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**CENTRAL RECEIVER
SOLAR THERMAL POWER SYSTEM
PHASE 1**

**CDRL ITEM 10
Final
Technical Progress Report**

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY

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SOLAR THERMAL POWER SYSTEM
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Final
Technical Progress Report**

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PREFACE

This report is submitted to the Department of Energy under Contract EY-76-C-03-1108 as the final documentation of CDRL Item 10, Final Technical Progress Report. It summarizes the analysis, design, and test efforts performed on the Phase I Central Receiver Solar Thermal Power System Program by the MDAC team between 1 July 1975 and 30 June 1977. This report was prepared for distribution by DoE to the technical public under Standard Category UC-62, as contained in Document TID-4500.

Specific efforts performed by the members of the MDAC team during the two-year contract period were as follows:

- McDonnell Douglas Astronautics Company
 - Commercial System Definition
 - Pilot Plant System Definition
 - Collector Subsystem Analysis and Design, and Subsystem Research Experiment Program
 - Master Control Subsystem Definition
 - Thermal Storage Subsystem Integration
- Rocketdyne Division of Rockwell International
 - Receiver Assembly Analysis and Design, and Subsystem Research Experiment Program
 - Thermal Storage Subsystem Analysis and Design, and Subsystem Research Experiment Program
- Stearns-Roger Inc.
 - Electrical Power Generation Subsystem Analysis and Design
 - Pilot Plant and Commercial Tower and Riser/Downcomer Analysis and Design
- University of Houston
 - Collector Field Optimization
- Sheldahl Inc.
 - Heliostat Reflective Surface Development
- West Associates
 - Utility Consultation on Pilot Plant and Commercial System Concepts

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ABSTRACT

Results of analysis and design efforts by McDonnell Douglas Astronautics Company (MDAC), Rocketdyne, Stearns-Roger, Inc., Sheldahl, Inc., and the University of Houston between 1 July 1975 and 30 June 1977 on DoE Contract No. EY-76-C-03-1108 are summarized. This is the Final Technical Progress Report published on the Phase 1 Central Receiver Solar Thermal Power System contract.

Historical summaries and final selection of 10-MWe pilot plant and 100-MWe commercial systems are presented, with emphasis on the collector field characteristics, overall system performance, selection of steam/feedwater operating conditions, and rationale for system and subsystem selection.

The commercial and pilot plant designs, as well as the subsystem research experiment activities for the collector, receiver, and thermal storage subsystems are presented, including a historical summary, design summary, and a description of the overall SRE test program and major test results for each of the subsystems.

Section 1

INTRODUCTION AND SUMMARY

An overview of technical progress on the Phase 1 Central Receiver Solar Thermal Power System contract during the period between 1 July 1975 and 30 June 1977 is presented in the following sections of this document. This is the seventh and final technical progress report to be published during Phase 1.

1.1 PROGRAM APPROACH

The objectives of the Central Receiver Solar Thermal Power System Phase 1 contract are two-fold: (1) complete a preliminary design of a 10-MWe pilot plant that best simulates a cost-competitive commercial system and provides technical and operational data necessary to evaluate pilot plant technical feasibility, performance, schedules, and costs; and (2) define and carry out a series of test programs to verify the critical subsystems contained in the design. The methodology used to accomplish this program is presented in Figure 1-1. Starting with a series of program inputs which included DoE, utility, and self-imposed constraints, along with representative environmental conditions, a preliminary design baseline phase was conducted. In order to provide proper focus to this activity, a commercial plant and related commercial system requirements were initially defined from which the pilot plant preliminary baseline definition and subsystem research experiment requirements could be derived. With the establishment of the preliminary baseline design, program activity continued into the subsystem research experiment (SRE) phase.

The SRE phase for the collector, receiver, and thermal storage subsystems progressed from the conceptual design of the test programs and hardware into the detailed design, fabrication, test, and evaluation of the SRE test results. A summary of the SRE program is contained in Sections 4, 5, and 6 for the Collector, Receiver, and Thermal Storage, respectively.

Figure 1-1. Central Receiver, Phase 1, Program Network

The systems effort during the SRE phase consisted of formulating pilot plant budgetary cost estimates (which were submitted on March 1, 1976); investigation of pilot plant design alternatives; and a top-level revision of the commercial system definition. The commercial system revision was initiated during the summer of 1976 and used as input to the refinement of pilot plant analyses and the formulation of the recommended approach to pilot plant design.

The preliminary design phase was initiated in late 1976 using the SRE test results to date, in combination with updated definitions of commercial and pilot plant requirements, to expand the commercial system definition, generate commercial system cost estimates, update the pilot plant preliminary design and refine the pilot plant cost estimates. The results of this effort were published in the Preliminary Design Report, MDC G6776, dated May 1977.

1.2 PROGRAM SUMMARY

The MDAC Central Receiver Solar Thermal Power System is shown schematically in Figure 1-2. It features a collector subsystem, which is composed of an array of tracking mirrors, or heliostats, which reflect incident solar radiation on the external surface of a receiver, or boiler, to convert water into steam. The working fluid enters the receiver tubes at the bottom as water and is heated sufficiently by the absorbed thermal energy to generate superheated steam at the receiver outlet. The receiver steam is routed either to the inlet of a conventional turbine-generator set for direct electrical power production, or it can be sent to a thermal storage subsystem, where the thermal energy can be extracted from the working fluid and stored in the form of high-temperature oil and rocks for subsequent regeneration of steam and electrical power production during periods of little or no available solar energy. The system is controlled through a master control unit, which uses a central computer, in combination with the required peripheral equipment and software, to monitor and control the subsystem interfaces and overall system operation during normal operating modes and transient conditions.

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Figure 1-2. Central Receiver Baseline Concept

A summary of accomplishments during the Phase 1 contract effort follows:

- The MDAC team has defined a single-module 100-MWe Commercial System with emphasis on cost-competitiveness, simplicity, and operating flexibility to meet future utility requirements.
- A cost-competitive 10-MWe Pilot Plant System preliminary design has been defined which simulates the commercial system to a high degree.
- The Collector SRE tests have successfully demonstrated precise heliostat pointing and tracking capability, low power consumption, and structural integrity throughout a wide range of operating environments.
- The Receiver SRE tests have successfully demonstrated operational flow stability without orifices, the capability to respond safely to rapid heat-flux transients and temporary loss of water, and better-than-expected heat-transfer characteristics and fatigue life of the receiver tubes.
- The Thermal Storage SRE tests have successfully met all objectives, including demonstration of long-term thermal stability and compatibility of the thermal storage fluid with rock and materials of construction, and the development, movement, and operational stability of a sharp thermal gradient, or thermocline, in a large-scale, dual-medium system.

The reviews conducted by Sandia, DoE, and Aerospace during the course of the 2-year contract effort are indicated in Table 1-1. Briefing manuals were prepared and delivered for each of the reviews tabulated, in accordance with specifications of the Contract Data Requirements List items. Additionally, five informal reviews were conducted by Sandia, as follows:

Collector Review	October 21 and 22, 1976	MDAC-Huntington Beach and China Lake (NWC)
Receiver Review	October 22, 1976	Rockwell B-1 Facility, El Segundo

Thermal Storage
Review

November 3, 1976

Rocketdyne-Canoga
Park

Elec Pwr Gen
Subsys Review

November 16, 1976

Stearns-Roger,
Denver

Systems Review

January 24 and 25, 1977

MDAC-Huntington
Beach

The Preliminary Design Baseline Report, MDC G6040, was published in January 1976, in accordance with the requirements of CDRL Item 1. Draft copies of the Preliminary Design Report, MDC G6776, dated May 1977, were prepared and submitted in early May in accordance with the requirements of CDRL Item 2, and were published in October 1977.

A summary schedule of the Phase 1 contract effort is shown in Figure 1-3.

Table 1-1
PROGRAM REVIEWS

Review	CDRL Item	Date	Location
First Project Review	10	July 29, 1975	Huntington Beach, California
Second Project Review and Preliminary Design Base- line Review (PDBR)	1	October 14, 1975	Huntington Beach, California
Receiver Conceptual Design Review (CDR)	4	October 15, 1975	Canoga Park, California
Thermal Storage Conceptual Design Review (CDR)	5	October 15, 1975	Canoga Park, California
Collector Conceptual Design Review (CDR)	3	October 16, 1975	Huntington Beach, California
Semi-Annual Review	10	November 17-19, 1975	Las Vegas
Third Project Review	10	March 2, 1976	Huntington Beach, California
Collector Detail Design Review (DDR)	6	March 2, 1976	Huntington Beach, California
Receiver Detail Design Review (DDR)	7	March 3, 1976	Canoga Park, California
Thermal Storage Detail Design Review (DDR)	8	March 3, 1976	Canoga Park, California
Semi-Annual Review	10	June 2, 1976	Washington, D.C.
Fourth Project Review	10	August 31 and September 1, 1976	Huntington Beach, California and Canoga Park, California
Semi-Annual Review	10	December 7 and 8, 1976	Denver
Fifth Project Review	10	March 1 and 2, 1977	Huntington Beach, California
Preliminary Design Review (PDR)	2	May 2 and 3, 1977	Livermore, California

Figure 1-3. Summary Master Program Schedule

Section 2

COMMERCIAL SYSTEM

The commercial system design effort, which was undertaken as part of this Central Receiver Solar Thermal Power System study, was performed to provide a design reference for the 10-MWe pilot plant and to permit the early development of economic data. The basic ground rule used in developing the commercial design was to define a configuration that would minimize the cost of electrical energy on an annual basis while being compatible with DoE-imposed design guidelines and requirements. In addition, an effort was made at all times to use existing off-the-shelf equipment to minimize the need for additional hardware development and to minimize program risk.

The methodology used to define the system was to work backward, in an energy flow sense, from the existing turbine and balance-of-plant equipment through the water-steam loop and receiver, terminating with the collector field. In this manner, it was possible to establish the steam conditions associated with the off-the-shelf equipment as requirements for the energy collection and thermal storage subsystems. In addition, design refinements made to the turbine and balance-of-plant equipment which influenced component efficiency had substantial economic leverage in that they impacted the sizing requirements for the energy collection and thermal storage equipment.

The major design activities for the commercial system occurred during two periods of the 2-year contract effort. The initial system design was developed during the first 6 weeks of the contract to support the definition of the preliminary baseline design for the pilot plant. The second, and more extensive design effort, occurred from Month 12 to Month 20 to support detailed pilot plant design evaluations and to develop detailed commercial system cost estimates.

2.1 REQUIREMENTS

The basis of the commercial system design effort was a series of requirements which evolved over the 2-year contract period. These requirements can be subdivided into general program guidelines; performance/design requirements and environmental/site related requirements. These guidelines and requirements are summarized in Tables 2-1 through 2-3 respectively as they existed at the end of the study.

The general design guidelines imposed by DoE (Table 2-1) represent a series of overall objectives as well as specific design requirements. The basic objective for the solar plant is to be a meaningful contributor of electricity to a utility grid while being as cost-competitive as possible with alternate sources of electricity. The system itself is to produce electricity exclusively from solar energy without an on-site fossil-fueled supplement. In addition, the system shall employ water/steam as the primary operating fluid while all heat rejection shall be accomplished with wet cooling. Finally, the system must be compatible with the local environment and must comply with all applicable codes and regulations.

Table 2-1

COMMERCIAL SYSTEM DESIGN GUIDELINES

-
- Solar-only plant.
 - System must provide electrical power to a utility grid.
 - System shall be compatible with the utility demand characteristics to the greatest extent practical.
 - Heat rejection must be accomplished with wet cooling.
 - Water/steam shall be the principal working fluid.
 - System must be compatible with the environment.
 - System must be economically competitive with alternative commercial sources of electricity.
 - System must be compatible with all applicable codes and regulations.
-

Table 2-2
PERFORMANCE/DESIGN REQUIREMENTS

		Source
Design-Point Power Level		
● From Receiver (Best sun angle 950 w/m ² insolation)	100 MWe net	DoE
● From Thermal Storage	70 MWe net	DoE
Solar Multiple = $\frac{\text{max power collected}}{\text{design-pt power to turbine}}$	1.7	MDAC
Hours of Storage	6	MDAC
System Start-up Times		
● Hot	20 min ⁽¹⁾	DoE
● Cold	6 hr	DoE
Plant Availability (Exclusive of sunshine)	90%	DoE
Operational Lifetime (With normal maintenance)	30 yr	DoE
⁽¹⁾ Minimize within practical limits		

Specific performance and design requirements, which are summarized in Table 2-2, are subdivided into DoE-imposed and MDAC-developed requirements. The MDAC-developed items were arrived at as a result of system cost and performance trade studies which were aimed at minimizing the cost of energy as restricted by operational considerations. The specific trade studies that were carried out to properly size the collector field capability and thermal storage capacity resulted in a collector field solar multiple of 1.7 for a 6-hour thermal storage capability. Details of these trade studies are summarized in the next section of this report.

Table 2-3

MAJOR ENVIRONMENTAL AND SITE-RELATED REQUIREMENTS

			Source
Temperature			
● Design Point	28°C (82.6°F) Dry Bulb		DoE
	23°C (74°F) Wet Bulb		DoE
● Survive	-30°C - +50°C (-20°F - +120°F)		DoE
Wind Condition			
● Max Operational with Gusts	16 m/sec (36 mph)		MDAC
● Max Survival			
Sustained (Tower only)	40 m/sec (90 mph)		MDAC
With Gusts (Other Subsystems)	40 m/sec (90 mph)		DoE
Seismological			
	Seismic Zone 3		DoE
	NRC Reg Guide 1.60		DoE
	Response Spectrum		
	OBE ⁽¹⁾ - 0.165 horiz. "G"		MDAC
	SSE ⁽²⁾ - 0.33 horiz. "G"		MDAC
Soil Conditions			
	Barstow Data		DoE

(1) Operational Basis Earthquake (Uninterrupted Operation)

(2) Safe Shutdown Earthquake

The DoE sizing guidelines require that the system be capable of producing 100 MWe net power operating exclusively from receiver steam when the field has its maximum cosine value and with an insolation level of 950 W/m². For the MDAC system, which employs a 360-degree collector field (completely surrounds the tower), this design point corresponds to equinox noon. Field performance falls off as either winter or summer noon is approached.

because of field cosine effects. The system must also be capable of producing at least 70 MWe net power when operating exclusively from thermal storage. From an operational standpoint, the system was to be capable of a 20-minute hot system startup and a 6-hour cold startup; however, the 20-minute requirement exceeded the capability of all steam turbine equipment, which forced the hot-start requirement to be relaxed. The indicated availability and lifetime requirements were used as inputs to the design effort in the areas of component redundancy and maintenance philosophy.

The environmental and site-related requirements summarized in Table 2-3 are indicative of a southwest desert location. Again, the source of the requirement is indicated, with MDAC requirements being those developed as the result of cost and performance trade studies based on available data. The indicated wind conditions are categorized into operational and survival levels. The operational value is the point at which heliostat tracking would be terminated and the heliostats would be commanded to a minimum-damage orientation. This condition also serves as the design point for wind-induced tower sway. The survival wind corresponds to a maximum 50-year wind. A higher wind condition (100-year occurrence) was assumed for the tower for additional conservatism. From a seismic standpoint, the indicated intensities for uninterrupted operation and safe shutdown are consistent with the NRC 1.60 Response Spectrum and a Seismic Zone 3. All design calculations on this contract assumed a simultaneous vertical acceleration at $2/3$ the indicated horizontal level.

