, structural steel, piping, electrical, instrumentation, painting, and insulation cost categories. Building area and volume were used for the factoring. Table 5-16 shows the cost breakdown for the EPGS.

Table 5-12

COST FOR MASTER CONTROL SUBSYSTEM

FIELD CONTROLS

Local heliostat controller

\$100 each; 6,000 active units required plus 100 spares = 6,100

\$100 x 6,100 = \$610,000 total

Local receiver controller plus flow valve

Microcomputer, multiplexer, A/D converter, etc.: \$500

Flow control valve: \$500

Active valves, 280

spare valves 3

Total valves 283

283 x \$500 = \$141,500

Active controllers 260

Spare controllers 3

263 x \$500 = \$131,500

TOTAL = \$263,000

Block controllers

Single-board computer with enclosure 10 required at \$3,700 = \$37,000

1 spare computer 3,000

TOTAL = \$40,000

Data communication paths

(This information also given in on-site construction form attached)

Rental of D-8 caterpillar tractor \$ 18,000

650,000 ft of cable at \$0.25/ft 162,500

Plow trench, lay cable, cover trench (labor) 3,000

TOTAL = \$183,500

CENTRAL CONTROL ROOM

Plant operational control minicomputer \$ 42,500

Peripheral control and data acquisition minicomputer 42,500

Operator console and displays 12,000

Array controller (1 active plus 1 spare) 10,000

Pyroheliometer, meteorological input and other misc. 10,000

TOTAL = \$117,000

EMERGENCY CONTROL

Emergency features are included in previous estimates. They are broken down as follows:

Local heliostat \$12,960

Local receiver 3,000

Block 3,000

Central computer 48,000 (allocation)

\$66,960

Note: Not to be added to previous totals; already included.

EQUIPMENT RENTAL, D-8 CAT \$ 18,000

Trench and cable

Subcontractor, \$3,000

Material \$162,500

\$165,000

TOTAL \$183,500

CONSTRUCTION COST ESTIMATE,
MASTER CONTROL SUBSYSTEM BUILDING

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Table 5-14

**CONSTRUCTION COST ESTIMATE,
HEAT TRANSPORT SUBSYSTEM**

A/C NO.	ITEM & DESCRIPTION	MAN. HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil	3,890	51	-	-	51
B	Concrete	56,260	731	-	419	1,150
C	Structural Steel	4,590	60	-	184	244
D	Buildings	-	-	-	-	-
E	Machinery & Equipment	16,950	220	25	5,640	5,885
F	Piping	242,511	3,153	-	6,440	9,593
G	Electrical	573,923	7,461	-	5,906	13,367
H	Instruments	2,543	33	-	282	315
J	Painting	-	-	-	-	-
K	Insulation	-	-	4,201	-	4,201
	DIRECT FIELD COSTS	900,667	11,709	4,226	18,871	34,806
L	Temporary Construction Facilities					2,810
M	Construction Services, Supplies & Expense					3,279
N	Field Staff, Subsistence & Expense					2,693
P	Craft Benefits, Payroll Burdens & Insurances					7,552
Q	Equipment Rental					2,810
	INDIRECT FIELD COSTS					19,144
	TOTAL FIELD COSTS					53,950
R	Engineering					2,698
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					2,698
	TOTAL FIELD & OFFICE COSTS					56,648
V	Contingency					8,495
W	Fee					2,443
	TOTAL CONSTRUCTION COST					67,586

Table 5-15

CONSTRUCTION COST ESTIMATE,
ENERGY STORAGE SUBSYSTEM

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil	6,110	79		66	145
B	Concrete	5,130	67		38	105
C	Structural Steel					
D	Buildings					
E	Machinery & Equipment	3,020	40	5,625	425	6,090
F	Piping	1,958	25		383	408
G	Electrical	7,245	94		55	149
H	Instruments	810	11		73	84
J	Painting					
K	Insulation			486		486
	DIRECT FIELD COSTS	24,273	316	6,111	1,040	7,467
L	Temporary Construction Facilities					76
M	Construction Services, Supplies & Expense					88
N	Field Staff, Subsistence & Expense					73
P	Craft Benefits, Payroll Burdens & Insurances					204
Q	Equipment Rental					76
	INDIRECT FIELD COSTS					517
	TOTAL FIELD COSTS					7,984
R	Engineering					399
	Design & Engineering					
	Home Office Costs					
	R & O					
	TOTAL OFFICE COSTS					399
	TOTAL FIELD & OFFICE COSTS					8,383
V	Contingency 15%					1,257
W	Fee					362
	TOTAL CONSTRUCTION COST					10,002

CONSTRUCTION COST ESTIMATE,
ELECTRICAL POWER GENERATING SUBSYSTEM

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5.4.1.9 Other Qualifications

Within the equipment estimate, a factor of 7% was added to all equipment costs to reflect the cost of freight, vendor representatives, and installation of bulk items. This factor is consistent with Bechtel experience.

5.4.2 Exclusions

The following items were excluded from the project scope:

- Any special construction such as widening and strengthening of existing roads
- Railroads
- Switchyard and power transmission lines beyond the plant high voltage terminals
- Client engineering, financing, and other client costs
- Process royalties and licenses
- Allowance for funds during construction, including interest during construction
- Training plant operators
- Plant operation
- Escalation of costs beyond the price levels of the second quarter in 1979
- Final site landscaping.

5.5 Operation and Maintenance Cost

5.5.1 General

The computer program used to calculate BBEC uses an assumed O&M cost of 1% of the capital investment; this figure is compatible with the value used for the strawman. The materials, supplies, and labor for plant operation and maintenance reflect price and wage levels for the second quarter of 1979. They have been included in the estimate on the basis of the following discussion.

5.5.2 Balance of Plant (EPGS, ESS, HTS)

This estimate was based on a survey of the O&M costs for similar sized power plants throughout the United States. The source of this information was a recently published Federal Power Commission report on plant operating data (Ref. 5-4). The annual costs range from \$1.2 million to \$2.8 million; the median cost is approximately \$1.9 million (1979 dollars). An annual O&M cost figure of \$1.9 million, representing both material and labor expenses, was selected for this study. The annual O&M costs for the ESS and HTS are small compared with that of the EPGS and are included in the \$1.9 million.

5.5.3 Operation and Maintenance Costs

The receiver module is simple in construction and has minimal controls and instrumentation. The only moving parts are the aperture doors and valves. The estimated O&M and cost of 1% of the capital cost is believed realistic. A trolley maintenance vehicle on each row of towers is envisioned to keep receiver maintenance costs low. The low profile of the heliostats will help keep heliostat maintenance costs low. The continual horizontal surface of the single-axis heliostats should provide for cost-effective cleaning of the mirror surfaces. Figure 5-13 shows an artist's concept of a self-propelled heliostat washing machine. The washing fluid and rinse water are recovered with vacuum systems and are continuously filtered for reuse. This minimizes the time required to refill the wash vehicle.

5.6 Conceptual Engineering and Construction Schedule

The engineering and construction schedule is based on the conceptual design engineering and conceptual estimate prepared for the study. The schedule is based on experience gained in the design and construction of fossil-fired plants, adjusted for facility differences.

FIGURE 5-13 HELIOSTAT WASHING MACHINE

5.6.1 Construction Sequence

The construction sequence and methods generally permit procurement and construction to commence before engineering is complete, thus shortening the schedule. The schedule is predicated on the economic use of the manpower included in the conceptual estimate, rather than the earliest possible completion date.

5.6.2 Schedule

Figure 5-14 shows the engineering and construction schedule for the plant and indicates that the project could be completed within 42 months. The governing factor is the ordering, manufacture, delivery and installation of the major equipment.

The major items discussed in the following paragraphs are included in the schedule.

5.6.3 Engineering Services

5.6.3.1 Engineering

Contract award is the point at which siting studies are complete and optimization studies leading to design engineering begin. Design starts with the development of sufficient detail to initiate the procurement of the major equipment. Design is approximately 60% complete when construction begins, and continues for 12 months thereafter. Engineering support is provided throughout construction.

5.6.3.2 Environmental Report

An environmental report is submitted to the appropriate government agency, which must approve the project before the notice to proceed with construction is given to the client. The schedule assumes that the report preparation and approval will not be a critical time constraint.

FIGURE 5-14 ENGINEERING, PROCUREMENT AND CONSTRUCTION SCHEDULE,
SOLAR LINE-FOCUS POWER PLANT

5.6.3.3 Long Leadtime Items

Lead times for the following items were based on current information from potential vendors:

<u>Item</u>	<u>Leadtime (weeks)</u>
Turbine-generator	120
Main condenser	58
Steam generator	145
Salt valves	52
High-pressure feedwater heaters	52

It is assumed that the purchase orders for these items can be issued in a manner consistent with the jobsite requirements.

5.6.4 Field Construction

5.6.4.1 Civil/Structural

The site work can begin as soon as the notice to proceed is issued. The excavation and concrete work will not be a governing factor in scheduling.

5.6.4.2 Mechanical

Installation of mechanical equipment can begin near the completion of civil/structural work. Major equipment will begin to arrive and be installed in the later part of the construction period. The critical elements of the schedule center around the purchasing, delivery and erection of the steam generator and the turbine generator. If, at the time of project inception, the leadtimes of these items are different, the entire project schedule will be affected.

5.6.4.3 Piping, Insulation, Valves and Instrumentation

Prior to erection, it is assumed that the pipe will be pre-fabricated in the field. Once the pipe trenches are completed, the pipe and valve installation will begin. Insulation of the pipe is

assumed to begin as soon as piping systems have been tested and heat tracing installed. The instrumentation will be installed during the later part of the pipe installation work, and will be the last construction work completed.

5.6.4.4 Electrical

Underground cabling will begin as soon as the cable arrives on site. Heat tracing will begin as the piping systems are tested. Electrical work will continue through final plant checkout.

5.6.4.5 Structural Steel

The structural steel required for the towers and bridges is assumed to be ordered at the beginning of the procurement cycle. An onsite shop is utilized to prefabricate as much of the steel as possible. Field labor will install the prefabricated sections which, in the case of the bridges, includes the solar receiver and piping systems.

5.6.4.6 Acceptance Testing

Acceptance testing starts when most of the equipment and bulk materials have been placed and continues until all testing is complete at turbine roll.

5.6.5 Ready for Plant Startup

The plant is ready for startup when all functional testing has been completed and the plant is accepted by the owner for operator startup.

REFERENCES, SECTION 5

- 5-1 Solar Cost Reduction Through Technical Improvement, The Concepts of Learning and Experience, Frank Krawiec and Theresa Flaim, February 1979.
- 5-2 Letter from K. W. Battleson and J. J. Bartel of Sandia Laboratories (Livermore) to E. Lam of Bechtel National, Inc., date 27 June 1979, concerning heliostat and land costs.
- 5-3 Low Cost Point-Focus Solar Concentrator, Phase I Final Report, Acurex Report 79-340, March 1979.
- 5-4 Steam-Electric Plant Construction Cost and Annual Production Expenses, 1976, Twenty-Ninth Annual Supplement, U.S. Department of Energy, August 1978.

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6.0 SUBSYSTEM EXPERIMENT: RECEIVER CRYOGENIC WIND TUNNEL TEST

6.1 Introduction

The heat losses due to wind and natural convection affect the receiver efficiency for the high-temperature line-focus (HTLF) central receiver system being studied. The complex geometry of the receiver precludes analytical prediction of these heat losses. Wind tunnel tests in the University of Illinois Cryogenic Wind Tunnel under the direction of Professor Arthur Clausing were performed to provide data to give insight into the loss mechanisms and means for estimating the losses from the full-size receiver.

Convective heat transfer is governed by two dimensionless parameters: the Reynolds number and the Grashof number. The cryogenic tunnel allows achievement of a high Grashof number, which is the dimensionless parameter governing natural heat convection.