2.2 DESIGN RATIONALE AND EVOLUTION

The starting point for the design of the total system was to identify a commercially available turbine which had a high cycle efficiency and was capable of operating from two independent steam sources. Due to the difficulty in producing reheat steam with a central receiver, a turbine design compatible with a nonreheat cycle was also a requirement. The turbine which was selected is a tandem compound, double-flow, automatic admission design with 50.8-cm (20-inch) last-stage buckets. A cutaway view of the turbine without the admission port is shown in Figure 2-1. The admission port would be located $\sim 1/3$ of the way along the high-pressure section (left-hand pressure shell) with its location being precisely specified so that the local internal turbine pressure exactly matches the steam pressure available from

Figure 2-1. 100-MW Commercial Turbine

thermal storage. The turbine generator unit is rated to produce 112 MW of electrical power with a turbine inlet steam condition of 510°C (950°F), 10.1 MPa (1465 psia) and an exhaust pressure of 2.5 in Hg. The cycle efficiency which occurs at this operating condition is 0.376, including generator losses.

The turbine inlet steam conditions were arrived at by comparing permissible conditions with restrictions that would be imposed by the balance of the system. General Electric, who supplies the turbine, indicated that inlet steam temperatures could be specified between $\sim 482^{\circ}$ and 538°C ($\sim 900^{\circ}$ - 1000°F), while the pressure could be specified somewhere in the range of 8.72 - 10.1 MPa (1265 - 1465 psia). The steam temperature was selected by considering limitations of the receiver and piping as well as those of the turbine. In order to ensure that the maximum steam temperature remains below 538°C (1000°F) at all points in the system, where material strength properties experience a dramatic reduction, the receiver outlet steam temperature design point was selected at 516°C (960°F). This condition is sufficient to produce 510°C (950°F) steam at the turbine inlet. The selected steam

pressure, which corresponds to the maximum level for the turbine, was selected to enhance turbine cycle efficiency and improve the charging potential for thermal storage. This latter point is more fully treated in conjunction with the thermal storage charging and discharging operations in the following paragraphs. With the turbine conditions and cycle efficiency established, it was possible to specify many of the critical temperature, pressure, and flowrate conditions throughout the balance of the system.

In selecting the preferred approach to thermal storage, several key system issues were addressed. First, from an overall requirement standpoint, the thermal storage must serve as a source of steam of sufficiently high temperature and pressure to allow the turbine-generator to produce at least 70 MWe net when operating exclusively from thermal storage steam. In addition, this generation rate should occur with approximately the same turbine flow as would occur during design-point operation directly from receiver steam. This eliminates the need to oversize the turbine for thermal storage operation in an off-design flow condition occurring whenever the turbine operated exclusively from receiver steam. Once a thermal storage approach was determined which satisfied these minimum design and performance conditions, other options to thermal storage were considered that increased performance at an increase in cost. Before a final selection was made, issues such as simplicity and the impact on water/steam loop controllability were also considered.

The baseline thermal storage approach which was originally proposed is shown schematically in Figure 2-2 along with the charging and discharging side water/steam conditions. The approach represents a minimum-cost, moderate-temperature approach to thermal storage which employs a Caloria and rock mixture as the storage media. Extensive discussions were carried out with General Electric to determine the suitability of the resulting steam conditions with the 70-MW turbine operating requirement. It was determined that these steam conditions were satisfactory for the generation of 70 MWe net power at a turbine steam flow rate very similar to the flow rate experienced during design-point operation directly from receiver steam.

Figure 2-2. Baseline Commercial Thermal Storage Concept

A series of higher temperature thermal storage options were considered in an effort to improve turbine cycle efficiency when operating on admission steam. These options all replaced the desuperheating function that was necessary to protect the temperature-limited Caloria with a separate high-temperature storage loop which would be used to significantly increase the superheat level of steam produced in the steam generator.

The results of the comparative evaluation are shown in Figure 2-3. The first case represents the baseline design which is capable of producing steam at a temperature of 296°C (565°F). The rest of the cases all include a Hitec topping loop and assume a thermocline approach to energy storage. For Cases 3 and 5, a Hitec/rock mixture was assumed for the upper storage loop. The results, expressed on a relative basis, indicate that the baseline design approach using the moderate-temperature storage approach is superior to all but the third case. This means that the storage subsystem cost associated with achieving higher steam temperatures and higher cycle efficiencies more than offset the benefit to be derived through an increase in annual energy output. The overall superiority of Case 3 is expected since it preserves the attractive Caloria/rock feature of the baseline design but also adds a cost-effective Hitec/rock topping cycle. Due to unknowns related to

Figure 2-3. Comparison of Thermal Storage Concepts

Hitec/rock compatibility, as well as Hitec solidification issues in a rock bed, this case was not selected as the baseline design approach. Additional subsystem development work to verify this approach appears to be attractive. As a result of the turbine compatibility issues and economic factors just presented, the moderate temperature approach to thermal storage was selected. This approach is based on the use of Caloria/rock mixture which employs a thermocline as the method of storage. The Caloria will be exercised over a temperature range from 232° to 316°C (450°-600°F).

The thermodynamic operation of the thermal storage is illustrated in Figure 2-4. The horizontal distance between the charging steam flow and the Caloria line represents the local temperature potential for heat transfer during charging operation, while the horizontal distance between the Caloria and the discharge flow represents the potential for heat transfer during discharge. In order to ensure proper operation, a positive temperature potential must exist at all points through the heat exchangers. This means that the lines on this diagram may not intersect. The pinch points (locations of minimum temperature potential), which occur on the charging and discharging

Figure 2-4. Thermal Storage Charging and Discharging Characteristics (Commercial System)

sides, are points that limit the operating temperature range of the Caloria. The temperature and pressure limitations that exist for the discharge steam are also shown. An increase in the steam temperature would impose a problem at the outlet end of the steam generation equipment (top of diagram), while an increase in pressure would cause the vertical line to shift to the right, restricting the pinch point. Physically, this would impose a problem in the feedwater-preheating heat exchanger. Including line losses, the resulting steam conditions at the turbine admission port are 296°C (565°F), 2.52 MPa (365 psia). The location of the admission port on the turbine is matched to this pressure condition.

A simplified plant schematic which summarizes the overall design of the water/steam loop is shown in Figure 2-5. This figure illustrates the inter-relationships between the receiver, thermal storage, and the two-port turbine. Also shown in the figure are the five stages of feedwater heating that use extraction steam from the turbine and the principal feedwater pumping stations.

Figure 2-5. Simplified Commercial System Schematic

In order to size the collector field, receiver, and water circulation equipment, it was necessary to define the desired system capacity. Factors influencing system capacity are the turbine generator rating and thermal storage capacity. Since the rating of the turbine generator was fixed by DoE, the remaining factor influencing collector and receiver size is thermal storage capacity. The results of an economic trade study which shows the impact of collector field size on the cost of energy are given in Figure 2-6 for a stand-alone solar plant. The solar multiple parameter is the ratio of the maximum power collection capability (occurs at equinox noon) divided by the thermal power required to operate the turbine at its nameplate rating. This figure shows the optimum field sizes for two assumed thermal storage capacities. The figure also illustrates the economic coupling effect that exists between the thermal storage capacity and the sizing of the energy collection equipment. Solar multiples to the right of a minimum-cost condition result in a situation where more power is collected than can be processed through the turbine and thermal storage, thus producing a situation where thermal power must be thrown away. Moving to the left of the minimum-cost solar

Figure 2-6. Relative Cost of Annual Energy

solar multiple results in an under-utilization of the thermal storage because of insufficient collection of thermal power. The point selected as the MDAC design condition was a solar multiple of 1.7 with 6 hours of storage. Although this condition is slightly more expensive than the indicated minimum for 3-hour storage, it was selected to provide sufficient capacity to meet most utility's early-evening demands, particularly during winter periods when energy collection ceases at 3 to 4 PM.

A summary of the energy collection and utilization by the turbine and thermal storage equipment is presented in Table 2-4 for representative days of the year. The Table subdivides the energy collected during the day into the portion fed directly to the turbine and the excess energy that goes to charging thermal storage. For summer periods, the excess energy exceeds that necessary to fill thermal storage while during winter periods, an underfill condition exists. This is a natural result that would be expected following an optimization analysis. The net electrical energy produced on an annual basis from this system was estimated to be 425,000 MWh. This value includes consideration of all plant parasitic loads. It also assumes a

Table 2-4

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**SYSTEM PERFORMANCE SUMMARY
(SOLAR MULTIPLE = 1.7, 6-HR STORAGE)**

DAY	TOTAL COLLECTION CAPABILITY (MWHt)	DIRECT TURBINE OPERATION (100 MWe)		EXCESS ENERGY (MWHt)	ENERGY TO STORAGE (MWHt)	PERIOD OF OPERATION FROM STORAGE 70 MWe (HR)	SPILLAGE	
		REQD ENERGY (MWHt)	PERIOD (HR)				MWHt	(%)
JUN 21	5,290	3,485	11.7	1,805	1,757	6.0	48	0.9
MAY 21/ JUL 21	5,202	3,396	11.4	1,806	1,757	6.0	49	0.9
APR 21/ AUG 21	4,905	3,158	10.6	1,747	1,747	5.96	0	0
EQUINOX	4,422	2,860	9.6	1,562	1,562	5.33	0	0
FEB 21/ OCT 21	3,731	2,443	8.2	1,288	1,288	4.40	0	0
JAN 21/ NOV 21	3,122	2,115	7.1	1,007	1,007	3.44	0	0
DEC 21	2,832	1,966	6.6	866	866	2.96	0	0

950 W/m² steady daytime insolation and a downtime of 35 days per year, equally distributed throughout that period. In order to increase the duration of operation from thermal storage during winter periods over that shown in the table, the start-up of the turbine would have to be deferred or the turbine would have to be operated at part power to increase the flow to thermal storage.

Implicit in the previously discussed optimization analysis are the energy collection characteristics of a baseline collector field configuration. The collector field definition, which involves all aspects of the collector-receiver optical interface, was determined as a result of an extensive cost and performance analysis carried out by the University of Houston. The goal of the optimization was to determine the field configuration and heliostat spacing that produces the lowest cost of energy on an annual basis. In order to carry out the required analysis, cost and performance models were assumed for the collector subsystem, receiver, and tower. These models were based on component, assembly, and subsystem cost and performance data devel-

oped during the course of this contract. The type of data required included:

Shading and blocking of adjacent heliostats

Guidance error model

- Slope errors of reflectors
- Surface waviness
- Aim errors of beam sensors (if included)
- Tracking errors

Abberation model for canted or focused heliostats

Heliostat aim strategy

Cost model

- Heliostats
- Tower
- Receiver
- Piping in tower
- Land for heliostats
- Wiring and trenching for heliostats

Energy loss model

- Mirror reflection and receiver absorption
- Absorptivity versus angle of incidence on receiver
- Reradiation and convection from receiver
- Atmospheric attenuation between heliostat and receiver
- Interception losses at the receiver

The optimization activity starts by breaking down a large collector field, which includes the tower and receiver, into a series of computational cells (typically an 11 by 11 matrix). A representative heliostat is placed at the center of each cell and serves as the basis for evaluating all heliostats in the cell. The optimization procedure goes through a series of steps which begin by determining an ideal figure of merit for each of the test heliostats, assuming no neighbors exist, and then modifies the figures of merit to reflect the influence of neighboring heliostats; this influence is a function of heliostat arrangement and packing density. The analysis is carried out in an iterative fashion, with the initial inputs for the program being estimated heliostat densities for each cell and an initial estimated figure of merit. The computation proceeds by adding or subtracting glass to each cell as dictated by the assumed figure of merit, ordering the cells sequentially by decreasing

performance, calculating a new figure of merit, and trimming away ineffective cells from the field. Using the revised figure of merit and glass coverage densities for each cell as inputs, the process is repeated on an iterative basis until convergence is reached.

The data produced as a result of this procedure enables the size, shape, and critical geometric parameters of the collector-receiver interface to be specified, and determines the heat flux environment for the receiver. The collector field layout along with critical receiver, tower, and heliostat data are shown in Figure 2-7. The sawtooth field outline results from the 11 by 11 computational matrix used by the University of Houston. In reality, the field outline would follow a smooth continuous curve as illustrated by the outer dashed line. The heliostats themselves would be arranged along circular arcs with heliostats in alternate arcs being staggered to increase the optical window to the receiver. The heliostat coverage fraction varies from ~ 43% near the central exclusion area to ~ 14% along the northern perimeter with a field average coverage fraction of 23%. The total number of heliostats required is 22,914.

CR76

Figure 2-7. Commercial System Field Layout

The heat-flux environment experienced by the receiver along with basic receiver design information is summarized in Figure 2-8. The cylindrical receiver, which is composed of 24 independent panels, experiences a maximum incident power level of 560 MWt. The corresponding maximum power level absorbed into the water/steam is 506.4 MWt when operating at a 516°C (960°F) outlet temperature. When operating the receiver at a derated condition [368°C (694°F) outlet temperature], which is sufficient to charge thermal storage, the design-point absorbed power level is 285 MWt. In order to distribute the heat flux over the surface of the receiver, a three-point vertical aim strategy was adopted. With this strategy, alternate heliostats throughout the field are aimed at the equator and ± 6 m from the equator. The resulting incident heat flux profile for the northern panel (panel receiving maximum power) is shown in the lower right-hand side of the figure. The peak heat flux using the three-aim-point strategy corresponds to $\sim 0.85 \text{ MW/m}^2$.

CR76

- EXTERNAL CONFIGURATION
 - 24 PANELS (4 PREHEAT PANELS ON SOUTH SIDE)
- DIMENSIONS – 17M (55.8 FT) DIAMETER
25.5M (83.6 FT) LENGTH
- 3 AIM POINTS – RECEIVER MIDPOINT
MIDPOINT + 6M
MIDPOINT – 6M
- OPERATING CONDITIONS
 - MAX INCIDENT POWER – 560 MWt
 - MAX ABSORBED POWER – 506.4 MWt (RATED STEAM)
– 285 MWt (DERATED STEAM)
- RECEIVER HEAT FLUX PROFILE (NORTH PANEL – EQUINOX NOON)

Figure 2-8. Commercial System Receiver

A detailed breakdown of the commercial system performance is presented in a waterfall format for the equinox noon design point in Figure 2-9. Included in the figure is a detailed tabulation of the appropriate step-by-step efficiencies. For the design case, a total of 810 MW of thermal power is assumed to strike the heliostat surfaces if they are oriented normal to the incident sunlight. Implicit in this value is a conservative heliostat outage assumption of ~ 2%. If the full field of 22,914 heliostats were operational, the solar power striking the heliostats oriented normal to the sun would be 826.1 MW. This would require a corresponding adjustment to be made in the various waterfall levels shown. It should also be noted that in determining overall system efficiency, care must be exercised to subtract the power flow to thermal storage before any overall efficiency is calculated based on the power levels shown.