6.2 Test Results

Preliminary review of raw data from the University of Illinois cryogenic wind tunnel experiments indicated a heat transfer coefficient for receiver natural convection that is less than used in the performance calculations. The results for forced convection, however, were larger than that assumed for the prototype plant. Close examination of the raw data revealed several anomalies: (1) The heat transfer coefficients for flow from behind the receiver were equal to those for flow into the face of the receiver. (2) When plotted in a nondimensional form (i.e., Nusselt number versus Reynolds number) the data scatter along the relationship describing externally heated cylinders in cross-flow. (3) The heat transfer coefficients were not constant during the cryogenic transient tests; in fact, the values increased dramatically during the experimental time of interest.

6.3 Analysis

Because of the importance of these tests, and the potential consequences to system performance from large wind losses, SRI reviewed the results to determine if they were reasonable, and if not, to propose explanations for the data trends.

SRI asked Dr. Robert Moffat, Professor of Mechanical Engineering and Director of Thermosciences at Stanford University, to participate in the review. Dr. Moffat is a nationally recognized expert in the field of experimental methods and heat transfer.

It was strongly felt that an internally heated cavity was unlikely to behave like an externally heated cylinder. The method used to determine heat transfer coefficient did not account for losses through the insulation or into the copper-constantan thermocouple wire during the short duration transient tests. Any heat loss would thus result in a higher heat transfer coefficient being predicted. The loss through the insulation was estimated to be on the order of 10%; however, the unsteady heat flow into the copper wire could be quite large.

It was reasoned the presence of nonprototypical axial flow within the receiver model could result in nearly identical heat transfer coefficients between forward and reverse flow conditions. End caps were placed on the model before the experiment to simulate long receiver geometry; however, the caps only covered the diameter of the cavity, and did not extend to include the field of flow to stop axial flow. The 5-cm clearance between the model and the wall of the wind tunnel would provide a flow channel past the model, which could encourage axial flow. For conditions flowing into the receiver cavity, flow would stagnate at the center of the model and move axially until flow would spill into the side streams. For flow conditions into the back of the receiver, the high velocity side streams would create a low-pressure area in front of the receiver cavity. Flow into the midspan of the receiver cavity caused by entrainment of the high-velocity side streams would generate stream paths similar to those of forward flows. Neither condition is prototypical, and could explain the data trends.

It was suggested that of the two sets of data (cryogenic transient tests and room temperature transient tests) the room temperature tests are less likely to exhibit large errors from thermal losses. However, the nonprototypical flow conditions described above would still be present, causing high heat transfer coefficients to be predicted.

It is recommended that further tests be performed using steady-state room temperature flows. Arrangements have also been made to use the Ames Research Laboratory 7-ft by 10-ft wind tunnel for model tests.

6.4 Receiver Water Table Tests

After the results of the cryogenic wind tunnel tests were made available, SRI performed water table flow tests to obtain a visual record of flow paths and relative velocities outside and inside the cavity of the receiver model. Both "upwind" and "downwind" experiments were performed. In addition, baffles were strategically placed to minimize circular flow within the receiver cavity. These panels are rotably mounted and deflected horizontally during tower survival wind conditions. The scale model of the receiver, shown in Figures 6-1 and 6-2, is the same size and shape as the one tested in the cryogenic wind tunnel at the University of Illinois. It has a single eyelid above the cavity. The model thickness is greater in proportion to the receiver diameter than the actual receiver. The free-stream velocity is ~ 6.75 in/s. Velocity in the cavity is ~ 0.45 in/s, about 7% of free-stream velocity. Rotation is counterclockwise. (The two large streaks in the photographs are reflections of the camera tripod legs.)

Figure 6-2 shows the same model with a deflection vane attached below the cavity aperture. Flow traces show the free-stream water at the same velocity (~ 6.75 in/s) whereas the cavity velocity now approaches zero (about 0.1 in/s) or about $1\frac{1}{2}\%$ of free-stream water velocity. Note the movement of the flow stagnation point near the center of the cavity aperture. This modification would probably reduce forced convection losses to an acceptable level.

SINGLE-EYELID MODEL WITH FLOW INTO CAVITY

FIGURE 6-1 WITHOUT DEFLECTOR

FIGURE 6-2 WITH DEFLECTOR

TWO-EYELID MODEL WITH FLOW INTO CAVITY

FIGURE 6-3 WITHOUT DEFLECTOR

FIGURE 6-4 WITH DEFLECTOR

Figures 6-3 and 6-4 show the scale model, closer to scale in thickness, of a two-eyelid design experiencing forward flow. In Figure 6-3, the free-stream water velocity is ~ 6.3 in/s. The cavity velocity is ~ 0.68 in/s, or about 10% of free-stream velocity. Again, rotation is counterclockwise.

In Figure 6-4, a deflection vane is added below the receiver. Flow velocity inside the cavity approaches zero. Free-stream water velocity is ~ 6.75 in/s. Note that the stagnation point is now below the aperture.

Figure 6-5 shows the original model in a reverse flow of 7.2 in/s and a velocity in the cavity of ~ 0.23 in/s, about 3% of free-stream water velocity. When the deflection plate is added (Figure 6-6) with a free-stream water velocity of ~ 9.45 in/s, there is essentially no flow in the cavity.

In Figures 6-7 and 6-8, the two-eyelid model is seen in a reverse water flow of ~ 6.76 in/s. In Figure 6-7 some slight motion is observed in the cavity of about 0.5 in/s, without the lower deflection vane (7% of free-stream velocity).

Finally, in Figure 6-8, when the deflection vane is added in a flow of 6.75 in/s, no movement occurs in the cavity and very little appears in the large space in front of the aperture.

6.5 Conclusions

These water table tests indicate that the velocity in the cavity and the resultant convection heat losses in the line-focus central power receiver may be satisfactorily controlled by the correct aerodynamic design of the receiver and its surroundings. Verification of these results by larger scale wind tunnel tests in air is suggested.

The wind tunnel tests highlighted the importance of the convective heat transfer from the receivers. Analysis of the test equipment, test

SINGLE-EYELID MODEL WITH FLOW BEHIND CAVITY

FIGURE 6-5 WITHOUT DEFLECTOR

FIGURE 6-6 WITH DEFLECTOR

TWO-EYELID MODEL WITH FLOW BEHIND CAVITY

FIGURE 6-7 WITHOUT DEFLECTOR

FIGURE 6-8 WITH DEFLECTOR

procedures and other considerations leads to the belief that the estimates are reasonable. Further, the use of demonstrated air flow control techniques can reduce the forced convection losses even further.

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7.0 MARKET ASSESSMENT OF COMMERCIAL SCALE LINE-FOCUS CENTRAL POWER SYSTEMS

7.1 Summary

The total market for intermediate-load electric generating capacity was estimated to the year 2000 by careful estimation of the requirements of 61 major electric utilities representing approximately 75% of the total generating capacity of the western United States, and an estimated 90% of the total market. The markets to 2000 were derived from analyses of data collected as part of a former SRI study (Ref. 7-1). The markets beyond 2000 were estimated by conservatively assuming power demand growth rates in the range of 2.0 to 2.5% per year in the regions most suited to line-focus central power system market penetration.

Demand in the eastern United States is roughly estimated at 1.5 times that in the western United States. However, almost all of this demand is in regions with low direct normal insolation, averaging 4.5 kWh/m^2 per day or less. As the economics of the HTLF system generally do not favor its use unless it is located in higher insolation regions (5.5 kWh/m^2 per day or above), the eastern United States is not likely to be an important market and no detailed analysis of this potential market was performed.

When a relatively small (100 MW_e) system is considered for base load application, it must be compared with nuclear units or large coal-fired units. Nuclear plants are likely to produce electricity at more favorable rates under all conditions. The large coal units will generally produce electricity less expensively than the line-focus central power units unless the coal escalation rate is several percentage points above the general inflation rate.

The reference system applied to intermediate-load service operates in an economically competitive range in the year 2000 for most of the

western and a very small fraction of the eastern United States, if the cost of coal escalates at an average rate of 8% per year--3% above the assumed general inflation. Depending on whether the initial coal price is closer to \$1.00 or \$1.40/MMBtu, a potential total of 150 to over 350 units is estimated for installation by 2010. These totals will be limited by the reluctance of utilities to commit large fractions of installed capacity to an insolation dependent source. At a lower coal price escalation rate (6%), the system would be likely to capture only a very intermediate-load market. It is estimated that several units, located in the most favorable locations, would be an expected level of penetration by 2010. Significant penetration of eastern United States markets, which are characterized by relatively low solar insolation levels, would require high coal price escalation rates of more than 12% per year (7% per year greater than the assumed general inflation rate).

SRI studies indicate that if production or cost is not restrained by U.S. policy, coal prices will remain constant or change only slightly in constant dollar terms. The lower coal price escalation rates of 6% and 8% are therefore considered much more likely and SRI has concentrated on assessing the markets for intermediate-load units based in the western United States. The assessment indicates a good potential for application at the higher (8%) coal price escalation rate. Table 7-1 provides a summary of the line-focus central power system market penetration results for the western United States under a variety of coal prices assumptions.

7.2 Potential Improvements

The potential improvements to the line-focus solar power plant include performance improvements, capital cost reductions, and economies of scale in the plant size. Because of the system's modularity, additional savings not included would result from incremental startup during construction.

Table 7-1

SUMMARY OF LINE-FOCUS CENTRAL RECEIVER SOLAR THERMAL POWER PLANT
MARKET PENETRATION IN WESTERN U.S. MARKETS
(In Cumulative Numbers of 100 MW Units)

Year	\$1.00/MMBtu Coal Price		\$1.40/MMBtu Coal Price	
	Coal Price Escalation Rate		Coal Price Escalation Rate	
	6%/yr	8% yr	6%/yr	8%/yr
1990	0	0	0	0
1995	0	2	0	12
2000	0	24	0	92
2010	0	162	2-4	362

7.2.1 Performance Improvements

The performance improvements in the EPGS, HTS, and ESS of the 100-MW_e power plant are limited. The net EPGS cycle efficiency could be improved by increasing the throttle and reheat steam temperatures or by decreasing the condenser temperature. However, the 2400 psig/1000°F/1000°F turbine generator that was selected for the conceptual design represents the highest throttle and reheat steam temperatures available in a commercially manufactured turbine. As specified in the requirements definition document, wet cooling towers are used in the heat rejection system. Compared with dry or wet-dry towers, wet cooling towers provide the lowest condenser temperature. As the balance of the EPGS operating parameters were optimized in Tasks 2 and 3, few performance improvements can be seen for the EPGS.

A moderate performance improvement in the HTS could be realized by substituting motor frequency controls for the hydraulic pump and motor variable speed drives of the primary salt pumps. A detailed economic analysis would have to be conducted, however, to determine if the increased efficiency of the frequency control drives offsets their higher capital cost.

As the ESS design is based on the N+1 rather than the thermocline system, few performance improvements can be foreseen for this subsystem.

7.2.2 Capital Cost Reductions

7.2.2.1 Heat Transport Subsystem Cost Reduction

Electric heat tracing, together with its associated switchgear, transformers, and power cable, represents a significant portion of the HTS cost. Heat tracing is primarily intended to preheat the field piping following plant construction or an extended shutdown. It may be possible to preheat the field piping with steam trace heating or hot air supplied by an oil-fired heater. The design and economics of these systems have yet to be addressed.

The capital cost of HTS is somewhat higher than estimated for the parametric analysis. It may be economic, therefore, to increase the number of collector rows from the present 24 per receiver row. This involves a tradeoff between increases in the collector costs and decreases in the field piping and heat tracing costs.

7.2.2.2 Energy Storage Subsystem Cost Reduction

The parametric analysis selected the N+1 storage tank system with a value of N equal to 2. The analysis, however, was based on a carbon steel tank design with internal insulation. The choice of stainless steel for the hot salt tanks in the conceptual design may influence the optimum value for N. A brief economic study could be conducted to determine this value. However, if the value does change, the effect on the plant capital cost is estimated to be small.

7.2.2.3 Receiver Tower Cost Reduction

The receiver towers and bridges could be made of Corten[®] formed sections rather than hot rolled galvanized sections. The receiver wind deflection analysis indicated that the receiver towers are more rigid than required.

7.2.2.4 Receiver Cost Reduction

A potentially large cost reduction for this system lies in the proposed use of Roll-bond[®] heat absorber panels. The present design uses a conventional boiler tube design, which uses thicker walls than necessary for welding. Although this modular tube design lends itself to automation cost reduction for the 550 24.4 m (80 ft) modules for each 100 MW_e plant, the Roll-bond[®] absorber can be made much thinner and lighter. The panels are blanked out in large sheets, automatically seam-welded to a passageway pattern, and expanded to form passageways. This type of construction is made possible by the low vapor pressure of draw salt and by the fact that draw salt contracts when it solidifies.

7.2.2.5 Collector Subsystem Cost Reduction

A potential reduction in the collector foundation labor cost may be realized by using a machine that automatically augers the foundation hole, sets the rebar, and places the concrete. The feasibility of such a machine probably depends on the construction of a number of line-focus plants in similar soil conditions.

As with the point-focus central power system, the greatest opportunity for reduction in cost of the line-focus power system is in the installed heliostat cost. Significant capital cost reduction for the heliostat has been identified based on alternative designs in two areas: substituting glass-reinforced cement (GRC) for the plastic sheet molding compound (SMC) in the reflector backing, and accounting for mutual wind shielding of adjacent rows of concentrators and inclusion of wind fences.

An analysis was conducted to specify a GRC reflector panel backing with optical quality comparable to that of the baseline sheet molding compound design. For identical stiffness (approximately 0.5 mr panel deflection at operating conditions) the sheet molding compound weighs 2.0 lb/ft² and costs \$2.04/ft², and the GRC substrate weighs 4.2 lb/ft² and has a material cost of \$0.15/ft², or about \$0.03/lb. This cost

advantage is somewhat reduced by the need to resize the support structure, bearings and foundations to carry this increased dead load; nevertheless, a significant net cost reduction is anticipated.

The second area of interest is further wind tunnel testing to quantify and document the effects on wind loads of mutual shielding by adjacent single-axis concentrator modules and perimeter wind fences. Martin-Marietta has performed such wind tunnel tests of two-axis tracking heliostats arranged in a typical field layout. The results are shown in Figure 7-1. As indicated, the foundation overturning moments on interior heliostats are reduced to less than one-half those of perimeter heliostats due to shielding. Also, the addition of a wind fence tends to unload the perimeter heliostats as well. The presumption here is that wind fences cost much less than the heliostats because they have no optical constraints, but must simply deflect the wind, resulting in a net cost reduction. With reduced wind loads the reflector panels, structure, drive and foundations of the line-focus heliostats can all be downsized.

FIGURE 7-1 TWO-AXIS HELIOSTAT REACTION LOADS

The cost comparison indicates that a nearly 20% net cost reduction in installed heliostat capital cost is likely due to the GRC reflector panel substitution and load reduction due to shielding effects. The reduction in the heliostat capital costs would reduce the calculated BBEC from 61 mil/kWh to about 56 mil/kWh for the 80th plant.

7.3 Economies of Scale

As outlined in the parametric analysis, the costs of the subsystems (with the exception of the MCS) and draw salt are linear functions of the plant size. The ESS storage tank and EPGS costs each have cost-scaling factors of less than 1.0. Since turbine efficiency improves with turbine size, plant sizes larger than 100 MW_e may have a performance and economic advantage. The incentive to design a larger plant, however, must be weighed against the operational problems of the plant. These operational concerns include salt freeze protection, uniform distribution of the salt flow, and reliability of the collector drives and receiver modules.

The performance of the EPGS in the conceptual design is based on a reheat turbine-generator. Reheat machines are commercially available only in capacities greater than 75 MW_e. In general, turbines with reheat have a better efficiency than turbines without reheat. Therefore, there is less incentive, in terms of performance, in exploring plant capacities below 75 MW_e for systems using steam. Smaller sizes (1-10 MW_e) should be explored, using Brayton cycle turbines and Rankine cycle turbines with organic working fluids, e.g., pyridene and water.

7.4 Market Analysis

7.4.1 Introduction

The commercialization of new systems is expedited if the market requirements for these systems are understood early in their design and development. In the case of line-focus central power system central power units, it is helpful to understand the potential size, or size range, of useful systems and the insolation conditions under which they

might prove competitive with other electric power producers. This information can guide the designer and enable him to select unit designs with greater commercial potential. Furthermore, it is necessary to estimate the total market potential for all competitive systems and the likely share of that total the system could obtain in order to estimate realistically the manufacturing requirements and costs. Realistic cost estimates for competitive systems lead to realistic estimates of their market shares. Finally, it is desirable to estimate the rate of market penetration (sales) to be expected. This latter parameter dictates the speed at which the systems are sold and new manufacturing facilities are needed. This estimate of sales prospects is also useful in establishing cash flow expectations and financial feasibility of new ventures.

There are additional reasons for conducting market analyses that indicate market size and share. Predictions of the potential use of solar electric generating systems and of the fossil fuel savings they make possible are useful to economists and government planners who are attempting to forecast the need for fossil fuels and the productive structure needed to supply them.

The market analysis reported here consists of estimates of overall market size derived from projections of electric power growth, examination of utility plans, and projections of potential governmental (regulatory) action (see Section 7.3.4). Market share is projected by comparisons of the levelized costs of busbar power produced by the line-focus central power units with costs of other electric power producers such as coal and oil-fired units (see Section 7.3.3). In these comparisons, standard economic and performance assumptions were applied to all plants (see Section 7.3.2).

Projections of market penetration depend on evaluation of utility attitudes toward new technologies and of potential environmental and other constraints to acceptance of line-focus central power systems.

7.4.2 Solar and Fossil Plant Financial, Economic and Performance Assumptions

Comparisons between units with differing ratios of capital to operating and fuel costs are frequently highly sensitive to the economic, financial, and performance assumptions made. Comparisons between fuel types have similar sensitivity. The influence of these assumptions is particularly strong in this instance, since the comparison is based on plants intended to go into operation after 1990 and to operate 30 years thereafter. The financial parameters set forth in Table 7-2 were suggested by DOE.

The cost values set forth in Table 7-3 were derived by SRI from a variety of sources. Primary reliance was placed on data found in the Technical Assessment Guide prepared by the Electric Power Research Institute (Ref. 7-2).

Table 7-2

FINANCIAL PARAMETERS USED IN LEVELIZED BUSBAR COST ANALYSIS

Debt fraction	0.55
Return on debt	0.09
Stock fraction	
Common	0.27
Preferred	0.18
Return on stock	
Common	0.15
Preferred	0.11
Income tax rate	0.50
Annual insurance/other taxes	0.0225
Investment tax credit	0.10
Depreciation method	Straight line
Depreciation life, years	30
Rate of general inflation	0.05
Capital escalation rate	0.065
O&M escalation rate	0.065

Table 7-3

ECONOMIC AND PERFORMANCE ASSUMPTIONS FOR HTLF
AND COMPETING POWER PLANTS

Base year for costs	1978
Year of commercial availability	1990
Plant life, years	30
Construction half completed (for 1990 startup) in:	
HTLF	1988
Coal, intermediate load (400)*	1987
Oil, resid-steam, intermediate load (400)	1988
Oil, resid, combined-cycle, intermediate load (250)	1988
Capital cost, \$/kW _e	
HTLF, 1st, 40% CF	1,492
HTLF, 80th, 40% CF	1,165
Coal, intermediate load (400)	750
Oil, resid-steam, intermediate load (400)	440
Oil, resid, combined-cycle, intermediate load (250)	300
Heat rate, Btu/kWh	
Coal, intermediate load (400)	11,000
Oil, resid-steam, intermediate load (400)	9,500
Oil, resid, combined-cycle, intermediate load (250)	8,500
Fuel costs, \$/MM Btu	
Coal	1.00/1.40
Oil, resid	2.65
Fuel price escalation, percent/year	
Coal	6,8
Oil	8
O&M cost, mil/kWh (first year)	
HTLF	1% of capital
Coal, intermediate load (400)	3.5
Oil, resid-steam, intermediate load (400)	1.0
Oil, resid, combined-cycle, intermediate load (250)	2.0

*Unit size, MW.

Source: EPRI and SRI.

The line-focus central power system capital costs are assumed to vary as a function of the number of plants constructed. Learning effects would be expected to allow for decreasing capital costs, and as shown in Table 7-3, a decrease of 22% has been assumed to occur between the 1st and 80th plants constructed. The A&E firm of Townsend and Bottum Inc. has observed cost savings of 13% for the fourth 800-MW coal-fired power plant (Ref. 7.3). In our market penetration analysis, we assumed a smooth transition from 1st to 80th plant costs takes place as a function of time according to the following equation:

$$\text{Cost}(t) = \text{Cost}(1st) \left[\text{TCL} + (1 - \text{TCL})e^{-t \cdot \text{TCR}} \right]$$

where

- Cost(t) = capital cost at time t
(t is measured in years following year of commercial availability)
- Cost(1st) = capital cost of the 1st plant.
- TCL = ratio of the ultimately achievable (lowest) capital cost, to the 1st plant cost.
- TCR = rate at which the ultimately achievable cost is approached.

Delivered long-term contract coal costs listed for 1978 generally ranged from \$1.00 to \$1.40 per million Btu. The lowest cost is representative of regions characterized by competitive markets, mine-mouth western coal, and/or captive mines. The higher cost is representative of coals shipped long distances or without competitive market. Actual coal prices can vary widely. SRI has observed early 1979 prices for delivered coal (long-term contract) ranging from \$1 to \$2 per million Btu. The values of \$1.00 and \$1.40 per million Btu were used in the final analysis of market share and market penetration.

A number of different methods of cost calculation are available for this type of analysis. First-year costs, average cost of service, and levelized costs, which are discounted costs averaged over time, are frequently used. The choice made here can also influence the competitive

status of the alternate systems under evaluation. Levelized costs, computed by the methodology set forth by Doane (Ref. 7-4), were used in the calculations and comparisons of Section 2.3.

All plants are assumed to start up in 1990, and all results are expressed in 1978 dollars.

7.4.3 Comparison with Fossil Plants

Solar power systems require storage and high capital cost for base load operation. This indicates that solar systems are best suited to intermediate-load service. Some consideration was given to the economics of a 60% capacity factor line-focus central power system with appropriate storage capacity. This unit could not compete with nuclear units nor large base load coal units until coal prices rise far above current levels. Within the intermediate-load power market, the stiffest competition is likely to be from steam-cycle coal plants ranging up to 400 MW in capacity, and somewhat smaller (about 250 MW) combined-cycle, oil-fired plants. To be consistent within this analysis, these plants are assumed to operate at a "typical" intermediate-load capacity of 40%. The important economic, financial, and performance assumptions used to characterize the line-focus central power system and competing power plants were presented in Section 7.3.2.

Using these assumptions within the framework of the costing methodology that was noted in Section 7.3.2, levelized busbar electricity cost estimates were computed for the various plant types considered.

Since the cost of power from a solar-thermal electric facility is highly dependent upon the availability of direct normal solar insolation, regional variations in this parameter were considered in performing the cost calculations. Characteristic insolation levels were developed for the various regions used in this study. As shown in Figure 7-2, direct normal insolation ranged from a high of $7.5 \text{ kWh/m}^2 \text{ day}$ (annual average) in limited portions of the southwestern U.S. to $4.5 \text{ kWh/m}^2 \text{ day}$ or less in northwestern, midwestern, and southern portions of the country. Variations in the cost of coal were also considered in this analysis;

FIGURE 7-2 U.S. SOLAR INSOLATION REGIONS (DIRECT NORMAL INSOLATION IN kWh/m² day)

delivered coal costs of \$1.00 and \$1.40 per million Btu (1978 dollars) were both used in determining the costs.

The results of the cost analysis that included variations in insolation and coal cost are presented in Table 7-4. These are levelized busbar costs, expressed in 1978 dollars, for plants that start in 1990. The annual average coal price escalation rates of 6 and 8% are higher than SRI's long-range forecasts (to the year 2000), of no real (constant dollar) rise in the price of coal.* At higher coal prices, the economic attractiveness of the line-focus central power plant is increased in relation to the conventional plants.