CR76

Figure 2-9. Commercial System Power Flow (Equinox Noon)

Section 3

PILOT PLANT SYSTEM

The stated purpose of the pilot plant is to provide a technical verification of the commercial system and an early indication of system economics – the first task that was addressed in designing the pilot plant related to the extent of verification required along with the potential cost impacts. The definition effort was initiated by identifying the elements of the commercial system that required verification and ordering them in terms of decreasing importance. After careful evaluation, it was determined that the design and operational aspects associated with the water/steam loop would be preserved as completely as possible as would be the general geometric characteristics of the collector field and receiver. This latter objective ensured the verification of the collector/receiver interface from an operational standpoint. In addition, the decision was made to preserve the full-size commercial heliostats, since they comprise the most expensive subsystem. Things that were compromised included peak receiver heat flux, the commercial solar multiple of 1.7 and the 6-hour thermal storage capacity – the first of which would be verified during subscale receiver heat flux test programs. In all cases, these design objectives were combined with the DoE-imposed pilot plant requirements to provide the overall guidelines for the overall pilot plant design effort.

3.1 REQUIREMENTS

The major requirements which serve as the foundation for the pilot plant design effort are summarized in Table 3-1. The principal sizing requirement is to be capable of producing 10 MWe of net electrical power at 2 PM on the worst cosine day, which is winter solstice for the MDAC system, with an insolation level of 950 W/m^2 . The system shall be capable of producing at least 7 MW of net electrical power during turbine operation exclusively from thermal storage for a period of up to 3 hours. In an effort to minimize the cost of the pilot plant, the design approach selected to satisfy these two

Table 3-1
Pilot Plant System Requirements

• DESIGN POINT POWER LEVEL RECEIVER (2 PM, WORST COSINE DAY) THERMAL STORAGE	10 MWe NET 7 MWt NET
• DESIGN INSOLATION	950 W/m ²
• SOLAR MULTIPLE	1.1
• THERMAL STORAGE CAPACITY	3 HR
• MAXIMUM THERMAL STORAGE CHARGING RATE	30 MWt
• SYSTEM STARTUP TIMES	
• HOT	20 MIN*
• COLD	6 HR
• SYSTEM AVAILABILITY	90%
• ELECTRICAL OUTPUT	COMPATIBLE WITH SCE GRID

*MINIMIZE WITHIN PRACTICAL LIMITS

requirements was to size the system to produce 10 MW of net electrical power at 2 PM on winter solstice with no excess thermal power available at that time to charge thermal storage. If it is desired to meet the turbine operating requirement from thermal storage, a delay in the daily turbine startup would be necessary to ensure that sufficient energy would be directed to thermal storage to fully charge the unit. The resulting collector field size necessary to satisfy this requirement has a solar multiple of 1.1 when measured at equinox noon.

The thermal storage charging requirement of 30 MWt which appears in Table 3-1 corresponds to the minimum design value specified by DoE. This minimum value was selected as a design upper limit to minimize the cost of the pilot plant system. Sizing the charging equipment to that value would allow the thermal storage unit to accept more than 85% of the maximum collector field output.

The startup and system availability requirements specified in Table 3-1 are identical to those discussed in Table 2-2 for the commercial system. Since the pilot plant will be part of the Southern California Edison network, the system output must be compatible with the SCE grid. Additional pilot plant requirements related to environmental factors are shown in Table 2-3 and will not be treated further in this discussion. It should be pointed out that for the most part, these environmental requirements represent typical locations in the southwest desert. Since Barstow, California has been selected for the pilot plant site, it is anticipated that some of those environmental factors may be modified to reflect the specific characteristics of Barstow.

Additional pilot plant requirements which have been imposed by MDAC call for the development of a computer-assisted control capability to aid in system startup, mode transitions, and shutdown. This was done in recognition of the unique operating nature of a solar electric system (it must react to changes in input power as opposed to controlling the input power as done in conventional plants). In addition, the computer-control capability is an absolute necessity to command the large number of heliostats in a coordinated fashion. Also, weather factors must be continually analyzed in order to anticipate weather-induced changes in system operation and aid in making operating mode selections.

3.2 WATER/STEAM LOOP DEFINITION

As in the case of the commercial system, the starting point for the definition of the pilot plant water/steam loop was the turbine selection. An existing family of turbines produced by General Electric, capable of operating from two steam sources (separately or simultaneously), was identified. A cutaway view of a typical turbine showing the two steam inlet ports is shown in Figure 3-1. The selected design-point steam conditions of 510°C (950°F) and 10.1 MPa (1465 psia) are identical to those selected for the commercial turbine, thus providing a good verification of the commercial system water/steam loop.

The selected turbine has a nameplate rating of 12.5 MWe which provides sufficient margin to generate 10 MWe of net electrical power in addition to providing the necessary power for plant parasitics. In addition, the turbine is capable of continuous operation at 10% over capacity. This allows the

Figure 3-1. 10-MW Industrial Turbine

turbine to accommodate 100% of the collected power throughout the year. The cycle efficiency when operated at nameplate rating and 2.5 in Hg is 0.347 including generator losses. When degrading the power to 11.2 MWe (required to provide 10 MWe for sale to the grid), the efficiency degrades slightly to 0.345. A heat balance for the turbine, employing four feedwater heaters is shown in Figure 3-2 for the winter 2 PM design point. In general, as the flow to the turbine increases, the stage pressure increases resulting in a corresponding rise in return feedwater temperature to the receiver. Reductions in turbine flow, on the other hand, reduce the stage pressures, which causes a reduction in feedwater temperature leaving the feedwater heaters.

In order to complete the definition of the water/steam loop characteristics, the thermal storage operating conditions must be identified. The selected operating temperature range for the Caloria and rock mixture, plus the charging and discharging functions are shown in Figure 3-3. In comparing these conditions to the corresponding commercial system conditions two important changes are noted. First, the Caloria HT-43 temperature is reduced by 14°C (25°F) over its entire operating range in the pilot plant for

Figure 3-2. Pilot Plant Heat Balance (Winter 2 PM)

Figure 3.3. Thermal Storage Charging and Discharging Characteristics (Pilot Plant)

reasons of design conservatism. The effect of this change is to relax the pinch point (the minimum horizontal distance between the charging steam line and the thermal storage fluid line) on the charging side, while tightening the pinch point on the discharge side. Second, the maximum discharge steam temperature for the pilot plant thermal storage configuration is 274°C (525°F). Based on turbine design considerations, the maximum admission steam pressure acceptable for this temperature is 2.65 MPa (370 psig) at the admission port, or 2.76 MPa (385 psig) at the steam generator. It should be noted that the admission steam for the pilot plant has 17°C (30°F) less superheat than the admission steam for the commercial system. The result is a greater moisture level at the last-stage buckets for the pilot plant turbine. This is acceptable due to the smaller bucket wheel diameter and corresponding lower bucket tip speed in the pilot plant turbine, which results in a reduced tendency for moisture erosion.

The effect of operating the turbine exclusively from thermal-storage-generated steam is shown in the heat balance diagram depicted in Figure 3-4. The steam flow is adjusted to produce an electrical output of 7.8 MW which is sufficient to provide 7 MWe to the grid. During this operating mode, the two high-pressure heaters are deactivated, with the boost pump downstream of the second heater supplying flow directly to the thermal storage steam generation equipment. The gross cycle efficiency for this mode of operation is 0.244 which includes the effect of generator losses.

A simplified version of the overall water/steam loop is shown in schematic form in Figure 3-5. This figure depicts the major components that make up the water/steam loop, as well as the interrelationships between the receiver, thermal storage, and balance of plant. The loop is operationally similar to the commercial version shown in Figure 2-5 except for some changes in the routing and pumping of the feedwater. The flash tanks which are shown in the figure are used any time a liquid or two-phase flow condition exists immediately upstream. In the case of the receiver flash tank, which would be mounted at the top of the tower and is schematically shown in the upper left corner of the figure, it would be activated during receiver start-up or shutdown as well as during periods when cloud passage produces a condition

Figure 3-4. Pilot Plant Heat Balance (Admission System Only)

Figure 3-5. Simplified Pilot Plant System Schematic

of two-phase receiver flow. The thermal storage flash tank is used to accommodate the high-temperature condensate that leaves the thermal storage heater. In both cases, the discharge from the tanks (either in the form of hot water or low-grade steam) is preferentially routed to the feed-water heaters to retain the heat in the loop, with backup lines being connected to the condenser where the heat would be rejected. The flow arrows shown in Figure 3-5 indicate all possible flows that may exist. In reality, only certain combinations of flow paths would be used depending on the operating mode selected. These modes will be treated later in this discussion.

3.3 COLLECTOR FIELD/RECEIVER DESIGN

In arriving at the final collector field/receiver configuration, a series of design alternatives were considered. In general, these alternatives involved the tradeoff between degree of commercial system simulation and cost. The optimization studies that were carried out to define the commercial system were based on a series of subsystem, assembly, and component level assumptions. Implicit in these cost models were assumptions concerning mass production of heliostats, local or regional manufacturing and assembly sites, cost reductions due to well established learning curve effects, etc. As a result, these studies assumed relatively inexpensive heliostats, which when included in the optimization analysis produced a 360-degree collector field that completely surrounded the tower. This meant that it was economically desirable to place some of the heliostats into the southern field even though they will experience relatively poorer performance than the balance of the system. This would imply that for a truly scaled version of the commercial system, a southern field should also be included in the pilot plant design.

By contrast, however, if the optimization analysis were carried out exclusively for a 10-MW pilot plant, a different set of component and subsystem cost assumptions would be appropriate. In particular, the biggest difference would be the loss of many of the mass production and learning curve assumptions which were made for the commercial system. The most significant effect of this change would occur in the heliostat costs. Since they would be a much more expensive part of the system on a percentage basis than for the commercial system, the optimization would dictate that they be placed in

only the highest efficiency areas of the field relative to the tower, i. e., the north field. Thus, a paradox exists in the design of the pilot plant. If economy is the overriding goal, a pilot plant with a north-only collector field is appropriate, while if the overriding goal is to simulate the ultimate commercial system, a full 360-degree collector field is dictated.

In addition to the 360-degree collector field, which was ultimately selected for the pilot plant, two alternative north-field designs were also considered along with appropriate receiver configurations. To give design visibility, one of the alternate fields was designed to minimize investment cost by sizing exclusively to the winter 2 PM design point. The second field was defined to minimize the cost of energy on an annual basis.

The first of these fields is shown in Figure 3-6 along with the approximate field dimensions, tower height, and receiver centerline elevation. In contrast to the 360-degree collector field, the tower in this case is well to the south of the collector field. Since the system was designed exclusively for a winter 2 PM sun condition, the resulting collector field is fairly narrow while the receiver is located on top of a relatively tall tower. The receiver configuration defined for this solar collector field was cylindrical with the 13 northernmost panels remaining as single-pass-to-superheat panels while the balance of the 24 (on the southern side) were replaced with sheet metal. This assumption was made to preserve aerodynamic symmetry.

The second alternate field configuration, which was designed to optimize annual energy, is shown in Figure 3-7. Since it was designed for an annual average sun condition, which has a moderately high elevation angle and moves between the east and west sky, the resulting collector field is wider with less depth in comparison to the winter optimum design. In addition, the higher average sun angle permits the use of a shorter tower. As in the previous case, a cylindrical receiver with 13 active northern panels was assumed. In both of the cases considered, a heliostat cost of \$300/m³ was assumed, which is representative of early heliostat costs prior to the implementation of large mass-production facilities.

TYPICAL FIELD

ASSUMED HELIOSTAT
COSTS - \$300/M²TOWER HEIGHT
132 MRECEIVER MIDPOINT
147 M

Figure 3-6. Winter Optimum Field Layout

TYPICAL FIELD

ASSUMED HELIOSTAT
COSTS - \$300/M²TOWER HEIGHT
106 MRECEIVER MIDPOINT
121 M

Figure 3-7. Optimum Annual Energy Field Layout

The results of a cost and performance comparison carried out between the two north field configurations and the 360-degree collector field design is summarized in the following tabulation.

Configuration	Relative Cost of Collector and Receiver Subsystems	Relative Annual Energy Production
360-Degree Field	1.0	1.0
Winter Optimum	0.92	0.92
Optimized for Annual Energy	0.93	0.97

Based on these data, the decision was made to adopt the 360-degree collector field configuration, which provides the most complete simulation of the commercial system, and accept the 7 to 8% penalty in collector and receiver equipment cost. Important simulation aspects gained by adopting the 360-degree collector field include the operational verification of heliostats at all possible relative locations anticipated for the commercial system, the control of a full 360-degree receiver, and interactions between southside receiver preheat panels and receiver boiler panels located over the rest of the receiver.

The final pilot plant collector field configuration, which represents a scaled-down version of the commercial system collector field, is shown to scale in Figure 3-8. The field is composed of 1760 heliostats with each of the heliostat symbols corresponding to the exclusion area required for the heliostat. Also shown is the arrangement of the power house and auxiliaries in the central exclusion area. The radial stagger nature of the collector field preserved the nature of the commercial field shown in Figure 2-7. The collector field requires $3.04 \times 10^5 \text{ m}^2$ (75 acres) of land with an additional $10,387 \text{ m}^2$ (2.6 acres) being used for the central exclusion area. The glass packing density varies from 45% near the central exclusion area to 13% at the northern perimeter. The average glass coverage factor is 23%. The elevated receiver placed at the center of the central exclusion area has a centerline elevation of 80 m (262 ft) above grade. The height of the

GENERAL NOTES

A. POWER HOUSE AND AUXILIARIES

- 1 RECEIVER TOWER AND FOUNDATION
- 2 POWER HOUSE
- 3 THERMAL STORAGE SUBSYSTEM
- 4 CONDENSATE STORAGE TANKS
- 5 ASSEMBLY, AS NOTED
- 6 DIESEL GENERATOR
- 7 WATER TREATING BLDG
- 8 ACID STORAGE TANK
- 9 LIME SOFTENER
- 10 CLEARWELL, COVERED
- 11 SEWAGE UNIT
- 12 AUX TRANSFORMER
- 13 MAIN TRANSFORMER
- 14 RECEIVER PANELS LAYDOWN AREA
- 15 POWER AND CONTROL CABLE TRENCH
- 16 BURIED PIPE LANE
- 17 TREATED WATER AND FIRE PUMPS
- 18 ROCK-FILLED CATCH BASIN
- 19 LUBE OIL STORAGE TANK
- 20 CONDENSATE STORAGE TANKS
- 21 FUEL OIL TANK

- B SOUTH ACCESS ROAD IS 11M (36 FT) WIDE
- C EAST, WEST, AND NORTH ACCESS ROADS ARE 5.5M (18FT) WIDE EACH
- D ⊗ • HELIOSTAT EXCLUSION CIRCLE

PREPARED BY COMPUTERIZED MAPPING		MDAC 10MW CENTRAL RECEIVER COLLECTOR FIELD LAYOUT	
MCDONNELL DOUGLAS CORPORATION		DATE 4-21-77	APPROVED BY

supporting tower is 65 m (213 ft), on top of which is placed the receiver structure.

An estimate of the collector field efficiency for the pilot plant is shown in Figure 3-9 for various sun locations. This efficiency data includes effects of field cosine, blocking/shadowing between adjacent heliostats, receiver interception factors, and sensor-post blocking and shadowing factors. A unity factor was assumed for both heliostat reflectivity and atmospheric attenuation. The data presented in Figure 3-9 should be multiplied by representative reflectivity and attenuation values to get a complete estimate of collector field efficiency. In sizing the pilot plant, a reflectivity of 0.88 and an atmospheric attenuation factor of 0.98 were assumed. In general, the collector efficiency shown for the pilot plant is ~ 2% higher than the corresponding data for the commercial system. This difference is accounted for by the higher receiver interception factor which occurs for the pilot plant system.