In the base load market, the availability of nuclear power and larger coal-fired plants substantially reduces the ability of a line-focus central power system to obtain a significant market share. Both of these systems are estimated to generate base load electricity at lower cost than options considered.

7.4.4 Market Assessment

7.4.4.1 Methods of Analysis

Markets for solar electric generating units are determined by three primary considerations. First, they are limited by the expected growth in demand for new electric generating units of all kinds. The definition of this expected market was an important part of the market assessment. The second important factor in the assessment of solar markets is the economic competitiveness of this system with all systems that could be used to produce electric power in the market period. This comparison, shown in Section 7.3.3, is used to compute an ultimate or equilibrium share of the total demand for capacity that should be captured by the line-focus central systems. Finally, markets (sales) in the near term are limited by the rate at which customers (electric utilities) accept a new technology (product).

* These escalation assumptions have a considerable impact on the conclusions drawn. The competitive ability of the line-focus central power system is strongly influenced by coal price assumptions.

Table 7-4

COMPARISON OF LEVELIZED BUSBAR ELECTRICITY COST PROJECTIONS

Intermediate Load (40% Capacity Factor)	Fuel Price Escalation (%/yr)	Plant Capacity (MWe)	Levelized Busbar Electricity Costs (mil/kWh, 1978 dollar basis)							
			\$1.00/MMBtu Coal Price				\$1.40/MMBtu Coal Price			
			Solar Insolation (kWh/m ² yr)				Solar Insolation (kWh/m ² yr)			
			4.5	5.5	6.5	7.5	4.5	5.5	6.5	7.5
HTLF (1st plant cost)	-	100	176	148	129	115	176	148	129	115
HTLF (80th plant cost)	-	100	137	116	101	90	137	116	101	90
Coal-steam	6	400	75	75	75	75	84	84	84	84
Coal-steam	8	400	90	90	90	90	106	106	106	106
Oil-steam	8	400	118	118	118	118	118	118	118	118
Oil combined cycle	8	250	103	103	103	103	103	103	103	103

Bases: 1990 startup for all plants; oil (resid) cost is \$2.65/MMBtu in 1978

The following paragraphs describe the definition of overall market and market share under varying competitive situations. Estimated overall markets are determined by examining projected demand (sales) of electricity and computing the electric generating capacity needed to meet this demand. The calculations required are:

- Regional projections of electric power demand
- Allocation of this demand to the individual states
- Allocation of state demand among the major utilities (primary power producers)
- Calculation of capacity requirements to meet demand for these utilities (representing approximately 80% of total area capacity)
- Estimates of markets represented by all utilities in the region
- Estimation of the ultimate market share.

The regional demand projections were based on previous SRI projections of regional markets for electricity (Ref. 7-1). These projections were derived from a detailed and regionalized computer analysis of energy supply and demand in the United States and the price competition that determines the choice between fuels (or between fuels and electricity).

The analysis emphasized those fuels used in electricity production and those other fuels in competition with electricity. The nationwide electricity growth was projected at 5.3% for 1975-1985, and 3.8% for 1985-2000. This latter period is of greatest interest for this study; however, a longer time frame (2000-2010) was also considered. Over this period, lower growth rates in the range of 2.0 to 2.5% were assumed by SRI for electricity growth in the regions comprising the most important markets for the line-focus central power system.

The effect of differences in growth rates is important to the market projections. If, instead of 5.3 and 3.8% annual growth rates for 1975-85 and 1986-2000, the rates were 4.8 and 3.3%, the gross market would drop 17%. If the rates were to drop to 4.3 and 2.8%, the markets would be reduced 30%. Increased electricity sales would show equivalent increases in capacity demand and markets for new units. Thus, while the projected

markets are based on reasonable estimates of growth in electric power demand, the actual markets could vary substantially from those projected on the basis of 5.3 and 4.8% annual growth.

The forecast demand (sales) in the west north central, west south central, mountain, and pacific regions was allocated to individual states. Reported sales for 1976 were used as a base. Trends were deduced by examining 1970 and 1973 (Ref. 7-5). Line losses (7%) were added to the state sales to obtain generation load requirements. Average capacity factors were estimated for each state. These factors included the reserve margins actually maintained by the utility. These factors, as for the state-by-state distribution of regional sales, were based on 1976 data and projected forward using recent trends as guidance (Ref. 7-6). The projection assumed the capacity factors would improve with the installation of modern equipment selected with the idea of obtaining improved on-line availability and performance, which is a major utility industry concern.

The overall generation allocation for each state was divided into requirements for base, intermediate, and peak load service. By dividing the hours of use for each load type into the proportion of generating capacity, the total capacity required to satisfy the load was derived. The average allocation of capacity was: base, 50%; intermediate, 31%; and peak, 19%. These hypothetical allocations can only be used as rough guides. A utility will operate its units as base, intermediate, or peak load depending on need, unit capability, and the direct cost of power. The lowest cost generation unit (or mix of units) will be preferred by the dispatcher.

The study was extended to the major utilities in each state examined. Again, 1976 was the base year, and trends from 1970 were considered in the projection of the allocations of the state totals (Ref. 7-6). Adjustments to sales were necessary for those utilities with sales in more than one state. Also, the individual utility sales were adjusted for interchange. The adjusted sales figures used were for sales within the service areas. Sales to municipally owned organizations were

included in the sales base. Entitlements (i.e., sales by governmental organizations to preferred customers) were included in available peak capacity (Ref. 7-6). Correction for average line loss experienced by each utility was applied to sales to calculate capacity requirements.

Capacity requirements for the individual utilities were projected using the projected sales corrected for system line loss, observed trends in system capacity, and the allocation of capacity (i.e., 50% base, 31% intermediate, and 19% peak).

Existing capacity by state and utility was obtained from DOE (Ref. 7-7), EEI (Ref. 7-8), and individual utility data (Ref. 7-9). This was corrected for each category--base, intermediate, and peak for:

- Announced additions (+)
- Expected retirements (after 30 years) (-)
- Expected transfers from base (-,+) to intermediate (units <400 MW and >15 years old)
- Entitlements (+).

Announced additions include those through January 1979. They were obtained from DOE, trade journals (Ref. 7-10), and various other utility reports. Jointly owned capacity was allocated to the individual owners and to the state of ownership to be matched against electric power demand in that state.

As indicated, SRI assumes that base load units would be transferred to intermediate service after 15 years. However, units with capacities above 400 MW are expected to remain in base load service. All plants are expected to be retired after 30 years of service. While these assumptions are in general accord with electric utility practice, retirements and shifts in service function can occur earlier or later than predicted. If individual utility operations indicated a surplus of base load and a deficit of intermediate load, a frequent occurrence, the few suitable base load fossil units would be switched to intermediate power service. Such factors as siting or other regulatory delays, or difficulty in attracting capital funds at acceptable rates of interest, could cause the utility to retain plants in service beyond 15 or 30 years.

Borrowing or pooling of electricity or even reduction of reserve margin may be used to defer ordering of new or replacement equipment.

The general thrust of such practice is to delay the ordering of new plants. With a given growth over time, the need for new generating equipment between 1980 and 2000 will remain constant. By delaying orders until 1990, for example, the utility concentrates the market in 1990-2000. This delay is advantageous to systems such as the HTLF that will not be demonstrated until the mid- to late-1980s. That effect is indicated in Table 7-5.

For each utility and group of utilities (state and power pool), the forecast of capacity need (column 6 of Table 7-5) was compared with capacity calculated in 1986, 1989, and 2001 (column 5). If calculated available capacity exceeded forecast capacity, no additional generating units were required and there was no market. If projected capacity was not sufficient to meet the forecast demand, then new capacity was assumed to be ordered before the end of the period, i.e., before 1986, 1989, and 2001 (column 7). As a final approximation, capacity ordered in 1986 and 1989 was added to the capacity calculated to be available in the succeeding period to arrive at a total potential capacity market (column 8). Also shown in Table 7-5 are the adjustments for announced additions, retirements, and transfers.

The requirements stated are for installed capacity; orders would be placed four to five years earlier. Thus, the markets indicated for 1987-1989 are not likely to be available to HTLF systems unless the utility ordered before demonstration. The information was included to indicate the incentive (potential increase in sales of solar units) that could result from an accelerated HTLF development program.

It was not feasible within the constraints of this project to analyze the hundreds of utilities in the western states separately; however, major utilities were analyzed for each state considered. In the analysis, major utility totals were accumulated by state and power pool or coordinating council. (Pooling of electric generation within states and within power pools is a normal mode of utility operation.)

Table 7-5

SAMPLE CALCULATION, GW
(Base Load Markets)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
	Actions, 1978-1986			Projected* Capacity 1986	Need* 1986	Buy Available Units (if deferred) [†]	HTLF Market
<u>Actual capacity, 1977</u>	<u>Announced Additions</u>	<u>Normal Retirement[‡]</u>	<u>Transfer to Intermediate[‡]</u>				
16.7	6.3	1	2	20.0	26.1	6.1	Up to 6.1
<u>Expected capacity, 1986</u>	Actions, 1987-1989			29.1	29.5	0.4 (6.5) [†]	Up to 6.5
26.1	3.7	0.7	0				
<u>Expected capacity, 1989</u>	Actions, 1990-2001			30.9	44.2	13.3 (19.8) [†]	13.3 to 19.8
29.5	2.6	1.2	0				

* Annual growth rate of overall demand to 1985 at 5.3%, from 1986-2001 at 3.8%.

[†] Quantity to buy if previous requirements were not filled by earlier purchase.

[‡] Example quantities.

7.4.4.2 Estimated Markets for New Generating Capacity

The utilities examined in detail are set forth in Table 7-6. As shown in the table, the utilities were categorized by utility coordinating councils for later analysis. The utilities listed own 72 to 91% of the capacity in the states considered.

The analysis concentrated on intermediate power plant requirements, since these are the most likely markets for line-focus central power systems. However, estimates of baseload capacity needs were also developed. The projected demands for electric generating capacity in the intermediate and baseload markets are set out in Tables 7-7 and 7-8. In these tables, the capacity needed was based on the following assumptions:

- Generated power is shared by all utilities within a state (state needs are summed)
- Generated power is shared by all utilities within a power pool or reliability council (pool needs are summed).

If perfect pooling is assumed, the total demand for new installations of generating units is estimated at 1.7 GW of base units and 14.6 GW of intermediate load units in 1987-89. In this period, a marked surplus of base load units can be found in three of the four power pools considered. However, it is unlikely that the intermediate power demand will be filled by derating additional units.

The single exception is in the Western Systems Coordinating Council (WSCC). The U.S. portion of this power pool will have a surplus base load unit capacity in 1986 and 1989 (21.5 and 23.7 GW, respectively). The Bonneville Power Administration (BPA) manages power produced by several government agencies. If BPA could change its supply contracts, some of the BPA-managed hydroelectric capacity of 13.2 GW could be shifted to cover intermediate power demands. This would relieve at least some of the estimated deficits of 9.7 and 16.5 GW in 1986 and 1989, respectively. This would have the effect of deferring purchase of units until sometime after 1990 and of increasing the intermediate power unit market in the WSCC region from 26.6 to perhaps as much as 35 GW for 1990-2001.

Table 7-6

UTILITIES EXAMINED FOR CAPACITY REQUIREMENT, 1977
(1977 Capacity in Thousands of Megawatts)

<u>Western Systems Coordinating Council</u>		<u>Total State Capacity (estimated)</u>	<u>Total Capacity Reported by Utilities Listed</u>	<u>Percentage* of State Capacity in Utilities Examined</u>
Washington	Seattle Dept. Lighting Washington Water Power Washington Public Power Supply System BPA* to Washington public power agencies Total	18.4	10.6	
Oregon	Pacific Power and Light (excludes Wyoming) Portland General Electric Puget Sound Power and Light BPA† to Oregon public power agencies Total	7.9	10.5	
California	Los Angeles Dept. Water and Power Pacific Gas and Electric Co. San Diego Gas and Electric Co. Southern California Edison Co. Sacramento Municipal District Total	35.7	32.0	
Nevada	Nevada Power Co.* Sierra Pacific Power Co.* Total	3.6 ¹	2.0	
Utah	Utah Power and Light Co.*	1.6	2.3 ²	
Arizona	Arizona Public Service Co.* Tucson Gas and Electric Co. Salt River Project Total	8.7	6.8	
Colorado	Public Service Co. of Colorado*	4.7 ³	2.6	
New Mexico	Public Service Co. of New Mexico	4.5 ⁴	0.9	
Montana	Montana Power Co.	3.1 ⁵	1.1	
Idaho	Idaho Power Co.*	1.8	1.8 ⁶	
Wyoming	Pacific Power and Light Co.,* Wyoming portion only Total of above states and utilities	<u>3.3⁵</u> 93.3	<u>1.8</u> 72.4	77.6%
<u>Electric Reliability Council of Texas</u>				
Texas	Central Power and Light Co. (Central and Southwest Corp.) Community Public Service Co. Dallas Power and Light Co. (Texas Utilities) El Paso Electric Co.* Gulf States Utilities* Houston Lighting and Power Co. San Antonio Public Service Board			

Table 7-6 (continued)

<u>Electric Reliability Council of Texas (continued)</u>		<u>Total State Capacity (estimated)</u>	<u>Total Capacity Reported by Utilities Listed</u>	<u>Percentage* of State Capacity in Utilities Examined</u>
	Southwestern Public Service Co. Texas Electric Service Co. (Texas Utilities) Texas Power and Light Co. (Texas Utilities) West Texas Utilities (Central and Southwest)			
	Total for above states and utilities	46.0	42.0	91.3%
<u>Mid-Atlantic Area Council</u>				
Dakota	Montano-Dakota Utility Co.* Otter Tail Power Co.*			
	Total	2.1 ⁵	0.9	
Dakota	Black Hills Power and Light Co.* Northwestern Public Service Co.			
	Total	2.2 ⁵	0.4	
Minnesota	Minnesota Power and Light Co.* Northern States Power Co.*			
	Total	8.2	7.1	
Nebraska	Nebraska Public Power District Omaha Public Power District			
	Total	3.9	2.9	
Iowa	Interstate Power Co.* Iowa Electric Light and Power Iowa-Illinois Gas and Electric Iowa Power and Light Co. Iowa Public Service Co. Iowa Southern Utilities			
	Total	6.2	5.0	
	Total of above state and utilities	22.6	16.3	72.1%
<u>Southwest Power Pool</u>				
Kansas	Kansas Gas and Electric Co. Kansas Power and Light Co.			
	Total	6.8 ⁷	3.5	
Oklahoma	Oklahoma Gas and Electric Co. Public Service Co. of Oklahoma (Central and Southwest Corp.)			
	Total	9.2	6.9	

Table 7-6 (concluded)

<u>Southwest Power Pool (concluded)</u>		<u>Total State Capacity (estimated)</u>	<u>Total Capacity Reported by Utilities Listed</u>	<u>Percentage* of State Capacity in Utilities Examined</u>
Missouri	Expire District Electric Co. Kansas City Power and Light Co. Missouri Public Service Co. Union Electric Co.			
	Total	13.4	11.7	
Arkansas	Arkansas Power and Light Co. (Middle South Utilities)	4.8	3.3	
Louisiana	Central Louisiana Electric Co. Inc. (Middle South Utilities) Louisiana Power and Light Co. (Middle South Utilities) New Orleans Public Service Co. (Middle South Utilities) Southwestern Electric Power Co.* (Central and Southwest Corp.)			
	Total			
	Total of above states and utilities	<u>12.9</u> 47.1	<u>9.6</u> 35.0	74.3%

* Includes municipals, public power districts, rural electric cooperatives, and wholesale deliveries to large industrial companies.

† Utility capacity is frequently located among several states.

* Service territory extends to other states.

Notes to Table 7-6

¹ Large amounts of power are owned by the federal government and out-of-state utilities.

² Utility also supplies Wyoming and Idaho.

³ Capacity includes a large number of federal, municipal, and cooperative installations.

⁴ The majority of capacity is owned by utilities that have been listed within the WSCC coordinating council.

⁵ State capacity includes equipment owned by utilities in adjacent states. Those utilities have been included in this study.

⁶ Utility also supplies Nevada and Oregon.

⁷ The state capacity includes various small municipal and cooperative installations.

Table 7-7

PROJECTED ADDITIONAL BASE LOAD CAPACITY REQUIREMENTS AND POTENTIAL
MARKETS FOR GENERATING EQUIPMENT, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

	Additional Needed Capacity			Markets	
	1986	1989	2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>					
Arizona	S	S	S	0	0
California	6.1	6.5	19.8	0.4	13.3
Colorado	S	S	0.6	0	0.6
Idaho	S	S	S	0	0
Montana	0	0.1	0.9	0.1	0.8
Nevada	S	S	S	0	0
New Mexico	S	S	S	0	0
Oregon	S	S	S	0	0
Utah	S	S	S	0	0
Washington	S	S	S	0	0
Wyoming	S	S	S	0	0
Pooled	S	S	5.0	0	5.0
Sum of States	6.1	6.1	21.3	0.5	14.7
<u>Electric Reliability Council of Texas</u>					
Texas	S	1.7	27.7	1.7	26.0
Pooled	S	1.7	27.7	1.7	26.0
<u>Mid-Atlantic Area Council</u>					
Iowa	0.1	S	0.5	0	0.5
Minnesota	S	S	S	0	0
Nebraska	S	S	S	0	0
North Dakota	0.3	0.3	0.9	0	0.6
South Dakota	0.1	0.2	0.4	0.1	0.2
Pooled	S	S	S	0	0
Sum of States	0.5	0.5	1.8	0.1*	1.3
<u>Southwest Power Pool</u>					
Arkansas	S	S	2.7	0	2.7
Kansas	S	S	S	0	0
Louisiana	S	S	6.8	0	6.8
Missouri	S	S	0.2	0	0.2
Oklahoma	S	S	1.3	0	1.3
Pooled	S	S	10.8	0	10.8
Sum of States	S	S	21.8	0	11.0
Total Western United States (Pooled)				1.7	41.8
Total Western United States (Individual States)				2.3	53.0

S = Surplus, no additional units needed.

*Difference caused by change from deficit to surplus in Iowa between 1986-1989.

Table 7-8

PROJECTED ADDITIONAL INTERMEDIATE LEVEL CAPACITY REQUIREMENTS
AND POTENTIAL MARKETS, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

	Additional Needed Capacity			Markets	
	1986	1989	2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>					
Arizona	S	0	2.5	0	2.5
California	3.2	6.6	21.9	3.4	15.3
Colorado	S	S	1.1	0	1.1
Idaho	S	S	S	0	0
Montana	S	S	S	0	0
Nevada	S	S	0.7	0	0.7
New Mexico	0.3	0.4	0.5	0.1	0.1
Oregon	3.3	4.8	7.6	1.5	2.8
Utah	0.2	0.4	1.3	0.2	0.9
Washington	5.0	5.5	8.1	0.5	2.6
Wyoming	S	S	S	0	0
Pooled	9.7	16.5	43.1	7.8	26.6
Sum of States	12.0	17.7	43.7	5.7	26.0
<u>Electric Reliability Council of Texas</u>					
Texas	12.7	16.6	34.2	3.9	17.6
Pooled	12.7	16.6	34.2	3.9	17.6
<u>Mid-Atlantic Area Council</u>					
Iowa	1.6	1.6	2.6	>0.1*	1.0
Minnesota	1.8	2.2	3.5	0.5*	1.2*
Nebraska	0.6	0.7	1.9	0.1	1.1*
North Dakota	0.2	0.3	0.8	0.1	0.4*
South Dakota	0.2	0.2	0.4	>0.1*	0.2
Pooled	4.3*	5.1*	9.1*	0.8*	4.0
Sum of States	4.4*	5.0*	9.2*	0.7*	4.0
<u>Southwest Power Pool</u>					
Arkansas	1.5	2.0	3.9	0.5	1.9
Kansas	0.3	0.3	1.0	0	0.7
Louisiana	2.2	3.5	8.5	1.3	5.0
Missouri	3.0	3.4	5.5	0.4	2.1
Oklahoma	1.7	2.6	6.1	0.9	3.5
Pooled	8.8*	11.8	25.0	3.1	13.2
Sum of States	8.7*	11.8	25.0	3.1	13.2
Total Western United States (Pooled)				14.6	61.4
Total Western United States (Individual States)				12.9	60.8

*Differences due to rounding.

S = Surplus, no additional units needed.

Perfect pooling is unlikely. An approximation of the effect of imperfect pooling was obtained by considering state pooling rather than regional pooling. The effect is marked but not overwhelming. The potential market increase is 11.8 GW for baseload capacity and 2.8 GW for intermediate load capacity. Like deferred retirement of generating units or the purchase of power from others, pooling has the effect of deferring purchases by the individual utilities. The more the pooling, the later the demand for new capacity arises. Several regions currently have substantial planned surpluses, especially for baseload units, because large baseload units must be planned or ordered well before they are needed (up to 15 years for a large nuclear unit); until 1973-74, utilities were operating on historical growth rates of 6 to 8%. As noted earlier, SRI demand forecasts assume 5.3% annual growth to 1985 and 3.8% until 2000.

The difference in 1990 capacity requirements under an 8% growth assumption and the SRI lower growths is approximately 50%.

Given the assumptions above, with no deferral of equipment purchase, the projected need for installed additional electric generating capacity in 1987-89 and 1990-2001 is shown in Table 7-9.*

Additional generating capacity would be needed if all oil and gas (except) peaking units were retired. Such a retirement could call for repowering with line-focus central power heating units, replacing the oil and gas firing equipment and steam boilers, but using the existing turbogenerators. In other (perhaps most) cases, complete new generating units would be required. The effect of such early retirement is shown in Tables 7-10 through 7-14. Tables 7-10 through 7-13 consider base and intermediate load requirements separately and for two time periods for retirement (before 1986 and after 1990). The overall effect is summarized in Table 7-14. Early retirement of existing plants will

*Some of the near-term, recorded capacity surpluses are diminishing as utilities defer construction because of cash flow and capital market pressures. Such effects on the market are likely to be felt before 1990.

Table 7-9

SUMMARY OF MARKETS FOR NEW GENERATING CAPACITY
WESTERN UNITED STATES, GW
(Normal Retirement)

	1987-1989		1990-2001	
	Base	Intermediate	Base	Intermediate
<u>Western States Coordinating Council</u>				
By state	0.5	5.7	14.7	26.0
By pool	0	7.8	5.0	26.6
<u>Electric Reliability Council of Texas</u>				
	1.7	3.9	26.0	17.6
<u>Mid-Atlantic Area Council</u>				
By state	0.1*	0.7*	1.3	4.0
By pool	0*	0.8*	0	4.0
<u>Southwest Power Pool</u>				
By state	0	3.1	11.0	13.2
By pool	0	3.1	10.8	13.2
Total by states	2.3	13.4	53.0	60.8
Total by pool	1.7	15.6	41.8	61.4

* Differences because of rounding.