Variations in incident power over the surface of the cylindrical receiver created by the pilot plant collector field layout are shown in Figure 3-10 for various times of the year. The smooth curves were developed by connecting 24 discrete points, which represented the total power incident on each of the absorbing panels, with each of the panels accounting for 15 degrees of the total 360-degree circumference. The dominant effect of the collector field shape on the incident power causes the peak to occur at or near the north side of the receiver. The minimum heat flux always occurs on the south side of the receiver due to the fewer number of heliostats located to the south of the tower and the lower cosine values associated with these heliostats. Because of this low heat flux condition, which occurs on the south side of the receiver, the decision was made to design the six south side panels as preheat panels. The number was selected so that water entering the remaining 18 boiler panels would be at least 27.8°C (50°F) subcooled.

The system performance characteristics for the pilot plant are presented in Figure 3-11 for the winter 2 PM design point. The performance estimate

Figure 3-9. Collector Field Performance

Figure 3-10. Circumferential Power Distribution (Pilot Plant Receiver)

Figure 3-11. Pilot Plant System Power Flow (Winter 2 PM)

starts with the amount of power incident on the heliostats assuming they were all oriented normal to the sun. The power degrades on an incremental basis according to the tabulated efficiency factors shown in the upper right-hand portion of the figure, with the last bar representing the net electrical output to the grid. The total efficiency for the winter 2 PM design point is 16.3%. The corresponding efficiency which would occur at equinox noon (most favorable time) is 18.4%.

3.4 SYSTEM OPERATION

As part of the design effort for both the pilot plant and commercial system, a series of operating modes were defined which give maximum flexibility to the operation of the system and permit controlled operation during cloud-induced transients. The six basic modes which were defined for both systems are identified in Table 3-2 along with operating information for the receiver, thermal storage, and turbine equipment. The Rated Steam entry for the receiver refers to the design-point steam temperature of 516°C

Table 3-2. Operating Modes

MODE	RECEIVER OPERATION	THERMAL STORAGE OPERATION	TURBINE OPERATION
NORMAL SOLAR	RATED STEAM	CHARGING	THROTTLE STEAM
LOW SOLAR	RATED STEAM	DISCHARGING	THROTTLE + ADMISSION STEAM
INTERMITTENT CLOUD	RATED OR DERATED STEAM (OPERATOR CHOICE)	CHARGING/ DISCHARGING	ADMISSION STEAM
EXTENDED OPERATION	---	DISCHARGING	ADMISSION STEAM
CHARGING ONLY	DERATED STEAM	CHARGING	---
FULLY CHARGED THERMAL STORAGE	RATED STEAM	---	THROTTLE STEAM

(960°F), which is required any time the turbine is being operated directly from receiver steam. The derated entry corresponds to the receiver steam temperature which is sufficient to charge thermal storage [349°C (660°F) for the pilot plant]. The table indicates that in the low-solar mode, the turbine will be operated simultaneously from both steam sources, while during the intermittent-cloud mode, both the charging and discharging thermal storage loops are operating.

In order to facilitate the operation and control of the total system within the various operating modes as well as during periods of startup, shutdown, and mode transition, a master control subsystem has been developed and utilizes both manual and automatic control techniques. The design utilizes a full complement of manual controls and computer equipment to provide the operator and engineers with the flexible control and operational tools necessary to properly operate the system and carry out performance analyses. These factors are critical because of the reactive, operational nature of a solar plant as compared to a conventional plant, and the desire to maximize useful energy collected during the sunlight periods of the day. In addition, an automatic control system is required to coordinate the operation of a large number of discrete heliostats with the operation of the receiver and the rest of the water/steam loop.

Section 4

COLLECTOR SUBSYSTEM

4.1 INTRODUCTION

The function of the collector subsystem in a central receiver solar thermal power plant is to redirect and concentrate the sunlight on the absorber of the receiver subsystem. To accomplish this function, an array of two-axis tracking mirrors (heliostats) is employed. Each heliostat, with its control, tracks so that it reflects the sun to its assigned point on the absorber surface. The reflector mirror may be focused, flat with individual segments canted for focusing, or completely flat, depending on system requirements. The concentration comes partially from focusing, but primarily from the superimposition of images from many heliostats on the same area of the receiver.

4.2 HISTORICAL SUMMARY

The significant events, activities, and milestones for this program relating to the collector subsystem are shown on Figure 4-1.

During the first quarter following ATP (1 July 1975), the collector preliminary baseline design was established. The preliminary baseline design included establishing collector subsystem requirements, defining the heliostat and controls configurations, and performing sufficient analyses of the configuration to ensure that it would meet the requirements.

The next activity was the conceptual design of the collector subsystem research experiments (SRE). The PDBR requirements were modified to add test requirements as required. The baseline design was modified to ensure producibility at the SRE production level (five heliostats) with minimum tooling. The pilot plant requirements were analyzed and the requirements to be verified in the collector SRE identified. These requirements were collected into an integrated SRE test plan. The plan included control systems

Figure 4-1. Collector Subsystem Historical Summary

development and verification; verification of structural strength, deflection, and resonant frequency; environmental and life tests on the reflector, control sensors, and drive unit; and a heliostat array test in a desert environment.

Following the acceptance of the conceptual design by DoE, a detailed design of the collector SRE was prepared. The detailed design included detailed and assembly drawings for the collector, production plans, and detailed test plans. A series of informal design reviews was held, and following DoE approval, procurement, tooling, and fabrication activities were initiated.

As the program progressed, it became apparent that there was a significant, unresolved issue involving safety of the reflected beams from the heliostats. If the heliostats were stowed face up, as in the MDAC baseline, sunlight could reflect from the heliostats and cause local "hot spots" in the airspace surrounding the collector field. Moreover, during the transitions of the collector field from stowed to on-target modes and back, the possibility of regions of excessive light concentration outside the plant seemed to be

greater for face-up stowage than for face-down stowage. It seemed possible that these safety issues could be resolved by careful control of the heliostats in stowage and in transition between stowage and on target. However, it was also likely that DoE would want to specify face-down stowage capability for the pilot plant. MDAC, recognizing the importance of this issue, funded a design, fabrication, and test program for an alternate heliostat capable of inverted stowage. DoE supported this program by providing for design review and approval and technical monitoring of the design and tests.

Toward the middle of 1976, it became apparent that the first-surface mirrors designed for the pilot plant SRE were degrading due to inadequate protection of the silver by the acrylic coating. MDAC investigated and reported this problem to DoE. In October, DoE authorized MDAC to do additional work to develop second-surface mirrors and to improve the first-surface mirrors. While no solutions to the first-surface mirror problems were found, two technically acceptable second-surface mirror configurations were identified. Both laminated (mirror-to-glass) and foam core (mirror bonded to a foam core bonded to sheet steel) were fabricated and tested on the inverting heliostat.

The SRE are summarized in Table 4-1. In the structures tests, the reflector panels or segments were loaded with distributed weights to simulate the operational design point and survival wind loads. Strains and deflections were measured to verify requirements. The heliostats were excited to find the fundamental frequencies for vibration about the two drive axes.

Controls development tests were conducted to verify controller hardware and software, verify interfaces, and obtain tracking accuracy.

Environmental/life tests were conducted to verify critical elements of the heliostat and controls. Drive units for both heliostats were subjected to conditions of rain, icing, blowing dust, high and low temperatures, and rapid temperature change. Fatigue life was demonstrated through accelerated life tests. The mirrors were subjected to dust erosion, simulated hail impact, high and low temperatures, thermal cycling, chemical attack, and environmental exposure. Adhesives were subjected to fatigue, environmental exposure, high and low temperatures, creep, and moisture and chemical attack.

Table 4-1
SRE TEST SUMMARY

Test	Hardware
STRUCTURES	
Strength and deflection at survival wind	One complete octagonal heliostat
Deflection at operational wind load (26 mph)	One complete inverting heliostat
Fundamental frequency	Laminated-glass mirror
	Foam-core mirror
CONTROLS DEVELOPMENT	
	One octagonal heliostat and controller
	One inverting heliostat and controller
ENVIRONMENTAL/LIFE TESTS	
Rain, icing, dust, high and low temperature	One drive unit of each type with controls sensors
Dust erosion, hail impact, high and low temperature, environmental exposure, chemical attack	First-surface, laminated and foam-core mirrors
Fatigue life	Drive unit elements of both types (orbdrive, harmonic drive, and linear actuators)
Fatigue life, environmental exposure, high and low temperatures, creep, moisture and chemical attack	Urethane adhesives for laminated mirrors and foam-core panels
ARRAY TESTS	
Installation/maintenance procedures	Three octagonal heliostats with first surface reflectors
Checkout procedures	Two octagonal heliostats with laminated-glass mirrors
Tracking performance	One inverting heliostat with laminated- and foam-core mirrors
Beam errors	One field controller
Environmental effects	One simulated master control
Off-nominal operations	
Master control algorithms	
Cleaning	

An array of heliostats was installed and operated at the Naval Weapons Center, China Lake, California. Data were gathered on installation, maintenance, and checkout procedures. Tracking performance and beam error data were obtained. Environmental effects and operation during off nominal conditions were noted. Master control algorithms were verified. Data were obtained on mirror reflectivity histories, washing methods and their efficacy, and time to wash and quantity of materials.

ERDA also recognized the need for a greater definition of the commercial plant collector and the pilot plant collector production plans to provide a sound basis for the cost estimates on those two plants. Hence, MDAC was authorized in January, 1977 to perform further studies on the commercial plant collector and in February, 1977 to develop pilot plant collector production plans.

Concurrently, the preliminary design for the pilot plant was begun. The preliminary design consists of a baseline design updated by the results of the SRE and carried to a lower level of detail, analyses to ensure that the design will meet the requirements, and production plans, installation plans, and maintenance plans.

The preliminary design culminated in a preliminary design review and report in May of 1977. Subsequent to that review, minor changes to the control, drive unit, and reflector have been made and are described in this report.

The collector SRE met the verification requirements identified for Phase 1 of the program. A collector design has been demonstrated which meets the requirements of the pilot plant and has been verified by component or assembly tests or by similarity. The pilot plant collector is identical to the commercial plant collector in every respect except the number of heliostats, the degree of focusing, and changes for producibility in high volume which are yet to be defined.

4.3 DESIGN SUMMARY

4.3.1 Collector Subsystem Design Rationale

The heliostat employs flat, rectangular reflector panels canted in the pilot plant design to provide a degree of focusing. The reflector is mounted on a central pedestal and employs a rotary drive for azimuth and a linear actuator for elevation. The heliostat is arranged to provide for inverted or face-down stowage, using a second linear actuator.

Closed-loop control using a separately mounted beam sensor was selected for the pilot plant in Phase 1. An open-loop system was evaluated during the second half of 1977 and is now the recommended approach for heliostat control.

The principal rationale for the above design is:

- Flat reflector surfaces reduce cost and technical risk.
- Single-curved reflector panels may be used at a minimal cost-risk increase and achieve performance equal to that of double-curved panels.
- Foam-core panels provide rigidity and stability at low cost.
- Inverted stowage assures beam safety and reduces mirror soiling.
- Rotary/linear drives supply accurate, low-cost tracking throughout the required range of motion.
- Elevation/azimuth gimbals minimize survival (40 m/s) wind loads.
- Closed-loop control was baselined as a verified, accurate control system at low cost.
- Open-loop control offers a probable further cost reduction.

4.3.2 Collector Requirements Summary

The collector requirements include:

- Operating requirements
- Design requirements
- Environmental requirements
- Operating
- Survival
- Safety requirements

The operating requirements are a function of: (1) interfaces between the collector and the balance of the central receiver system, and (2) the operating modes.

Interfaces are defined by the relationship of the collector to the system. The collector reflects the sun's energy to the receiver. The receiver heats the transfer fluid and provides energy to operate the electrical power generation system and the storage units. The collector interfaces with the receiver, master control, and power distribution system. The collector interfaces with the receiver through the reflected beams from the heliostats. The heliostats track the sun and direct the reflected beams so that the light falls on the receiver with a prescribed flux distribution. Spillage of light missing the receiver is minimized, consistent with overall system economics. The collector interfaces through the field controller with the master control. Ephemeris data and collector operating mode commands are sent from master control to the field controller. Collector status and malfunctions are transmitted from the field controller to the master control. The collector receives power from the distribution system to operate the electric motors and control system.

The operating mode requirements are shown in Table 4-2. In the closed-loop system, the heliostats track the sun in the "normal tracking mode" by means of an optical sensor. The remaining modes of the closed-loop system and all the modes of an open-loop system utilize an output encoder/input shaft counter.

The collector design requirements are:

- Commercial and pilot plant collectors are the same
- Collector uses qualified components
- Collector is compatible with the receiver
- Heliostat is capable of inverted stowage
- Collectors have low cost commensurate with safety, availability, reliability, and ease of maintenance
- Heliostats have a rigid drive and structure to prevent excessive movement under gust conditions
- Reflectors are dimensionally stable with high optical efficiency

Table 4-2
OPERATING MODE REQUIREMENTS

Mode	Field Controller	Master Control
Normal Tracking	Autonomous	Monitor only
Synthetic Tracking	Track to master control requirements	Compute ephemeris or other tracking data
Command Position	Same	Direct position, e.g., maintenance, cleaning
Local Manual Control	Mobile test set controls	Locked out
Stowage	Stow to master control requirements	Select mode and command

The design operating environment is as follows:

- Design wind speeds (including gusts) are:
System performance at 3.5 m/s (7.8 mph) at 10 m elevation
Design deflection at 11.6 m/s (26 mph)
Stowage initiation speed at 16.1 m/s (36 mph)
- Temperature range = 0° to 49°C (32° to 120°F)
- Track the sun at low insolation levels (0.3 kW/m^2)
- 30-yr life

The survival environment is as follows;

- Maximum wind speed:
Stowed, elevation axis aligned with wind – 40.2 m/s (90 mph)
Any orientation – 22.4 m/s (50 mph)
- Ambient air temperature = -30° to 49°C (-22° to $+120^{\circ}\text{F}$)
- Earthquake – 0.25g peak ground acceleration
- Precipitation
250 Pa (5 psf) snow
50 mm (2 in) ice
100 mm (4 in) annual average rainfall
75 mm (3 in) maximum 24-hr rate
19 mm (3/4 in) hail, any orientation, at 20 m/s (65 ft/s)
25 mm (1 in) hail stowed, at 23 m/s (75 ft/s)

- Blowing dust devils, with wind speeds up to 17 m/s (40 mph)
- Thunderstorm gust fronts, with wind speeds up to 22 m/s (50 mph)

Safety requirements are consistent with OSHA, CAL-OSHA, National Electric Code, and Lightning Protection Code sections of the National Fire Protection Association, the American National Standards Institute Code No. ANSI-2, and the National Electric Safety Code. In addition, no unsafe beam conditions shall result which endanger personnel or equipment on the ground, either within the plant or outside of the plant or personnel in the airspace above the plant (altitude greater than 300 m (1000 feet) or in the surrounding airspace.