Table 7-10

COMPARISON OF NORMAL AND EXPANDED MARKETS DUE TO RETIREMENT
OF ALL OIL AND GAS BEFORE 1986 FOR BASE LOAD
GENERATING EQUIPMENT, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

		Markets					
		Normal		Expanded		Difference	
		1987-1989	1990-2001	1987-1989	1990-2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>							
Pooled		0	5.0	0	10.9	0	5.9
Sum of states		0.5	14.7	0.1	13.8	0.4	-0.9
<u>Electric Reliability Council of Texas</u>							
Pooled		1.7	26.0	2.3	19.4	0.6	-6.6
<u>Mid-Atlantic Area Council</u>							
Pooled		0	0	0	0	0	0
Sum of states		0.1	1.3	0.2	1.5	0.1	0.2
<u>Southwest Power Pool</u>							
Pooled		0	10.8	0.7	14.9	0.7	4.1
Sum of states		0	11.0	1.8	10.8	1.8	-0.2
Total western U.S. (pooled)		1.7	41.8	3.0	45.2	1.3	3.4
Total western U.S. (individual states)		2.3	53.0	4.4	45.5	2.1	-7.5

Table 7-11

COMPARISON OF NORMAL AND EXPANDED MARKETS DUE TO RETIREMENT
OF REMAINING OIL AND GAS AFTER 1990 FOR BASE LOAD
GENERATING EQUIPMENT, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

	Markets					
	Normal		Expanded		Difference	
	1987-1989	1990-2001	1987-1989	1990-2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>						
Pooled	0	5.0	0	10.9	0	5.9
Sum of states	0.5	14.7	0.5	20.9	0	6.2
<u>Electric Reliability Council of Texas</u>						
Pooled	1.7	26.0	1.7	34.8	0	8.8
<u>Mid-Atlantic Area Council</u>						
Pooled	0	0	0	0	0	0
Sum of states	0.1	1.3	0.2	1.5	0.1	0.2
<u>Southwest Power Pool</u>						
Pooled	0	10.8	0	15.7	0	4.9
Sum of states	0	11.0	0	15.9	0	4.9
Total western U.S. (pooled)	1.7	41.8	1.7	61.4	0	19.6
Total western U.S. (individual states)	2.3	53.0	2.4	73.1	0.1	20.1

Table 7-12

COMPARISON OF NORMAL AND EXPANDED MARKETS DUE TO RETIREMENT
OF ALL OIL AND GAS BEFORE 1986 FOR INTERMEDIATE LOAD
GENERATING EQUIPMENT, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

	Markets					
	Normal		Expanded		Difference	
	1987-1989	1990-2001	1987-1989	1990-2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>						
Pooled	7.8	26.6	5.0	19.5	-2.8	-7.1
Sum of states	5.7	26.0	4.5	20.3	-1.2	-5.7
<u>Electric Reliability Council of Texas</u>						
Pooled	3.9	17.6	3.7	12.0	-0.2	-5.6
<u>Mid-Atlantic Area Council</u>						
Pooled	0.8	4.0	0.8	3.8	0	-0.2
Sum of states	0.7	4.0	0.8	3.8	0.1	-0.2
<u>Southwest Power Pool</u>						
Pooled	3.1	13.2	2.3	9.8	-0.8	-3.4
Sum of states	3.1	13.2	2.3	9.8	-0.8	-3.4
Total western U.S. (pooled)	14.6	61.4	11.8	45.1	-2.8	-16.3
Total western U.S. (individual states)	12.9	60.8	11.3	45.9	-1.6	-14.9

Table 7-13

COMPARISON OF NORMAL AND EXPANDED MARKETS DUE TO RETIREMENT OF
REMAINING OIL AND GAS AFTER 1990 FOR INTERMEDIATE LOAD
GENERATING EQUIPMENT, WESTERN UNITED STATES, GW
(1987-1989 and 1990-2001)

		Markets					
		Normal		Expanded		Difference	
		1987-1989	1990-2001	1987-1989	1990-2001	1987-1989	1990-2001
<u>Western States Coordinating Council</u>							
Pooled		7.8	26.6	6.8	32.9	-1.0	6.3
Sum of states		5.7	26.0	5.7	32.3	0	6.3
<u>Electric Reliability Council of Texas</u>							
Pooled		3.9	17.6	3.9	23.1	0	5.5
<u>Mid-Atlantic Area Council</u>							
Pooled		0.8	4.0	0.8	4.1	0	0.1
Sum of states		0.7	4.0	0.8	4.1	0.1	0.1
<u>Southwest Power Pool</u>							
Pooled		3.1	13.2	3.2	15.1	0.1	1.9
Sum of states		3.1	13.2	3.2	15.1	0.1	1.9
Total western U.S. (pooled)		14.6	61.4	14.7	75.2	0.1	13.8
Total western U.S. (individual states)		12.9	60.8	13.6	74.6	0.7	13.8

Table 7-14

EFFECT OF TIME OF CHANGEOVER FROM OIL AND GAS TO
COAL/SOLAR SYSTEMS ON MARKET SIZE

	Change			
	Before 1986		After 1990	
	Base Load	Intermediate Load	Base Load	Intermediate Load
<u>Western States</u>				
Pooled States	+5.9	-7.1	5.9	6.3
	-0.9	-5.7	6.2	6.3
<u>Electric Reliability Council of Texas</u>				
Pooled States	-6.6	-5.6	8.8	5.5
	-6.6	-5.6	8.8	5.5
<u>Mid-Atlantic Area Council</u>				
Pooled States	0	-0.2	0	0.1
	+0.2	-0.2	0.2	0.1
<u>Southwest Power Pool</u>				
Pooled States	4.1	-3.4	4.9	1.9
	-0.2	-3.4	4.9	1.9
<u>Total Western U.S.</u>				
Pooled Individual states	3.4	-16.3	19.6	13.8
	-7.5	-14.9	20.1	13.8

require that replacement units be placed on-line before line-focus central power units are available. These plants will be relatively new and will not be retired in the time period of first interest to this assessment (1987-2001). On the other hand, later retirement will create a market for new generating equipment in the time frame of concern. Retirements of existing oil- and gas-fired units (excepting those used for peaking service) after 1990 will create an additional market of approximately 14 GW for intermediate load applications. Thus, the intermediate load market would increase 48 to 58% if all nonpeaking oil and gas units were removed from service after 1990.

The western markets analyzed here in detail represent only about 40% of U.S. installed capacity. The western states, however, provide the major market likely to be available to line-focus central power units. As indicated earlier in Figure 7-2, unfavorable insolation regions are more prevalent in the eastern United States, and the higher line-focus central power unit costs will result in smaller, sometimes negligible, market shares.

As indicated, SRI has not examined every utility in detail, particularly small, publicly owned utilities. These utilities are more likely to purchase electricity, and their demand has been covered partly in the reported sales of major utilities. Also, they will have preferential access to federal- and state-produced electricity. Finally, they are usually too small to require units as large as 100 MW or to venture into new technology. For these reasons, we project that, although the excluded utilities represent about 25% of the total capacity in the western United States, the additional market they represent is no more than 10% of the western U.S. markets. The markets without and with these small utilities are shown in Tables 7-15 and 7-16, respectively.

Extension of the total market estimates from the western region to the remainder of the United States was by approximation only. As will be shown below, the market share projected for hybrid solar units in regions with insolation of 4.5 kWh/m^2 day or less is very small, and thus the accurate estimation of market size is of little consequence.

Table 7-15

SUMMARY OF DEMAND FOR NEW INTERMEDIATE-LOAD ELECTRIC
GENERATING CAPACITY, WESTERN UNITED STATES BASED
ON SPECIFIC UTILITIES ONLY, GW*
(1990-2001)

Normal retirement only	61-61
Normal retirement, with 1987-89 needs added	74-77
Forced retirement with 1990-2001 needs only	75

*Data rounded to nearest GW.

Table 7-16

SUMMARY OF DEMAND FOR NEW INTERMEDIATE-LOAD ELECTRIC
GENERATING CAPACITY FOR ENTIRE WESTERN UNITED STATES,* GW
(1990-2001)

Normal retirement	67-69
Normal retirement, with 1986-89 needs added	82-85
Forced retirement with 1990-2001 needs only	82-83

*Including Louisiana.

As a first approximation, the growth in demand for electricity (and for new capacity) will proceed at the same rate over the eastern and western U.S. The current and near-term projected sales (capacity) for the regions is in the ratio 60/40. Using this ratio and the data of Table 7-16, we estimate total U.S. markets as shown in Table 7-17.

Table 7-17

SUMMARY OF DEMAND FOR NEW INTERMEDIATE-LOAD ELECTRIC
GENERATING CAPACITY FOR THE UNITED STATES

	<u>Western U.S.</u>	<u>Eastern U.S.</u>	<u>Total U.S.</u>
Normal retirement	67-69	100-104	167-173
Normal retirement with 1986-89 needs added	82-85	123-128	205-213
Forced retirement with 1990-2001 needs only	82-83	150-180	232-263

The actual markets will be limited by economic and other factors, as discussed in Section 6. Also discussed in Section 6 are the rates of market penetration to be expected.

7.4.4.3 Unit Size

A potentially important factor that could influence the economics of line-focus central power units (and therefore the market share of the concept) is the unit size. Larger units should be more economic and should gain a larger share of the total available market.

An analysis was made of the 61 electric utility systems used to establish the total available market in the western U.S. to see if there were limitations on plant size. As shown in Table 7-18, only five utility systems with 3-GW capacity (8% of total number and about 1.5% of expected capacity in 1989) would not be able to use units as large as 100 MW. Twelve additional utilities (20%) with 9% of the total capacity could use units up to 199 MW. Another 12 with 12% of the total capacity could use units up to 299 MW. Nine of the 61 utilities considered (13% of the capacity) could use units as large as 399 MW. The remaining 23 systems, representing 66% of the expected capacity, could use even larger units. A 300-MW_e unit design would be suitable for approximately 80% of all utility systems.

Table 7-18

UTILITY SYSTEMS WITH LIMITED CAPACITY TO ACCEPT
HIGH-TEMPERATURE LINE FOCUS SOLAR UNITS
OF SPECIFIED SIZES*

<u>Size of Acceptable Unit (MW)</u>	<u>Number of Systems</u>	<u>Total System Capacity (Rounded) (GW)</u>
100	5	3
100-199	12	18
200-299	12	25
300-399	9	29

* Unit should be no more than 20% of total capacity or one-third of intermediate load capacity.

7.4.5 Market Analysis

7.4.5.1 Introduction

The adoption of solar central station power systems will be determined by (a) the market needs, (b) the comparative economics of commercial and other near commercial systems for producing electric power and (c) other less tangible factors, including consumer preference, material constraints, and social and political pressures. The following analysis discusses the influence of comparative economics on market share. The impact of these factors on markets for line-focus central power systems will be evaluated and market projections presented.

7.4.5.2 Equilibrium Market Share

Economists believe that the need for goods or services will be fulfilled by competing suppliers. Unless the market is very restricted or one supplier has an overwhelming advantage in price, many suppliers will have a share. As new goods are introduced into the marketplace, they will also gain a share. Thus, the total available market (e.g.,

for base load electric generating capacity) is currently shared between various nuclear, coal, oil, or gas-fueled steam generating electric power units and hydropower units.

As an idealization, the share of a particular market that a single new technology or product can attain in competition with another technology at any particular time under steady-state conditions can be represented by the curve shown in Figure 7-3 and is given by:

$$\text{Equilibrium market share to solar technology} = \frac{1}{1 + \left(\frac{P_s}{P_a} \right)^\gamma}$$

where P_s and P_a are the marginal prices of the solar energy product (such as solar-derived electricity) and the alternative (competing) energy product (such as coal-derived electricity), respectively. This static representation indicates that when P_s and P_a are equal under

FIGURE 7-3 EQUILIBRIUM MARKET SHARE

steady-state conditions, the market will be shared equally. The market share parameter (γ) is a measure of market imperfections, price variations, and consumer preferences.

When two or more competing products are competing for a share of the same market as is the solar-derived product, a more general formula is used. For example, if N different competing technologies all produce the same product, then the solar market share is represented by the following equation:

$$\text{Equilibrium solar market share} = \frac{1}{1 + \left(\frac{P_s}{P_{a1}}\right)^\gamma + \left(\frac{P_s}{P_{a2}}\right)^\gamma + \left(\frac{P_s}{P_{a3}}\right)^\gamma + \dots + \left(\frac{P_s}{P_{aN}}\right)^\gamma}$$

where P_{a1} through P_{aN} represent the prices of the first through the N th alternative (competing) products. If all of the prices P_{a1} through P_{aN} and P_s were equal, each product would receive $1/(N + 1)$ of the market.

In the static economic analysis performed here, representative prices for electricity generated by the HTLF system under different assumed conditions are compared to those from alternative energy sources. Actually, significant individual variations from these representative prices do exist. The market share parameter compensates for the fact that the analysis uses representative prices instead of price ranges.

The noneconomic behavior of marketplace decision-makers is another factor considered by the market share parameter. Even if a new technology is somewhat more expensive than the alternative, some fraction of purchasers will choose it, perhaps because of novelty, environmental reasons, or "energy independence" considerations. Alternatively, some fraction of purchasers will continue to use familiar energy sources even if economic considerations dictate a change. Imperfect price information may also cause a decisionmaker to act in a noneconomic fashion.

In a perfect market with a high level of price sensitivity and none of these real world effects, γ would be infinite, and the energy product with even a very slight economic advantage would obtain a 100% steady-state market share. Such conditions do not describe real energy markets; instead, more realistic response patterns of various markets can be modeled by a suitable choice of γ . For example, large utility systems, such as those that might purchase an HTLF power plant, would generally be modeled with high γ values. These values reflect the strong response to price variations characteristically displayed by utility consumers who deal with large quantities of energy and are acutely aware of economic considerations. Much lower gamma values would be used to model smaller scale energy consumers who typically are influenced as much by personal values as by economics. Factors such as aesthetics, convenience, and novelty would weigh more heavily with a residential than an industrial consumer.

Within a single utility, many choices such as coal purchase are highly responsive to price, i.e., the low bidder almost always wins even though his marginal advantage is very small. In this case, γ approaches infinity. Choices between different electric generating methods, other factors such as familiarity with equipment suppliers, desire to have alternate fuels, and perceived attitudes of regulatory bodies may influence choices. The response parameter, γ , will decline from infinity to a high value of, say, 35. Finally, individual utility systems have different load demand patterns, mixes of existing generating capacity, and different regulatory bodies to which they must respond. Under these circumstances, the response is still broader. Since all utilities are strongly influenced by requirements to provide service at low cost, the demand parameter must still reflect this fact. As a practical matter, we have used a γ of 20 for this and other studies. With a γ of 20, and a single competitive product, the HTLF unit would obtain a 90% equilibrium market share if the ratio of levelized busbar costs P_s/P_a equalled 0.8959. Similarly, the unit would gain only 10% of the equilibrium market if the ratio P_s/P_a equalled 1.116.

Typical equilibrium market shares for an HTLF starting up in 1990 are presented in Table 7-19 under a number of assumptions. Included are variations in HTLF capital costs, insulation levels, initial coal prices, and coal price escalation rates. The prices of electricity from HTLF and competing systems used to derive the values in Table 7-19 were shown earlier in Table 7-4.

The market shares associated with 80th plant HTLF capital costs do not represent realistic values for the 1990 time frame. The real competition in 1990 will be between a 1st HTLF unit and several more conventional types of electric generating systems. Unless the manufacturer or the government provides a discount or subsidy, the 1st HTLF unit will produce electricity at a higher cost than the 80th plant and will consequently receive a lower equilibrium market share, as shown in Table 7-19. As HTLF units are installed, the price of electricity produced from them will fall, in constant dollar terms, until 80th plant conditions are reached.

7.4.5.3 Dynamic Market Response

The adoption of HTLF central station power systems will be influenced by both real and perceived impediments. These impediments can require plant modifications, leading to higher capital and/or operating costs, which in turn reduce the market share. Such real impediments are reflected in the calculations discussed in the previous section. Alternately, they can result in institutional adjustments required by siting, regulatory or other difficulties that affect the purchasing utility's willingness to buy. This change in willingness would be reflected in the rate at which the market is penetrated.

At the time of introduction of the HTLF system, and for some time thereafter, customer unfamiliarity and other market restraints will also

Table 7-19

EQUILIBRIUM MARKET SHARES FOR HTLF PLANTS IN INTERMEDIATE LOAD SERVICE
(no "behavioral lag" is considered; 1990 startup; 40% capacity factor)

	Coal Price Escalation Rate (%/yr)	Equilibrium Market Share (% captured in 1990)							
		\$1.00/MMBtu Coal Price				\$1.40/MMBtu Coal Price			
		Solar Insolation (kWh/m ² day)				Solar Insolation (kWh/m ² day)			
		4.5	5.5	6.5	7.5	4.5	5.5	6.5	7.5
<u>HTLF (1st plant capital cost)</u>	6%	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.2
	8%	0.0	0.0	0.1	0.7	0.0	0.0	0.7	6.3
<u>HTLF (80th plant capital cost)</u>	6%	0.0	0.0	0.3	2.5	0.0	0.2	2.4	19.8
	8%	0.0	0.6	8.5	48.3	0.2	5.4	47.6	90.1

Note: See Table 7-4 for the levelized busbar power costs used in deriving these equilibrium market shares.

inhibit purchase of the "new" system.* These factors influence the rate at which actual sales approach the equilibrium market condition (market penetration). Full delineation of markets as a function of time will be based on successive steps from the 1st to the 80th plant costs and on other market factors.

The equilibrium market share curve presented earlier in Figure 7-3 is only a static representation. To assess the dynamics of market penetration, a "dynamic market response curve" can be used to describe how fast the current market will move toward the static price-determined market share curve as a result of real-world behavioral response. This is called the behavioral lag effect.

The dynamic market response curve, given by

$$\frac{1}{1 + \left(\frac{h}{n}\right)^\alpha}$$

is shown in Figure 7-4 where

h = behavioral lag half-life (time required for one-half of the market to respond to the entrance of a new technology)

n = years since introduction of new technology

α = behavioral lag response parameter.

This curve tends to slow the introduction of a new technology based on the time that it takes for decision-makers to accept and switch to new energy sources, especially those that are based on unfamiliar technologies.

* While newness of the technology, siting problems, etc., may be viewed as impediments, other aspects of solar plant use may be viewed as incentives. For example, at present, a utility adopting solar technology is likely to gain in image among its customers--it is using what is popularly identified as a renewable and benign technology. Such positive factors will create a willingness to purchase that will also be reflected in the rate of market penetration.

FIGURE 7-4 DYNAMIC MARKET RESPONSE

The behavioral lag parameters, h and α , provide a means of quantifying the dynamic market response. The half-life, h , defines the number of years that it would take for 50% of the market to respond to the introduction of the new technology. The second behavioral lag parameter, α , fixes the relative shape (curvature) of the dynamic market response curve once the half-life parameter has been chosen.

The parameters chosen should be representative of past actions of specific industries, in this case the electric utility industry. Modifications of parameters to reflect system differences or impediments should not be necessary for the HTLF central station system. The values of h and α , namely 10 and 4, are those used in previous studies of solar central-station electric markets (Ref. 7-11).

7.4.5.4 Dynamic Market Share

To find the share of the open market captured by the solar technology in any particular year, the equilibrium market share and dynamic market response curves are multiplied. This results in a dynamic

market share for the solar technology that varies with time. This process is depicted by Figure 7-5. For given price ratios such as shown by A or B on the curve to the left, there is an associated equilibrium market share, as denoted by the dotted lines. These depict the ultimate market shares that would eventually be achieved if the price ratios were to remain constant. Curves A and B on the right in Figure 7-5 demonstrate the behavioral lag effect that delays the introduction of new technologies.

STEADY-STATE MARKET SHARE

BEHAVIORAL LAG EFFECT

FIGURE 7-5 CALCULATION OF DYNAMIC MARKET SHARE

Curve A, for example, presents the dynamic market share as a function of the time following commercial introduction. It rises from zero to a maximum of about 28% as the new technology becomes known to larger portions of the market.

In general, the price ratio, $P_{\text{solar}}/P_{\text{conventional}}$, does not remain constant. This ratio is expected to decline as learning effects decrease solar costs and as fossil fuel prices increase. Accordingly, a movement from point A to point B on the curve to the left could represent a realistic situation. In this case, the actual dynamic market share curve would lie between curves A and B on the right. This curve, labelled "market behavior" starts out the same as curve A, then rises with time to meet curve B.

Once the dynamic market share (or "market behavior") curve has been determined, the demand for HTLF systems is found by applying this curve, in time steps, to estimates of the market size that open up to capture.

7.4.5.5 Market Penetration Results

In performing long-term assessments for the marketability of power generation equipment, it is impossible to escape the conclusion that penetration results are highly sensitive to a number of important parameters. This fact was clearly demonstrated by the wide variations in the equilibrium market share shown earlier in Table 7-19. Those results ranged from zero to 90%, depending upon solar insolation levels and assumptions made about HTLF capital costs and the future fuel costs for the HTLF's major competitor, coal-fired power generation. Since a large degree of uncertainty surrounds almost any long-term cost projection, market penetration results based upon such projections are necessarily speculative. As a result, sensitivity analyses were performed as a part of this study to measure the response of the market to variations in initial coal prices and coal price escalation rate. As discussed previously, HTLF capital costs were assumed to vary with the number of units installed, starting with 1st plant costs and then eventually decreasing to 80th plant costs if sufficient market penetration is achieved.

This section presents the results of the HTLF market penetration analysis as a function of four time periods (1990, 1995, 2000, and 2010) and twelve regions that were developed by roughly grouping utilities according to solar insolation level. (These regions are depicted in Figure 7-6).

As might be expected, the ranges of coal prices and escalation rates employed in this study result in a wide variation in market penetration results. When both parameters are assumed to be in the favor of coal, (i.e., a \$1.00/MMBtu initial coal price rising at an annual rate of 6% per year*), the HTLF would not be expected to capture any of the available market.

At the higher coal price escalation rate of 8% per year, however, an entirely different market outlook results. Since a levelized cost analysis is employed in developing busbar cost estimates, the effect of a higher escalation rate is compounded over the 30-year life of a coal-fired power plant. As noted in Section 2.3, this results in a substantial increase in the cost of electric power from coal-fired plants and enhances the market position of HTLF facilities. Table 7-20 presents the results of this market penetration. A total of 24 HTLF units are projected as being installed in the western U.S. before the turn of the century and a total of 162 by the year 2010. The heaviest concentrations of solar usage are assumed to occur in the southwest. Approximately 80% of the penetration occurs in the two regions that encompass California, Arizona, New Mexico, Nevada, Utah, and Colorado.

If the initial (1978) price of coal is \$1.40 per million Btu and it escalates at 6%/yr, the solar market penetration would be very small. As shown in Table 7-21, only 200-400 MW of installations, occurring in Arizona and southern California, would be expected by 2010. Under these assumptions 80th plant conditions would not be reached until well after 2020.

* As noted in Section 2.2, a 5% per year general inflation rate has been assumed in this analysis. A 6% per year coal price escalation rate therefore implies a 1% per year real increase in the price of coal.

FIGURE 7-6 REGIONAL DISAGGREGATION EMPLOYED IN HTLF MARKET PENETRATION ANALYSIS
 (Insolation levels are expressed in kWh/m²-day)

Table 7-20

MARKETS FOR HTLF UNITS, CUMULATIVE INSTALLED CAPACITY, MW
 (\$1.00/MMBtu Coal Price; 8% Coal Price Escalation Rate)

<u>Region*</u>	<u>1990</u>	<u>1995</u>	<u>2000</u>	<u>2010</u>
1	0	200	1,900	9,400
2	0	0	300	3,700
3	0	0	0	0
4	0	0	0	0
5	0	0	100	700
6	0	0	100	1,800
7	0	0	0	100
8	0	0	0	0
9	0	0	0	100
10	0	0	0	0
11	0	0	0	400
12	0	0	0	0
Total units in interval		2	22	138
Cumulative total		2	24	162

* Based on 55 electric utilities in the western United States, representing about 75% of installed capacity in the year 2000 and an estimated 90% of the total market for the HTLF system. Regions are defined in Figure 7-6.

Table 7-21

MARKET FOR HTLF UNITS, CUMULATIVE INSTALLED CAPACITY, MW
 (\$1.40/MMBtu Coal Price; 6%/yr Coal Price Escalation)

<u>Region*</u>	<u>1990</u>	<u>1995</u>	<u>2000</u>	<u>2010</u>
1	0	0	0	200-400
2	0	0	0	0
3	0	0	0	0
4	0	0	0	0
5	0	0	0	0
6	0	0	0	0
7	0	0	0	0
8	0	0	0	0
9	0	0	0	0
10	0	0	0	0
11	0	0	0	0
12	0	0	0	0
Total units in interval		0	0	2-4
Cumulative total		0	0	2-4

* Based on 55 electric utilities in the western United States representing about 75% of installed capacity in the year 2000 and an estimated 90% of the total market for the HTLF system. Regions are defined in Figure 7-6.