4.3.3 Collector Design Definition

Both the pilot plant and commercial plant collectors are described in this section. As previously mentioned, the two collectors are identical to the level of definition presently available except for number and focusing. The final heliostat and control system configurations represent a hybrid version of the configurations and components studied under this program and with MDAC funding. Because the final heliostat configuration is not the same as either the octagonal heliostat or the inverting heliostat, several candidates will be described as appropriate and then the final selection and reasons described.

4.3.3.1 Heliostat

Two heliostat configurations were designed, fabricated, and tested. The noninverting octagonal heliostats are shown in Figure 4-2. The octagonal heliostat assembly consists of eight trapezoidal reflectors supported by radial beams that tie into a hub. The reflectors included first-surface mirrors with acrylic coating on 6.35-mm (1/4-in) thick float glass and second-surface laminated mirrors. The reflectors and substructure are rotated in azimuth by a harmonic rotary drive and in elevation by a jack. The assembly is supported by a conical pedestal with a reinforced slab foundation. The sensor is supported by a sensor post and is connected to the field controller. The field controller is commanded by the master control.

In response to potential changes in the stowage requirements, MDAC designed, fabricated and tested two inverting orthogonal heliostats shown in Figure 4-3. The orthogonal heliostat assembly consists of six rectangular

Figure 4-2. Heliostat Array

Figure 4-3. Inverted Stowage Orthogonal Heliostat

reflector panels supported by a central torque tube and four cross beams. The panels tested included second-surface laminated mirrors and second-surface foam-sandwich mirrors. The reflectors and substructure are rotated in azimuth and elevation by Orbdrive rotary drives. The assembly is supported by a cylindrical pedestal and a concrete foundation. The sensor and the electronics are identical to the octagonal heliostats.

The inverted stowage heliostat has been baselined for the pilot plant.

4.3.3.2 Reflective Surface

Two types of reflective surface have been considered: first-surface mirrors and second-surface mirrors. The first-surface mirror is deposited on 6.35-mm (1/4-in) commercial float glass. The glass may have any iron content and need not be free of cosmetic defects. Hence, the basic glass would be cheaper than comparable thickness glass used for second-surface mirrors. The silver is chemically deposited in about an 800⁰A (95 ±10 mg/ft²) thickness layer to provide optimum reflectance.

Then a top acrylic coating of 26-micron (1.0-mil) thickness is laid on and cured. The resulting coating has a high reflectivity (0.90-0.93) and a high degree of specularity. However, environmental tests showed the coating to have pinhole defects which lead to attack of the silver and undercutting. In addition some coatings showed an inadequate adhesion to the silver and delaminated. Finally, the first-surface mirrors proved to have more rapid dust buildup and were more difficult to clean than second-surface mirrors.

An investigation of second-surface mirrors was undertaken to find a more durable, lower risk approach to the reflective surface. The second-surface mirrors are either laminated to glass or bonded to a foam core. The mirror is made of medium- to low-iron (0.07-0.04%Fe) float glass. The thickness is from 2.38 mm (0.093-in) to 3.18 mm (0.125 in). Reflectivity may range from 0.87 to 0.94, with the iron content being apparently the most important variable. Most commercial chemical silvering processes give comparable results to within experimental accuracy. However, the mirror backing paint must be chosen to be compatible with the adhesive to be used.

Low-iron (0.05% Fe) float glass second-surface mirrors have been selected for the pilot plant preliminary design.

4.3.3.3 Reflector Panel or Segment

Three candidate panels or segments were considered: a trapezoidal segment for the octagonal heliostat, and laminated and foam-core rectangular panels for the inverting heliostat. The trapezoidal segment is supported on roll-formed channel beams (Figure 4-4). Bathtub end fittings are used to attach the segment to the center hub. The ends of the fittings are machined perpendicular to the reflector surface and index pins are installed parallel to the reflector surface. The attach holes are located for interchangeable clearance fit with respect to the hub assembly. The frame assembly is assembled flat within 1.52 mm (0.060 in) to control bond thickness between the glass reflector and support structure. Plastic spacers are used between the steel frame and the glass plate to limit the minimum bond line thickness to 1.27 mm (0.050 in). The tolerances are controlled so that no fit up in the field is required to align the reflector segments. An assembled segment is shown in Figure 4-5.

The laminated-mirror panels for the inverting heliostat use a second-surface mirror laminated to a float glass back lite. The total thickness of the laminate is about 6.35 mm (1/4 in). The mirror is bonded to longitudinal hat-section stiffeners. The hat stiffeners for the reflector are roll-formed 76 mm (3-in) wide and 40 mm (1.56 in) deep made from No. 16-gage (0.060-in) hot-rolled steel in lengths of 2.84 m (112-in) and 2.57 m (101.25-in). In assembly, the stiffeners are bonded to the mirror on a surface table. The six attach points on the stiffeners are held in a plane parallel to the surface table during the bonding operation. Laminated glass panels are seen on the right-hand side of the reflector of the inverting heliostat in Figure 4-3.

The foam-core panels also use second-surface float glass mirrors. The reflector consists of six panels. Each panel subassembly (Figure 4-6) is made up of a second-surface silvered mirror, 3.24 mm (1/8 in) thick of medium-iron (0.05%) float glass adhesively bonded to a foam core, which is in turn bonded to a 26 gage galvanized steel back sheet. Each reflector subassembly panel yields a 2.16 x 2.9-m (85 x 114-in) in overall dimension, thus providing for a total heliostat reflective surface area of 38 m^2 (408.3 ft^2) including a tracking mirror. The mirror is commercial grade as is the foam-core material. Environmental protection of the reflector is provided

Figure 4-4. Reflector Segment

by treating the panel edges with a polyurethane weatherseal compound. Attachment of each panel to the support structure is accomplished by four shallow circular steel cups bonded to the galvanized steel back sheet. One will note that the reflector panels are identical in size except that the two upper panels have clipped corners. This feature is incorporated to maximize heliostat packing density and thus reduce the land use requirements. The foam-core mirror panels are shown on the left side of the heliostat in Figure 4-3.

Foam-core mirror panels have been selected as the preliminary design for the pilot plant. The panel width may be decreased and single curvature used to increase the interception factor, depending on receiver/collector interface requirements. The foam-core panels were selected on the basis of lower projected cost, better flatness, and ease of focusing.

REFLECTIVE SURFACE

Figure 4-5. Reflective Surface and Segment

SEGMENT BONDING

Figure 4-6. Reflector Panel Assembly

4.3.3.4 Reflector Support Structure

The octagonal heliostat uses a central hub to mount the reflector segments. Hub details are sketched in Figure 4-7. The hub assembly uses deep drawn upper and lower shells. The upper and lower face sheets are stiffened by beads formed in the pressing. The upper and lower rings are made from structural angles and machined after assembly to the required dimensions. The inner and outer support fittings are forged or cast. The lower shell, cross channels, lower ring, and internal fittings are assembled and joined by spot welding. The upper shell and upper ring are assembled as a unit, then joined to the lower assembly. The final assembly of the hub is made by joining the upper skin to the cross channels by means of rosette welds or rivets. Fusion or seam welding is used to attach the two shells. After assembly, the outer cylindrical surface of the rings is machined circular and perpendicular to the machined top surface. The key slot is machined parallel with the upper surface to provide accurate alignment for the reflector segments which bolt to the hub in field assembly. The attachment holes for the segments are drilled for an interchangeable clearance fit with respect to the segment hole pattern. The SRE hub is shown during assembly in Figure 4-8.

Figure 4-7. Ring-Stiffened Hub Assembly

Figure 4-8. Hub

The support structure for the inverting heliostat consists of a main torque tube and four channel cross beams. As depicted in Figure 4-9, the cross beams provide the structural support for six reflector panels. The cross beams are spot-welded to the main torque tube which, in turn, is bolted to the heliostat drive unit. The main torque tube is attached to the drive unit by two drive attachment fittings. Ring flanges are used to attach the cross beams to the main torque tube. The attach cups of the reflector panels bolt to the channel cross beams through punched or gang-drilled holes. The slot formed between each panel group provides clearance for the pedestal when the reflector is rotated to the inverted or face-down stowage position. The SRE heliostat support structure is also shown in Figure 4-3. The orthogonal support structure is selected for the pilot plant baseline as consistent with the desire to invert for stowage.

4.3.3.5 Drive Unit

The two heliostat configurations have different drive units because of differing requirements. Both heliostats incorporate azimuth and elevation drive mechanisms to produce movement of the reflective surface about its axes of rotation. The drive mechanisms are composed of motors, drive trains, position feedback transducers, limit switches, reflector-support bearings, and structural housings.

The octagonal heliostat drive unit is shown in Figure 4-10. The input 230V, 3-phase, 42-frame, 4-pole, AC torque motor drives an integral 30:1 gear-head, whose output shaft is coupled to a 15:1 worm reduction. The worm-reduction in turn is connected to the wave generator of a 242:1 harmonic drive, whose output produces reflector motion about the azimuth axis. The worm-gear reduction is required to prevent back driving of the drive train under high external loads. The elevation drive train employs the same motor-gear-head as the azimuth train. The motor-gear-head is attached to a linear screw-jack having a 6:1 worm input and an output reduction through the acme-type screw of 9.52 mm (0.375 in) per revolution. The antiback-drive nature of the screw jack prevents back driving and excessive power consumption under high external loads. The drive unit hardware is shown in Figure 4-11.

Figure 4-9. Reflective Surface Supporting Structure

Figure 4-10. Elevation-Azimuth Drive Unit

Figure 4-11. Drive Unit Hardware

The azimuth and elevation drive trains for the inverting heliostat SRE are schematically identical. The input 230V, 3-phase, 42-frame, 4-pole, AC torque motor drives an input 45:1 reducer, whose output shaft is coupled to a 961:1 output-stage reduction. Both stages are Orbidrive transmissions. Figure 4-12 schematically shows the functional parts of a typical Orbidrive. The reaction and output cams are integrally machined as a common piece which is fitted to the input shaft's eccentric, causing the reaction cam to orbit within the rollers of the reaction disk when the input shaft is rotated. Since there is one less reaction cam lobe than the number of reaction disk rollers, the cam indexes one roller for each cam orbit. In a similar fashion, the number of rollers on the output disk is one less than the number of output cam lobes, so that the output disk indexes one roller in the opposite direction relative to the output cam for each orbit. The differential motion of the output disk relative to the reaction disk provides the high-speed reduction desired.

The Orbidrive elevation drive train is shown in Figure 4-13. The azimuth drive employs identical motor, input reducer, output reducer cam, and reaction disk, and even the same Timken support bearings.

The selected configuration is rather more like that of the octagonal heliostat than that of the SRE inverting heliostat. As shown in Figure 4-14, dual linear actuators are employed. The housing of the lower actuator is attached to the azimuth drive housing. The jack screw is attached to a drag link. This lower actuator is used for stowage when the reflective surface is to be stowed face down. The upper actuator connects between the drag link and the reflector support structure and is used for tracking. This actuator is also extended for face-down stowage.

The azimuth drive is similar to that of the octagonal heliostat. The motor gear-head reduction ratio is reduced to 15:1, with other elements about the same.

The harmonic drive for the azimuth drive train was chosen on the basis of excellent stiffness, wear, and fatigue life (note that Orbidrive is also excellent for stiffness and fatigue, but marginal for wear) and minimum cost and

Figure 4-12. Orbidrive Operating Principle

Figure 4-13. Orbidrive Elevation Drive Train

Figure 4-14. Pilot Plant Heliostat Design

schedule risk. The double-jack elevation drive was chosen as a lower cost alternative to Orbidrive and harmonic drive and also promises excellent performance and minimum risk.

4.3.3.6 Pedestal/Foundation

The pedestal/foundation for the octagonal heliostat is illustrated in Figure 4-15. The pedestal is composed of a steel cone with attach angles at the top and bottom with eight bolt holes in each for attachment to the drive unit and the foundation. The foundation is circular and buried to a depth of 23 cm (9 in) below grade with the base on the slab 40.6 cm (16 in) below grade. The base of the foundation is a flat slab 17.8 cm (7 in) thick and 2.13 m (84 in) in diameter, reinforced with 9.5-mm (3/8-in) diameter reinforcing rods. After the cone is attached to the foundation, the interior of the cone is filled with soil or sand to provide mass to resist overturning moments.

Figure 4-15. Pedestal Foundation Assembly

The inverting heliostat pedestal (Figure 4-16) is a thin-walled steel column, 0.503 m (20 in) in diameter and 2.74 m (108 in) long, welded at the lower end to a standard-weld neck-pipe flange. The upper end has a 19-mm (0.75-in) thick steel plate for attachment of the drive unit. Attachment to the concrete foundation is provided by eight 31.8 mm (1.25-in) diameter anchor bolts equally spaced on a circle of 0.686-m (27-in) diameter. The foundation is a precast reinforced concrete, cone-type spread footing, with the base 1.14 m (45-in) below ground. A central concrete column, 0.813 m (32 in) in diameter and 0.61 m (24 in) high extends from the cone top to 76.2 mm (3 in) above grade.

It was necessary to choose the 0.508 m (20-in) tube to minimize the slot dimensions in the reflector and still provide clearance for inverting.

4.3.3.7 Sensor Pole/Foundation

For closed-loop tracking, a sensor pole (Figure 4-17) is used to support the tracking sensor at a location approximately on a line between the center of the

Figure 4-16. Pedestal/Foundation

Figure 4-17. Sensor Pole

heliostat and the receiver. A thin-walled tube 0.127-m (5 in) in diameter with a 3.05-mm (0.120 in) wall, located 4.83 m (190 in) from the heliostat centerline, is used for the vertical column. A tubular steel boom 76.2 mm (3 in) in diameter with 3.05-mm (0.120-in) wall with a 45-degree bend at the lower end is attached to the vertical column with a standard swage reducing pipe nipple. The nipple is welded to the boom and clamped to the vertical pole with four set screws to maintain a rigid position. The tracking sensor is installed on the upper end of the boom by means of a sliding mount fitting for adjustment of position along the boom. For lateral adjustment, the boom swings around the pole and is locked in position by four set screws. The booms are of five different lengths to accommodate the total field. The sensor pole foundation is 1.22 m (48 in) in diameter and 0.508 m (20 in) deep.

4.3.3.7 Control/Electronics

The preliminary design control/electronics closed-loop control system consists of AC motors, encoders, optical sensors, and triac switches; heliostat, field and master controllers; and power and data distribution systems. A block diagram for the pilot plant field is shown on Figure 4-18. Power distribution is accomplished by primary feeders at 2.4 kV and secondary feeders at 240 V. The secondary power buses go to each heliostat. The data distribution network encompasses the central computer, field controller, and the heliostat controller. The heliostat controller receives inputs from the field controller, motor encoders, and control sensors.

The field controller/heliostat controller is illustrated in Figure 4-19. Control sensors are required in the system to enable the control electronics to determine the mirror attitude and to correct that attitude when required. There are two basic sets of sensors used: the tracking sensor and the position encoders. The tracking sensor is a unit associated with each individual heliostat for the purpose of fine-tracking control of the reflected beam direction toward the receiver. It is mounted on a pole between the heliostat and the receiver so that it receives the reflected beam from some part of the center mirror. The axis of the sensor is oriented to intersect the aim point on the receiver. If the reflected beam is parallel to the sensor axis a null signal is generated from the sensor. If the reflected beam is not parallel to,

Figure 4-18. Branch-Collector Field Network

Figure 4-19. Power and Data Distribution - Collector Field Network

but within ± 5 degrees of the sensor axis, an output signal is produced proportional to the angular error. A constant error signal is produced for beam alignment errors from ± 5 to ± 15 degrees to aid sun acquisition. The sensor has two orthogonal error directions lying in the plane normal to the sensor axis. The motor shaft counter and gimbal axis encoders provide data for accurate tracking when not on target.

4.4 COLLECTOR SUBSYSTEM RESEARCH EXPERIMENTS

4.4.1 Helio- stat Configurations

MDAC has fabricated and tested two heliostat configurations under this contract. The heliostats are described together with the heliostat array tests and the laboratory tests in the collector SRE in this section.

In June 1975, DoE initiated the Phase 1 Solar Thermal Power System contract with MDAC. Under this contract, MDAC has designed, fabricated, and tested six noninverting octagonal heliostats as shown in Figure 4-2. The octagonal heliostat assembly consists of eight trapezoidal reflectors supported by radial beams which tie to a hub. The reflectors include first-surface mirrors with acrylic coating on 1/4 in thick float glass and second-surface laminated mirrors. The reflectors and substructure are rotated in azimuth by a harmonic rotary drive and in elevation by a jack. The assembly is supported by a conical pedestal and a reinforced slab foundation. The sensor is supported by a sensor post and is connected to the field controller. The field controller is commanded by the master control.

MDAC has designed, fabricated, and tested two inverting orthogonal heliostats as shown on Figure 4-3. The orthogonal heliostat assembly consists of six rectangular reflectors supported by a central torque tube and four cross beams. The reflectors include second-surface laminated mirrors and second-surface foam-sandwich mirrors. The reflectors and substructure are rotated in azimuth and elevation by "Orbidrive" rotary drives. The assembly is supported by a cylindrical pedestal and a concrete foundation. The sensor and the electronics are identical to the octagonal heliostats.

The noninverting octagonal and inverting orthogonal heliostats underwent extensive assembly-level and subsystem-level tests.

The configurations of the complete heliostats fabricated for test are listed in Table 4-3. The tests done at the Naval Weapons Center (NWC), China Lake, California were:

- Array

The tests done at MDAC-Huntington Beach, CA, were:

- Structural
- Controls development
- Harmonic drive/jack life

Tests done at Compudrive, Boston, Massachusetts, were:

- Orbidrive life

Table 4-3

HELIOSTAT HARDWARE FABRICATED

Heliostat	Reflector	Test
Noninverting Octagonal		
H1	First-surface	Structural Array
H2	First-surface	Controls Development Array
H3	First-surface	Array
H4	Laminated	Array
H5	Laminated	Mobile Array
H6	Laminated	Drive unit life
Inverting Octagonal		
IH 1 - var 1	Laminated	Controls development Array
- var 2	1/2 laminated 1/2 foam-core	Array
IH 2	1/2 laminated 1/2 foam-core	Structural

4.4.2 Subsystem Level Tests at NWC

A subsystem-level test program was conducted at NWC involving:

- Engineering performance
- Installation, operation, and maintenance

The heliostat array test location is shown on Figure 4-20. Five heliostats were installed on foundations 640 ft north of the two towers supporting the target. A photograph of the five heliostats is shown in Figure 4-2. The sixth heliostat was mounted on a trailer to permit operation from different locations in the field.

Engineering performance tests provided the following information:

- Data on the operating modes
- Mirror reflectivity
- Heliostat beam irradiance distribution at the target
- Heliostat power consumption.

Figure 4-20. Heliostat Array Test Locations

Operating data showed that:

- The heliostat and field controller were stable and that the units operated satisfactorily in the following modes:
 - Sun acquisition
 - Synthetic tracking
 - Command position
 - Off-nominal performance (to 1/16 normal sun intensity)
 - Stowage (as applicable)
 - Multiple aim

The algorithm for slewing through a tracking "pole" was not verified due to time constraints.

- Array tracking performance

Typical Time Averaged Error Band (mr)		
Wind Velocity	Noninverting Octagonal with Harmonic/Jack	Inverting Orthogonal with Two Orbidrives
Calm	1/2	3/4
10 m/s	3/4	2

The mirror reflectivity was measured at regular intervals by Epply pyrheliometers. The average reduction in reflectivity due to dust accumulation between washings was:

- Noninverting heliostat
 - Acrylic mirrors ~ 7 to 8%
 - Laminated mirrors 6-1/2%
- Inverting heliostat ~ 3%.

Tests performed on out-of-flatness of the laminated mirrors using two beam sensors showed as much as 6-mr beam error.

The heliostat beam irradiance distribution at the target was given by the digital image radiometer (DIR). The DIR block diagram is shown on Figure 4-21. The radiometer consists of a video camera and scanning electronics, A/D converter, NOVA 1200 computer, cassette tape storage, CRT

Figure 4-21. Digital Image Radiometer Block Diagram

display, video monitor, and keyboard. The radiometer and target were calibrated against thermopile pyrhemometers so that the irradiance (W/m^2) as measured by the DIR is within an accuracy of 2% for any given data point. The 3600 data points from the DIR were recorded on tape and processed to plot the beam intensity distribution, weighted centroid, total power, and total energy. The beam quality of the heliostats at NWC was determined and correlated with the insolation level and reflector characteristics. Some of the significant results are as follows:

- The DIR determines the power within 2% for full-scale heliostats.
- The optical performance of the laminated reflectors was about 5 to 7% better than the sandwich-core reflectors. Subsequent reflector tests at Huntington Beach showed that this was probably due to a combination of initial waviness of the mirror and the reflector assembly process. Photographs showed marked light and dark striations of the reflector image on a nearby target. Subsequent reflector development work has resulted in flat panels that do not show shadowing of the images.

The heliostat power consumption was remarkably low. The normal tracking power was approximately 12 watts/heliostat. The power during sun acquisition was 16 to 18 watts. The energy to stow the inverted heliostat (180 degree motion) was 65 watt hours. .

The installation, operation, and maintenance of the heliostats provided the following significant data:

- Installation times for future estimates.
- Simple, rapid, closed-loop sensor alignment procedures were developed.
- Meteorological data at NWC pertinent to pilot plant operation. (NWC is 45 miles away from Barstow, and has similar weather conditions.)
- The acrylic first-surface reflectors showed unacceptable degradation due to pinholes, staining, and acrylic delamination.
- The heliostat and sensor pole foundations were quite stable.
- Two cracks were observed on the sandwich foam-core reflectors of the inverted heliostat after the plastic covers were removed. The sandwich reflectors (only) were covered with plastic to permit comparative measurements of beam intensity with the laminated reflectors. The installation of the covers caused the glass temperature underneath to rise to approximately 130°F. When the covers were removed, the reflectors cooled rapidly to ambient temperature (75°F). The resulting thermal stress is believed to have caused cracking of the glass.
- Validation of reflector washing requirements and timelines. Surprisingly, frost and snow turned out to be an excellent cleaner.

4.4.3 Collector Component Development

The collector components included reflectors, structural support, drive units, pedestal, foundation, sensor pole/foundations, field controller, and master control. The components were developed by trade studies, design, fabrication, and test.

4.4.3.1 Reflector

The reflector development included:

- The basic characteristics of the silver deposition, and the effect of glass chemistry and thickness.
- Acrylic coated first-surface mirrors.
- Laminated and foam-sandwich core second-surface mirrors.

The solar reflectance efficiency of vapor and chemical deposition silver was measured at Air Mass Two. The chemically deposited silver was selected, and little difference was found between various commercial processes. Extensive testing was conducted at various acceptance angles (4 to 125 mr) with the Beckman and specular spectrophotometer over the solar spectrum and the specular photometer at 550 nm. In conjunction with other sources, data were obtained on solar transmittance and reflectivity as a function of glass thickness, chemistry (iron content), and aperture angle. The results showed that an average second-surface reflectivity of 0.88 was reasonable for vertically stowed mirrors for pilot plant. An average reflectivity of 0.91 for second-surface mirrors for commercial plant can be achieved with currently available processes.

Extensive development was conducted on first-surface mirror coatings. These included various formulations of Sheldahl type 822-12A acrylic coatings and other coatings from Texton Plastics, Dupont, Sierracin, Owens Illinois, De Soto, and others. The Sheldahl coating was selected, and subjected to salt spray, reflectance, abrasion, and exposure tests. A number of full-scale trapezoidal panels were made and installed on noninverting octagonal heliostats. Subsequent tests showed that the first-surface mirrors are subject to pinholes, staining, and delamination of the acrylic. The conclusions are that:

- For pilot plant, the technology is not ready for use of first-surface mirrors using polymerized coatings.
- Cost and performance advantages indicate a desirable development area for commercial system cost reduction.

Laminated second-surface mirrors with white mirror-backing paint were developed and installed on both types of heliostats. The laminates were

bonded to main beams for the octagonal configuration, and to three stiffeners attached to the cross beams for the orthogonal configuration. Problems of reliability were encountered with the grey mirror-backing paint. The grey backing paint was replaced by the white backing paint, primarily because of backlighting problems. Even with white backing paint, continuous exposure to backlighting conditions over a year's time period caused cracking of the laminate. This is considered to be too severe a test, and results are discounted. The laminates were not as flat as desired (0.060 to 0.070 in. out of flatness). No cracking of MDAC laminate glass due to the environment was observed. The cost of production of laminate reflectors is about 20% greater than the foam-core sandwich (which will be discussed). All in all, the MDAC laminated mirror performance over the past 8 months has been satisfactory.

Cost/performance trade studies of a number of different types of reflectors were made. These included sandwich, corrugations, and other types of stiffened configurations. The 1/8 in second-surface mirror, 2-in thick styrofoam core and 0.022 in galvanized steel backing sheet was selected. Component tests were made to establish material, structure and core creep properties. Creep of the 2 lb/ft² core was not a problem. Full scale panels (42.5 x 114 in) were fabricated and tested at NWC.

For both the laminated and the foam-core sandwich, the following tests were conducted:

● Flatness	● Salt spray
● Thermal/backlighting	● Accelerated UV
● Thermal cycle	● Adhesive compatibility
● Thermal distortion	● Beam creep
● Washing	● Beam bending
● Reflectivity	● Hail impact
● Exposure	

For both configurations, the component tests indicated satisfactory performance. However, the results at the NWC showed that the full-scale sandwich reflectors had lower optical performance than the laminated reflectors, due to surface irregularities. Subsequent modifications to the manufacturing

process have produced a number of small (30 x 42.5 in) panels judged to be flat as shown by the lack of grey areas in the reflected image and comparison with acceptable laminated panels.

As far as cracking is concerned, the glass edges cannot have large/sharp nicks and the transient, differential in-plane temperatures must be kept below 20°F. This can be accomplished with careful manufacturing process, edge protection, and avoidance of in-plane temperature differential.

As a result, MDAC recommends the foam-sandwich reflector and believes that current development will result in efficient, low-cost reflectors. The laminated reflectors have been qualified and are a satisfactory back-up.

4.4.3.2 Structural Support

Each heliostat incorporates a structural support for the reflectors and is attached to a drive unit located on top of a pedestal and foundation.

For the octagonal heliostat, the reflectors were supported by 16 radial beams that tied to the hub. In the orthogonal heliostat, the reflectors were supported by a central torque tube and four cross beams.

Heliostat structural tests were conducted to verify the heliostat strength, deflection, and fundamental vibrational frequency. Complete heliostats of both the octagonal and inverted type were used for the structural tests. One segment or panel was given a finely distributed load. The remaining segments or panels were given concentrated loads simulating their contribution to total heliostat loading. The strain and deflection measurements were compared with predictions from analysis using a finite-element computer program, NASTRAN.

Deflection levels under the simulated operational loading of 11.6 m/s (26 mph) were within 10% of the predictions from the NASTRAN model. Measured strains under the survival wind loads of 46.5 m/s (104 mph) for the octagonal heliostat and 44.7 m/s (100 mph) for the inverted heliostat were equal to or below NASTRAN predictions. Vibrational frequencies for pitch and yaw

modes were within 5 to 10% of predictions based on the NASTRAN model. The closeness between the measurements and predictions verifies the heliostat design and the validity of the model.

4.4.3.3 Drive Units

Each heliostat incorporates a drive unit which produces appropriate reflector motions for solar tracking, emergency slewing, routine reflector positioning, and stowage. The drive unit consists of an azimuth and elevation drive train, each comprising a motor with several stages of reduction, capable of producing the needed high-torque, low-speed output.

The requirements for the drive unit included: travel, operating loads, stowage loads, backlash, minimum resonant frequency, no back-drive and 30-year life.

For the octagonal heliostat the drive unit consisted of a harmonic drive in azimuth and a linear actuator in elevation. For the inverting heliostat, further trade studies were conducted in order to attempt to reduce the cost of the drive unit. The studies considered Orbidrive, harmonic drive, Helicon gears, linear actuators, spur gears, and chains. As a result of the trade studies, Orbidrive was selected for both axes because of the potentially lower cost and acceptable (1/2 mr) backlash. The Orbidrive was designed, fabricated, and tested at Compudrive Corp., and was subjected to additional testing by MDAC.

Thirty-year life-cycle qualification tests were successfully completed on the harmonic drive, Orbidrive, and linear actuators. The data included information on integrity, efficiency, backlash, slew rates, wear, and back-drive characteristics. It is significant that the test results of the backlash for the harmonic drive and linear actuators (0 to 1/2 mr) was less than for the Orbidrive (2 mr).

The drive units and control sensors were subjected to environmental tests to verify their performance and survival during temperatures from -30°C (-22°F) to +60°C (140°F), rain to 5 cm (2 in) per hour, icing from freezing rain, blowing dust, and hail impact up to 25 mm (1 in).

4.4.3.4 Controls/Electronics

The controls and electronics system for the SRE consist of:

- AC drive motors controlled by triac switches actuated by a field controller.
- Heliostat control data provided by optical sensors in the sun tracking mode and potentiometers/limit switches in the command mode.
- A field and master controller connected by a data distribution system.
- Power at 240 Volts provided by utility company transformers.

This control system was qualified in the SRE program and met all the system requirements.

4.4.4 Collector SRE Conclusions

The results of the SRE testing on the pilot plant collector test articles lead to the following conclusions:

- A. A controls system has been defined using a closed-loop beam sensor that is capable of orienting the reflector surface to within 1 mr of the desired direction throughout the range of operating wind speeds.
- B. Open-loop operation using gimbal-axis potentiometers was demonstrated. While this tracking was not designed to be sufficiently accurate for normal heliostat tracking, incorporating accurate position transducers to track open loop does not appear to be a problem.
- C. Reflectors incorporating sufficient surface contour accuracy for both pilot plant and commercial plant operation can be produced. The production processes identified acceptable costs.
- D. The production processes for the foam-core reflector can be readily adapted to produce curved reflector panels for additional focusing, if desired.
- E. The proposed configurations of the reflector, drive unit, and support structure are able to withstand the anticipated environment, including winds up to 100 MPH, blowing dust, rain, hail, icing, and high and low temperatures.
- F. Multiple heliostat control by a field controller presents no significant problems.

- G. Heliostat power consumption is very low (12 watts average). The parasitic load should be only about 0.25% of the net power generation.
- H. Reflector cleaning techniques have been identified which require minimal time and materials and provide excellent cleaning performance.
- I. Installation, operation, and maintenance data for the collector were obtained and documented for a site which is close to the pilot plant site and has similar weather and soils.
- J. Pedestals and foundations were quite stable.
- K. Modeling of the heliostat on NASTRAN accurately represents the actual measurements and provides confidence that the effects of minor changes to the structure can be accurately forecast.
- L. The digital image radiometer has been shown to be a very accurate and valuable tool for measuring and recording reflected beam accuracy and power distribution.

4.5 PILOT PLANT MANUFACTURING PLANS

Production of the collector subsystem and ancillary hardware will be accomplished at the MDAC Huntington Beach facility, the MDAC site facility at the Southern California Edison Coolwater Plant, and with suppliers generally in the Los Angeles and Orange County industrial area. Wherever practical, the manufacturing processes and methods employed for the production of pilot plant hardware will be similar to that planned for the commercial plant. A brief review of the pilot plant manufacturing flow is depicted in Figure 4-22.

A series of operations is being planned to systematically develop and validate the production design, manufacturing, tools, processes, and procedures. The initial validation will occur through the manufacture of a preproduction heliostat using the final design drawings but without production tooling. This initial unit will form the basis to verify that the detail design drawings result in a finished product acceptable to MDAC. This will be followed by a second heliostat manufactured from the selected production tooling and practices. This unit may be considered as the final checkpoint before full-scale production commences.

Figure 4-22. Manufacturing Summary Flow

To ensure an efficient and orderly integration of raw materials and components, buildup to the required production rate of 10 heliostats per day will occur over a period of approximately 2 months. This approach will provide a reasonable time period during which the manufacturing and assembly personnel will be brought to the level of efficiency necessary to satisfy the production rate demands imposed by the program schedule.

MDAC is currently planning for the manufacture and assembly critical hardware items such as the drive unit and reflective surface at the Huntington Beach facility. Subassembly of the structural elements as well as the final heliostat assembly will be performed on the SCE Coolwater Plant site in the MDAC assembly building. Upon completion of the heliostat assembly it is transported by forklift to the field for tiedown to the foundation. Transportation plans have been completed which outline the logistics from the acquisition of raw materials to the delivery of the completed heliostat at the foundations.

Section 5

RECEIVER SUBSYSTEM

5.1 INTRODUCTION

The function of the receiver subsystem in a central receiver solar thermal power plant is to absorb the solar radiation reflected from the collector subsystem and convert that energy into superheated steam for delivery to either the turbine of the electrical power generation subsystem or to the thermal storage subsystem. The receiver subsystem consists of the receiver unit, the tower on which the receiver unit is mounted above the collector field, and the supporting control and instrumentation equipment.

5.2 HISTORICAL SUMMARY

The dominant receiver subsystem activity during the first quarter following the ATP of 1 July 1975 was the definition of the preliminary design baseline for the 10-MWe solar pilot plant. The baseline design as shown in Figure 5-1, was completed on schedule and accepted by DoE/Sandia in the fall of 1975.

The conceptual design of the receiver subsystem research experiments and hardware necessary to verify the pilot plant design was also initiated during the first contract quarter and accepted by DoE/Sandia in October 1975. In order to avoid the risks inherent in having to scale up SRE test results and to obtain fabrication, cost, and performance experience directly applicable to the pilot plant, full-scale pilot plant hardware assemblies were designed for SRE testing. Detail design of the SRE tests and hardware and initial procurement activities were begun in the second contract quarter.

Following DoE/Sandia approval of the SRE detailed designs in March 1976, the fabrication, installation, and checkout of the test hardware and test facilities began.

Figure 5-1. Baseline Receiver Unit Design

Fabrication of the full-length, single-tube test article and its installation into the radiant-heat test facility were completed in June 1976. A series of tests conducted during July culminated in the stable generation of superheated steam at 1,500 psig without the need for stability orificing of any kind.

During July 1976, the full-length, five-tube test receiver for multitube performance investigations was completed, inspected to ASME code requirements, and passed its ASME certification as a commercial boiler. Test facility modifications were completed in late August, permitting the five-tube test program to begin on 3 September. The low-flux, low-flow stability investigation phase of the testing was successfully completed by 17 September and confirmed the multitube, unorificed flow stability of the receiver design under the most severe pilot plant flow conditions. Following radiant heater modifications, successful testing of the five-tube panel under maximum pilot plant flux and flow conditions was completed early in October. All tests again indicated stable and uniform flows and temperatures in all tubes.

Maximum tube wall temperatures were lower than expected over the entire range of heat fluxes indicating thermodynamic performance better than predicted. Emergency simulation tests demonstrated that the design could survive flow cessations for significant periods without damage. In general the five-tube test results established high confidence in the receiver design.

Fabrication of the full-scale pilot plant receiver panel for the final SRE tests was completed in September 1976. Minor water seepage from a tubing flaw was detected during hydrostatic testing and corrected in accordance with ASME code requirements. The subsequent ASME certification of the pilot plant as a commercial boiler cleared the way for full-scale verification testing of the pilot plant receiver design.

Beginning with low-power, low-flow stability investigations early in January 1977, the tests proceeded smoothly through a series of closed-loop control runs, and were completed with high-flux tests which exceeded expected pilot plant flux/flow conditions. Flow stability was excellent in all tests without the use of flow-stabilizing orifices. Thermodynamic performance exceeded predictions with smooth temperature transients and lower than expected tube-wall temperatures under all test conditions. The design adequacy of the full-scale panel thermal expansion provisions was also successfully demonstrated.

In summary, the receiver subsystem research experiments demonstrated with full-scale pilot plant hardware that the receiver design met or exceeded pilot plant requirements and expectations and thus established high confidence in the commercial plant receiver concept.

Concurrently with the analysis of the data from the SRE test program, design analyses were reinitiated to complete the commercial plant receiver design in sufficient detail to finalize the design requirements for the pilot plant receiver and to provide sufficient depth to estimate commercial receiver costs. Based on these studies and the SRE test data, the pilot plant receiver preliminary design task was also reinitiated to prepare for the Preliminary Design Report submittal.

The Preliminary Design Report was completed and submitted to DoE on schedule in May 1977, in conjunction with oral presentations of the final commercial and pilot plant receiver subsystem preliminary designs.

The schedule summarizing the total receiver program is shown in Figure 5-2.

5.3 DESIGN SUMMARY

5.3.1 Receiver Concept Rationale

Clearly, the primary criterion by which the solar power plant designs must be judged is cost of electricity. The receiver concept selected must be consistent with the DoE's central receiver solar power program objective to develop the solar commercial power plant with the lowest investment and operational costs. The use of cost as a primary criterion is dictated not only for solar to be competitive with alternate energy sources, but also to make maximum use of available research and development funds.

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Figure 5-2. Receiver Schedule

A cost-effective receiver design must necessarily be as simple as possible to manufacture, assemble, operate and maintain. The McDonnell Douglas/Rocketdyne receiver unit is composed of a number of structurally identical panels constructed of multiple tubes of a high-strength, high-ductility, corrosion-resistant steel (Incoloy 800). The tubes are laid side by side and joined together thermally and structurally and made light-tight by full-length longitudinal seam welds. The simple, welded, straight-tube panel design concept permits the use of easier and more economical assembly-line techniques to fabricate the identical panel modules. The lightweight modular panels may also be easily transported and easily handled during the hoisting and installation of the receiver onto the tower. The modular panel concept also provides a receiver with a unique maintenance capability in that a panel could be removed from the tower and replaced within 12 hours, if necessary, to minimize system down time.

Conceiving the solar receiver as a combination of modular panel assemblies provides the freedom to size the panels and arrange them into whatever geometric shape is determined by the overall system analyses to be optimum for a central receiver solar power plant. The receiver geometry does not dictate the collector field shape, heliostat shape, or pointing accuracy requirements, but is instead determined by whatever combination of these subsystem elements produces the lowest overall system cost. The simplest and most direct approach to the commercial power plant is to design a single receiver that reduces the distance to the outermost heliostats, thus reducing atmospheric attenuation and tower height by distributing the heliostats in an optimum pattern 360 degrees around the receiver. Energy collection and transmission to the power station is thus accomplished predominantly optically, and the cost and complexity of an extensive steam and water piping network is avoided.

Following the dictates of the foregoing analysis, the selected receiver design concept for both the pilot and commercial solar plants mounts the individual tubular panel assemblies of the receiver unit on a central core support structure to form a single, large and essentially circular (multisided), cylinder atop the tower. The heliostats surrounding the receiver direct their

collected insolation to the external surface of the cylindrical receiver. By designing the receiver to accept insolation over the total external surface of the cylindrical receiver, the aiming accuracy and focusing requirements imposed on the heliostat and its drive mechanisms are minimized and heliostat costs are reduced. Moreover, the external receiver is insensitive to damage caused by operational aiming inaccuracies, since the uncooled structure adjacent to the insolation target area is minimized and none of it need be crossed by the heliostat beams when the heliostat is slewed toward or away from the receiver.

The selected solar receiver design should provide a high probability of technical success, while minimizing development time and therefore costs. In a new technology development program such as solar energy, the probability of success is directly related to the flexibility of the selected design to adapt to the unexpected. Not only should the design be easily modifiable to be compatible with the optimum system which will be defined during the development program, but it should also be reasonably insensitive to off-nominal conditions that may occur during the process.

In keeping with the low-technical-risk concept selection criterion previously specified, the welded, multitube boiler panel concept had been tested during a DoE R&D program accomplished prior to the present pilot plant preliminary design effort and proved to be a viable design at a flux level approximately three times greater than the maximum anticipated for the pilot plant receiver. Moreover, although testing to date has indicated that tubing material less expensive than Incoloy-800 could be used to fabricate the receiver, the added safety margins, and thus the development flexibility provided by its higher material characteristics and the extensive and successful history of its use in conventional boilers and super heaters, more than justifies its use for the first solar power plant. Calculations show that even if water flow to the receiver unit were to be totally lost during maximum power operation, the receiver could tolerate maximum solar insolation without damage for a period many times that required to slew the heliostats away from the receiver.

Technical risk was further reduced with the modular panel concept by permitting the receiver subsystem research experiments of the present contract to be carried out with pilot-plant-scale test articles, thus permitting early validation of the design, construction, and testing of actual pilot plant hardware and eliminating the dangers inherent in scaling up from SRE test hardware and results. Although scaling will eventually be required to move from pilot plant to commercial plant power levels, the scaling risk will be minimal since the maximum commercial flux level is less than that already demonstrated by test to be compatible with the multitube boiler panel concept.

Finally, the selected receiver design must be compatible with the unique central receiver solar environment. The receiver must have high thermal response in keeping with the transient nature of insolation due to the diurnal cycle and clouds. It must be capable of extracting in useful form the maximum amount of annual thermal energy under these transient conditions without damage to itself or the other subsystems. Because of the necessity in the central receiver concept to locate the receiver on a tower several hundred feet above the collector field, the receiver must be as light as possible to minimize the tower and receiver unit structural weights and thereby the construction costs required to resist expected seismic loads amplified by the elevated receiver position.

The single-pass to superheat tubular panel concept provides a receiver with the lightest possible thermal mass and thus the highest thermal response of any available boiler type and is, therefore, consistent both with the highly transient nature of insolation and the seismic sensitivity of a tower-mounted central receiver.

The MDAC/Rocketdyne design team has conceived, designed, fabricated, and proven by test the receiver concept that will satisfy the DOE central receiver solar program objectives of achieving the lowest commercial plant initial and operational costs. In summary, the proposed external multipanel concept provides:

- Lowest commercial plant cost
- Identical modular panel assemblies to permit
 - Low-cost assembly line fabrication
 - Ease of transport, handling, and installation

Developmental flexibility and adaptability to optimum system requirements

Rapid maintenance

- Rapid thermal response
- Minimal seismic sensitivity
 - Low weight
 - Shorter tower
- Low technical risk
 - Pilot-plant-scale hardware proven in SRE
 - High-capability materials
- High tolerance to emergency conditions
- A single receiver and tower with growth capability beyond 100 MWe
- Minimum impact on other subsystems
 - Reduced heliostat aiming accuracy
 - Optimum collector field shape

5.3.2 Commercial Receiver Design Definition

The mission of the commercial receiver subsystem is to efficiently transfer the energy from the concentrated solar radiation reflected from the mirrors of the collector subsystem into the water supplied by the electrical power generation subsystem to produce steam for delivery at precisely controlled superheat conditions to the plant turbine generator and/or the thermal storage subsystem.

Nominally, the commercial receiver will be required to accept water from the flow distribution system at 15.5 MPa (2250 psia) and 231°C (448°F), and deliver superheated steam at rated conditions of 11.1 MPa (1,615 psia) and 516°C (960°F) to the electrical generation subsystem. Any rated steam generated in excess of turbine power requirements will be diverted to the thermal storage subsystem.

The receiver subsystem must also be capable of accepting feedwater at 15.5 MPa (2250 psia) and 249°C (480°F), and delivering steam at 11.1 MPa (1,615 psia) and 368°C (694°F) when it is required to charge the total receiver energy output into thermal storage.

The receiver must safely and efficiently absorb incident solar radiation at a maximum flux of 0.85 MW/m^2 . In addition, the receiver must be able to accept, without damage, thermal gradients imposed by radiation transients from essentially zero to maximum flux in as little as 10 seconds as a result of precipitation or the intermittent passage of clouds over the collector field.

As a forced-flow steam generator, the receiver was designed and will be certified to the requirements of Section I of the ASME Boiler and Pressure Vessel Code. Fatigue life will be verified using the more sophisticated analysis techniques of Section VIII, Division 2.

The major hardware assemblies comprising the commercial receiver subsystem are the receiver unit, the tower on which the receiver unit is mounted above the collector field, and the supporting control and instrumentation equipment. The receiver unit is made up of 4 modular preheater panels and 20 boiler panels, flow control and instrumentation equipment and the supporting structure.

All boiler panel assemblies are constructed of 170 tubes of high-strength, corrosion-resistant Incoloy 800 laid side by side, joined together thermally and structurally, and made opaque to incident light by full-length longitudinal welds. All boiler tubes are 12.7 mm (0.5 in) OD x 6.9 mm (0.27 in) ID.

Preheater panels are of identical construction with the exception that they are made up of only 113 tubes which are 19.0 mm (0.75 in) OD x 13.2 mm (0.52 in) ID. As will be discussed later, the solar flux environment is much less severe on the preheater panels and a larger tube size may be permitted to reduce the pressure drop through the preheater circuit.

An Incoloy 800 water manifold is located at the lower end of each panel assembly and functions to distribute water equally to all panel tubes. An Incoloy 800 manifold is also located at the upper end of each panel assembly to act as a collector manifold for all tubes. To ensure leak integrity, all panel tubes are welded to both manifolds.

All tubes are 27m (89 ft) long. The exposed length is 25.5 m (83.6 ft). Additional length provides for folding over at the top and bottom of the panels to protect the inlet and outlet manifolds and support structure from radiation.

The surfaces of the tubes exposed to solar radiation are coated with Pyromark paint which has demonstrated an absorptivity of 95% over a wide range of wavelengths. The coating is resistant to weathering and was tested for long-term compatibility with high-intensity solar radiation during the subsystem research experiments.

Each panel tube bundle is mounted to a panel back-up structure to maintain the panel shape and hold it to the receiver tower structure in proper location, while allowing for thermal growth and providing support for wind and seismic loads. A series of sliding clips and channels permit unrestrained lateral and vertical thermal expansion of the panels to prevent the buildup of thermal stresses. Each panel is insulated on the back side to reduce thermal loss and protect the support structure and control components.

The individual tubular panel subassemblies of the receiver unit are mounted on a central core steel support structure to form a single large and essentially circular (24-sided) cylinder 17m (55.8 ft) in diameter by 25.5m (83.6 ft) long. The heliostats surrounding the receiver tower direct their collected insolation onto the full 360 degree external surface of the receiver unit (Figure 5-3).

The receiver tower is of jump-formed concrete construction and extends 242m (794 ft) above grade to the interface with the receiver unit steel support structure which continues up through the receiver unit to an elevation of 281m (921 ft) where it is crowned with a 20-ton capacity service crane. The midpoint of the receiver unit is 268m (879 ft).

The top of the concrete tower has an outside diameter at 15.3m (50.25 ft) with a nominal 0.305m (12-in) wall thickness. The base of the tower has an outside diameter of 45.7m (150 ft) with a 0.46m (18-in) wall thickness. The foundation, which is located 4.9m (16 ft) below finished grade, is a 61m (200-ft) outside diameter annular circular mat with a 30.5m (100-ft) inside diameter. The foundation thickness is 3.8m (12.5 ft).

Figure 5-3. Commercial Plant Receiver and Tower

Control of the receiver can best be understood by following a typical day's operation. To initiate start-up, water from the water treatment equipment of the electrical power generation subsystem is forced up the receiver tower riser by the receiver feed pumps and into the receiver inlet filter assembly. Leaving the filter assembly, the water enters a manifold which distributes the flow into the inlets of the two parallel sets of two panels in series located on the south side of the receiver and designed to function as preheaters (Figure 5-4). The water absorbs heat as it flows up through the first panel of each of the sets and then down the preheater through the second panel, where it joins the flow from the other preheater in a ring manifold supplying the remaining 20 panel assemblies designated as boiler panels. Passing through a modulating flow control valve located at the inlet to each boiler panel, the water again flows vertically upward absorbing heat from the incident solar radiation and leaving the upper end of the boiler panels as superheated steam. The individual boiler panel inlet valves provide the flow control necessary to maintain constant outlet temperature despite diurnal

Figure 5-4. Receiver Control Schematic

and seasonal variations in heat load at each panel. The control valves also control cloud-induced transients and regulate the start-up and shutdown sequences. With this arrangement, active control of the water flow through the preheater panels is not required.

As the steam exits from each of the panel discharge manifolds, it passes through a pair of cyclone-type water separators as a precautionary measure to ensure absolutely dry steam. The steam enters the steam downcomer collection manifold where it is mixed with the discharge flow from the other boiler panels and is finally carried away by the downcomer to the turbine of the electrical power generation subsystem or the thermal storage subsystem, or both as directed by master control.

During start-up or other conditions, when insolation is too low to produce the proper superheat conditions, a combination of valves is used to divert receiver discharge flow away from the downcomer and into a receiver flash tank assembly until proper superheat conditions are achieved.

Further definition of the commercial receiver subsystem design may be found in the McDonnell Douglas Preliminary Design Report, Volume IV, Document MDC G6776, dated May 1977.

5.3.3 Pilot Plant Receiver Design Definition

The commercial receiver design concept was selected to satisfy the DoE central receiver solar program objective to achieve the lowest commercial plant investment and operational costs. Since the basic mission of the pilot plant is to demonstrate the validity of the commercial plant design concept, the pilot plant receiver design is dictated by a flow-down from the commercial receiver concept. The pilot plant receiver design is therefore necessarily a scaled-down version of the commercial receiver.

Nominally, the pilot plant receiver will be required to accept water from the flow distribution system at 13.8 MPa (2000 psia) and 205°C (401°F), and deliver superheated steam at rated conditions of 10.4 MPa (1515 psia) and 516°C (960°F) to the electrical generation subsystem. Any rated steam generated in excess of turbine power requirements will be diverted to the thermal storage subsystem.

The receiver subsystem must also be capable of delivering steam at 10.4 MPa (1515 psia) and 349°C (660°F) when it is required to charge the total receiver energy output into thermal storage.

The receiver must safely and efficiently absorb incident solar radiation at a maximum flux of 0.30 MW/m². In addition, the receiver must be able to accept, without damage, thermal gradients imposed by radiation transients from essentially zero to maximum flux in as little as 10 sec as a result of precipitation or the intermittent passage of clouds over the collector field.

As a forced-flow steam generator, the receiver will be designed and certified to the requirements of Section I of the ASME Boiler and Pressure Vessel Code. Fatigue life verification will be made using the more sophisticated analysis techniques of Section VIII, Division 2.

The major hardware assemblies comprising the pilot plant receiver subsystem are the receiver unit, the tower on which the receiver unit is mounted above the collector field, and the supporting control and instrumentation equipment. The receiver unit is made up of 6 modular preheater panels and 18 boiler panels, flow control and instrumentation equipment, and the supporting structure.

All panel subassemblies are constructed of 70 tubes of high-strength corrosion-resistant Incoloy 800 laid side by side. They are joined together structurally and thermally, and made opaque to incident light by full-length longitudinal seam welds.

An Incoloy 800 water manifold is located at the lower end of the panel assembly and functions to distribute water equally to all panel tubes. An Incoloy 800 steam manifold is also located at the upper end of the panel subassembly to act as a collector manifold for the effluent steam from all tubes. To ensure leak integrity, all panel tubes are welded to both the water and steam manifolds.

The exposed part of each tube is 12.7 mm (0.50 in) OD x 6.9 mm (0.27 in) ID by 12.5 m (41 ft) long. A section is folded over at the bottom and top to protect the inlet water manifold, lower support structure, and outlet steam manifold from insolation.

The surfaces of the tubes exposed to solar radiation are coated with Pyromark paint, which has demonstrated an absorptivity of 95% over a wide range of wavelengths. The coating is resistant to weathering and was tested for long-term compatibility with high-intensity solar radiation during the subsystem research experiments.

Each panel tube bundle is mounted to a panel backup structure to maintain the panel shape and hold it to the receiver tower structure in the proper location, while allowing for thermal growth and providing support for wind and seismic loads. A series of sliding clips and channels permit unrestrained lateral and vertical thermal expansion of the panels to prevent the buildup of thermal stresses. Each panel is insulated on the back side to reduce thermal loss and protect the support structure and control components.

The individual tubular panel assemblies of the receiver unit are mounted on a central core steel support structure to form a single large and essentially circular (24-sided) cylinder 7m (23 ft) in diameter by 12.5m (41 ft) long. The heliostats surrounding the receiver tower direct their collected insolation to the full 360-degree external surface of the receiver unit (Figure 5-5).

The receiver tower is of steel construction and extends 65m (213 ft) above grade to the interface with the receiver unit steel support structure, which continues up through the receiver unit to an elevation of 86m (283 ft) where it is crowned with a 5-ton capacity service crane. The midpoint of the receiver unit is 80m (262 ft).

The base of the tower is 12.2m (40 ft) square, tapering to 4.6m (12 ft) at the 65m elevation. The square concrete foundation is composed of a 0.61m (2-ft) thick mat which is 15.2m (50 ft) on a side and located 3.96m (13 ft) below finished grade. Concrete wall and pedestals extend 5.48m (18 ft) upwards to meet the steel structure at an elevation of 1.52m (5 ft) above the grade.

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Figure 5-5. Pilot Plant Receiver

Control of the pilot plant receiver is essentially identical to that of the commercial plant, which was previously described in Section 5.3.2.

More detailed design information on the pilot plant receiver can be found in Volume IV of the McDonnell Douglas Preliminary Design Report, Document MDC G6776, dated May 1977.

5.4 RECEIVER SUBSYSTEM RESEARCH EXPERIMENTS

The receiver SRE were designed to eliminate any risks inherent in the performance, control, stability, and mechanical integrity of the pilot plant receiver design. The conceptual and detail designs of the receiver subsystem research experiments and hardware were completed on schedule and accepted by DoE as the basis for the SRE testing which successfully verified the specific capabilities of the pilot plant receiver design.

The receiver SRE included tests of subassemblies as well as of a complete panel of a full-scale pilot plant receiver. The lower level tests included tests of single and multiple tube configurations to provide thermal, hydrodynamic, structural, and life data.

5.4.1 Single-Tube Tests

A single-tube identical to pilot plant tubing in dimensions, material, and configuration was tested in vertical orientation under anticipated pilot plant operating conditions using a radiant heat input.

Objectives of the single-tube tests were (1) facility checkout, (2) preliminary demonstration of thermal performance, (3) preliminary demonstration of safe tube operation under nominal conditions, and (4) demonstration of flow stability in a single tube.

The tests were conducted at the Rockwell B-1 facility in El Segundo. The tube was installed vertically in a reflectorized enclosure and heated by 13 graphite heaters oriented along the axis of the tube. Strain gage transducers were used to measure pressure and thermocouples were used to measure both surface and fluid temperatures. Data were recorded on strip charts and magnetic tape.

Superheated steam was successfully produced without the need of stabilizing orifices in the tube to stabilize the water flow. The successful demonstration of unorificed flow stability and better than expected thermal performance of the single-tube test article cleared the way for the initiation of five-tube full-length receiver panel testing. This testing was begun in early September 1976.

5.4.2 Narrow-Panel Tests

The objectives of the narrow panel tests were (1) to demonstrate thermal performance of the absorber, (2) to verify safe operation and wall temperatures consistent with 30-year life under pilot plant operating conditions, (3) to demonstrate flow stability in a multiple-tube panel, (4) to provide a preliminary demonstration of fabrication and handling techniques, (5) to verify the adequacy of the backup structure with respect to restraint and thermal expansion capability, and (6) to evaluate this effect during emergency operating conditions.

Fabrication of the five-tube test panel was successfully completed in July 1976, as evidenced by its ASME certification in accordance with Section I of the boiler code. The panel consists of five full pilot plant size receiver tubes with inlet and outlet manifolds and a backup structure similar to the 70-tube pilot plant panel (Figure 5-6).

Following installation of the five-tube panel into the test facility, tests were conducted covering the full range of flux levels, pressures, and temperatures anticipated for the pilot plant receiver. Temperature and pressure data were recorded on strip charts for immediate visibility and on magnetic tape for subsequent printout and CRT display. The strip chart recorders included continuous plotting recorders as well as multipoint indicating recorders. Absorbed heat loads were determined by multiplying the difference in the specific enthalpy of the water at the exit and inlet of the panel by the water flowrate. Steam temperatures as high as 593°C (1100°F) were produced without damage to the panel. All tests were conducted at approximately 10.4 MN/m^2 (1500 psig) pressure.

Figure 5-6. Five-Tube SRE Panel

Maximum predicted and experimental wall temperatures were compared for a high and a low heat flux test. The results of the comparison indicate the predicted values to be conservative over the entire range of heat fluxes.

One of the major items investigated was the ability of the panel to operate stably without inlet orifices or, if orifices were required, to determine the minimum orificing which would permit stable operation. All tests except one calibration run were conducted without orifices and indicated uniform temperatures from tube to tube confirming stable and uniform flows in each tube. Temperatures at the discharge of the tubes and at the entrance of the downcomer indicated only minor oscillations during the start transients which quickly damp out as steady-state is reached.

The time to achieve steady-state conditions depended upon the heat flux level and effluent temperature conditions. The times ranged from a maximum of approximately 12 minutes for low heat fluxes with low discharge temperatures to 7 minutes with high heat fluxes and high discharge temperatures.

The effects of passing clouds were investigated when the electrical power was reduced to approximately 10% of the test value and brought back over a total transient time of approximately 1 minute. The transient was accomplished twice with no evidence of instability or temperature overshoot occurring.

To determine the ability of the panel to survive loss of water, the water flow was terminated by shutting off the inlet water valve for 2 seconds, 10 seconds, and 60 seconds. The maximum wall temperature during the transients was 880°F (nominal 850°F). Thus, stoppages of water flow for significant time periods can be tolerated by the panel without excessive wall temperatures or radical changes in effluent discharge temperature.

In summary, the five-tube panel tests confirmed the functional capability of the thermal expansion mechanisms, flow stability during start and other transient and steady-state conditions, and tube-to-tube flow uniformity without individual tube orifices. The test results also demonstrated the ability of the panel to respond safely to rapid heat flux transients and to temporary loss of water. In addition, the heat-transfer characteristics of the panel have exceeded expectations, resulting in better performance and higher tolerance to abnormal operating conditions than predicted.

5.4.3 Full-Scale Pilot Plant Panel Testing

Objectives of the full-scale receiver tests were to (1) demonstrate pilot plant panel fabricability and transportability, (2) verify pilot plant receiver performance including compatibility with thermal storage and electrical power generating subsystem interfaces and operation at safe wall temperature with design pressure drop, (3) verify control and stability of a pilot plant segment including steady-state stability and accuracy, stability and accuracy during start and shutdown sequences, stability and safety during cloud cover transients, component dynamic characteristics, and response to standard control signals, and (4) verify thermal stress and expansion provisions.

Fabrication of the full-scale pilot plant receiver panel was completed (Figure 5-7), certified to ASME code requirements (Figure 5-8), and transported to the B-1 test facility in El Segundo, Calif., in October 1976. More than 100 thermocouples were installed on the panel to measure heated,

Figure 5-7. Completed Pilot Plant Panel

Figure 5-8. ASME Boiler Code Certification of Pilot Plant Panel

backwall, and steam temperatures. Grit-blasting of the heated surface and painting with Pyromark absorptive paint were successfully completed immediately thereafter.

Upon completion and erection of the test tower, the full-scale panel was vertically installed into the test tower and testing was begun (Figure 5-9).

Solar flux on the panel was simulated using a stack of 165 resistance heaters in front of and across the panel absorptive surface. Flow stability and uniformity testing of the 70-tube tube panel was initiated using Inconel heater elements to obtain long heater life during the low flux phase of the test program. Test results fully confirmed the expectations created by the single- and five-tube panel testing. No indications of flow instability were detected even at very low flowrates. Flow uniformity was also excellent across the panel as indicated by uniform steam discharge temperatures. Flow stability and uniformity were maintained during simulated pilot plant starts in which temperature and pressure were manually stepped from 275°C (530°F) to 325°C (620°F) at 2.76 MN/m² (400 psi) and then to 450°C (840°F) and 10.4 MN/m² (1500 psi).

Figure 5-9. Pilot Plant Panel in Test

The pilot plant panel tests provided the first opportunity to check out the panel temperature control loop using the pilot plant control valve, temperature sensors, and control electronics. Some difficulty was initially experienced in holding the steam discharge temperature within the target control band of $\pm 28^{\circ}\text{C}$ (50°F) at low panel powers due to the low flowrates. Adjustment of the control loop gains and modification of the control valve poppet size corrected the problem in later tests. Temperature control within $\pm 17^{\circ}\text{C}$ (30°F) of the set point was maintained automatically even with instantaneous set-point shifts from 315°C to 515°C (600°F - 960°F) and back again, and with commanded variations of $\pm 20\%$ in input power level and back pressure.

The fatigue life capability of the basic panel design had been demonstrated during the five-tube panel test program under high flux levels up to 90% of the maximum flux anticipated for the pilot plant. Maximum hot wall tube temperatures during those tests were 103°C (185°F) below