Much more favorable sales of HTLF units result if the initial coal price of \$1.40 per million Btu is assumed to escalate at 8%/yr rather than 6%/yr. As Table 7-22 indicates, there would be significant market penetration even by 1995 under these conditions. Units are installed in 7 of the 12 areas by 2000 and all but one area by 2010.

A concise summary of expected market (sales) for HTLF systems is shown in Table 7-23.

As applied, the penetration model does not include the effects of some external variables that could limit market saturation. One such is the penalty that could be assigned because of the dependence of the HTLF system on the weather. Southern California Edison (SCE) Company (Ref. 7-12) has estimated the additional capacity required to assure equivalent reliability for solar as for conventional-fueled systems. If, in a normal system, a 15.3% reserve margin is required,* then for a 5% penetration of solar units with energy storage capacity (the baseload HTLF unit) we estimate from SCE data a requirement for 16-16.5% reserve margin. With a 20% penetration of these solar units, the reserve margin requirement is in the range of 28.5-29.7%. Thus, extreme penetration of the market may lower the value (raise the apparent cost) of the HTLF units. The degree to which this occurs will depend strongly on the particular utility system.

Table 7-24 summarizes the individual region intermediate load capacity requirements as of the year 2000 and compares these with the projected markets for HTLF units, assuming an initial coal price of \$1.40 per million Btu. It can be seen that with a 6% coal price escalation rate, the solar market penetration is not sufficient to necessitate consideration of saturation effects. Even assuming an 8% coal price escalation rate, these effects would not be important through the end of the century; only Region 1 is likely to utilize sufficient solar capacity by 2000 to warrant a slightly greater system reserve

*SRI calculations of capacity requirement assumed a 20% reserve margin.

Table 7-22

MARKETS FOR HTLF UNITS, CUMULATIVE INSTALLED CAPACITY, MW.
(8% Coal Escalation; \$1.40/MMBtu Coal Price)

Region*	1990	1995	2000	2010
1	0	800	4,200	12,500
2	0	300	2,700	9,400
3	0	0	0	0
4	0	0	0	400
5	0	0	400	1,100
6	0	100	1,100	4,400
7	0	0	200	1,600
8	0	0	0	800
9	0	0	100	1,200
10	0	0	0	300
11	0	0	500	4,400
12	0	0	0	100
Total units in interval		12	80	270
Cumulative total		12	92	362

*Based on 55 electric utilities in the western United States representing about 75% of installed capacity in the year 2000 and an estimated 90% of the total market for the HTLF system.

Table 7-23

SUMMARY OF HTLF MARKET PENETRATION IN WESTERN U.S. MARKETS
(In Cumulative Numbers of 100 MW Units)

Year	<u>\$1.00/MMBtu Coal Price</u>		<u>\$1.40/MMBtu Coal Price</u>	
	<u>Coal Price Escalation Rate</u>		<u>Coal Price Escalation Rate</u>	
	<u>6%/yr</u>	<u>8%/yr</u>	<u>6%/yr</u>	<u>8%/yr</u>
1990	0	0	0	0
1995	0	2	0	12
2000	0	24	0	92
2010	0	162	2-4	362

margin than would otherwise be needed. By the year 2010, however, several regions may have reached practical saturation levels. Thus, the markets projected for the 8% coal price escalation case may be overstated.

7.4.5.6 Conclusions

The HTLF system has a potential for filling market needs for intermediate load service. The extent of market penetration is highly dependent upon relative capital costs for HTLF and competing systems, the location (insolation), and the assumed price and price escalation for coal.

The best opportunity for early service is in regions of high insolation, especially southern California, Arizona, and New Mexico. Other areas, notably Texas, may also offer large markets. The eastern United States is not likely to use HTLF units.

The maximum number of units installed under the most favorable assumptions studied (i.e., coal at \$1.40 per million Btu and with an 8% escalation rate or 3% in constant dollar terms) is projected at 362 by 2010. This number may not be reached, however, because of additional

Table 7-24.

PROJECTED INTERMEDIATE-LOAD CAPACITY BY REGION
IN 2000 AND ESTIMATED SOLAR PENETRATIONS
IN 2000 AND 2010

Region	Projected Intermediate Load Capacity in 2000 [†] (GW)	Projected Solar Capacity (GW)*			
		Coal Price Escalation Rate			
		6%/year		8%/year	
		2000	2010	2000	2010
1	21.1	0	0.2-0.4	4.2	12.5
2	16.7	0	0	2.7	9.4
3	2.6	0	0	0	0
4	16.9	0	0	0	0.4
5	2.2	0	0	0.4	1.1
6	7.2	0	0	1.1	4.4
7	6.6	0	0	0.2	1.6
8	24.1	0	0	0	0.8
9	3.2	0	0	0.1	1.2
10	7.6	0	0	0	0.3
11	13.8	0	0	0.5	4.4
12	4.0	0	0	0	0.1

* Initial coal price is \$1.40/MMBtu.

[†] Intermediate load capacity is assumed to be 31% of total installed capacity.

reserve capacity requirements associated with the use of non-firm solar generation capacity.

In addition to the increase in cost of fossil fuels above the general inflation rate, environmental factors may also accelerate the commercialization of solar thermal power.

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8.0 DEVELOPMENT PLAN

The following program elements for short-range and long-range development of the line-focus central power system are presented to assist DOE in its implementation of low-cost electricity from solar thermal central power plants.

8.1 Short-Range Development Plan

8.1.1 Reduction of Heliostat Cost

About \$2 of the estimated \$5/ft² of the heliostat cost is represented by the cost of the mirror and the mirror plastic substrate, particularly the latter. We propose to analyze the use of glass-reinforced concrete (GRC), a much lower cost material, for the mirror substrate. The following tasks should be performed during this analysis:

- Stress and deflection analysis of GRC substrate
- Mirror/GRC adhesives analysis
- GRC weight and material cost estimate
- GRC production technique analysis
- Substrate experiment: experiment design and cost estimate
- Revised heliostat cost estimate
- Letter report.

8.1.2 Reduction of Receiver Cost

Performance and design requirements for the line-focus receiver will be reviewed. These requirements will include minimum and maximum salt flow rates per linear foot of receiver, inlet and outlet salt temperatures, inlet pressures, allowable pressure drops, insolation fluxes, and methods for applying optical coatings to the receiver panels.

Based on the design requirements outlined above, a number of potential Roll Bond[®] type heat exchanger panel designs will be prepared. The assessment will include material and manufacturing technique selections. The materials considered will include carbon and austenitic stainless steels. The manufacturing techniques will include embossed and welded construction and welded and hydraulically expanded construction. From this assessment, the panel material and construction technique will be selected.

From the above results, a conceptual design will be prepared for the Roll Bond[®] type heat exchanger panel. The design will include specifications of the materials, flow patterns, pressure drops, heat transfer coefficients, and required heat transfer area. Once the panel design has been finalized, the impact of the design on the receiver length and the salt pump power requirements will be addressed. Automated techniques for producing large quantities of the receiver panels will be considered. The study will focus on conceptualizing a manufacturing facility dedicated to the mass production of the panels.

Capital costs of the receiver panels will be developed. Costs will be developed for small quantities available from an established manufacturer. Costs will also be developed for large quantities, based on the capital cost of the dedicated manufacturing facility and the purchase of the heat exchanger material in large quantities. Installation costs for the receiver panels will also be estimated.

8.1.3 Further Wind Tunnel Tests of Receiver Model

Since the purpose of the cryogenic wind tunnel was to approximate natural convection heat losses more nearly, and since the cryogenic wind tunnel tests showed natural convection heat losses to be very low, we propose further testing of receiver models in air. We have arranged to measure heat losses with a larger scale model in the Ames Research 6 ft x 10 ft wind tunnel, with and without wind diverting panels. The air velocities inside the cavity and the reduction of heat losses using these wind diverting panels can be measured.

The following tasks would be performed during the proposed wind tunnel model tests:

- Experiment design and test plan
- Equipment design and construction
- Tests of unbaffled receiver model
- Tests of baffled receiver model
- Test data reduction and analysis

8.2 Long-Range Technology Development Program

The following technologies necessary to the economic success of the line-focus central power system are now being developed as part of the point-focus central power system program:

- Molten salt storage techniques
- Molten salt related metallurgy
- Low-cost reflective surfaces
- Efficient reflective surfaces.

These and other technologies are common to both point-focus and line-focus central power systems. The line-focus central power system will also benefit from the advancements already made in the point-focus central power program.

To implement low cost electricity from line-focus solar thermal central power systems in the 1980s, it is necessary to reduce to practice several technologies more specifically important to the economic success of the line-focus central power system. These are:

- Lower cost single-axis heliostats
- Lower cost line cavity receiver
- Receiver selective surfaces production capability
- Proof of concept experiment
- Low-cost cleaning of single-axis heliostats.

Although these technologies are important to the economic success of the line-focus central power system, they are also useful to other

solar technologies (e.g., glass-reinforced cement materials for two-axis heliostats, and selective surfaces for flat-plate and parabolic-trough collectors).

On the basis of these advantages, it is proposed that DOE plan for a long-range technology development program, as highlighted in paragraphs 8.2.1 through 8.2.6.

8.2.1 Low-Cost Single-Axis Heliostat

- Glass-reinforced concrete (GRC) technology--Make test pieces of GRC for measurement of properties, durability, and manufacturing techniques (1 year).
- Heliostat construction--Build a heliostat module with GRC for measurement of properties, durability, and study of manufacturing techniques (2 years).
- Reflective surface construction--Build a reflective surface with GRC substrates for measurement of properties, durability, and study of manufacturing techniques (2 years).
- Heliostat module construction--Build and test at HTTF, Sandia, Albuquerque, a heliostat module for measurement of performance, durability, and study of manufacturing techniques (2 years).
- Further heliostat cost reduction--Perform studies of further manufacturing cost reduction techniques and performance/cost tradeoffs (1 year).

8.2.2 Low-Cost Line Cavity Receiver

- Roll Bond[®] panel construction--Make expanded metal heat transfer panel and test with molten salt with radiant lamps under thermal cycling and freezing regimes (1 year).
- Receiver test--Test production prototype receiver with solar energy at HTTF (2 years).
- Panel cost reduction--Make manufacturing process and tooling studies for complete automation of panel manufacturing (1 year).

8.2.3 Receiver Selective Surfaces Oven

- Chemical vapor deposition (CVD) oven design--Based on the proposed preliminary studies on the current project, design and detail an atmospheric pressure CVD oven capable of thin film selective surface deposition on line-focus cavity receiver panels (1 year).

- CVD oven construction and test--Build the CVD selective surface oven and coat line-focus receiver panels with high solar absorptance/low infrared emissivity surfaces (2 years).
- CVD oven modification--Modify CVD oven to coat other solar thermal absorbers, e.g., parabolic trough receivers and flat panel collectors (2 years).

8.2.4 Proof of Concept Experiment

- Receiver experiment--Build and test a receiver row 60 m long on 60-m tall towers in front of the HTTS tower. Test performance with extant two-axis heliostats, simulating calculated one-axis heliostat field solar concentration at the receiver aperture (2 years).
- Proof of concept--Move test receiver to Barstow, add additional receivers and test with prototype single-axis heliostat field (3 years).

8.2.5 Low-Cost Cleaning of Single-Axis Heliostat

- Design and build self-propelled single-axis heliostat washing machine--this requires no technological breakthroughs, but no such machine is commercially available. This development can be done in conjunction with the proof of concept experiment (2 years).

8.2.6 10 MW_e Experiment (3 years)

This experiment will be similar in scope and purpose to the power tower Barstow experiment.

8.2.7 Long-Range Development Schedule

The long-range DOE development program for line-focus central receiver solar thermal power system is shown graphically in Figure 8-1.

FIGURE 8-1 LONG-RANGE DEVELOPMENT PROGRAM FOR LINE-FOCUS
CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM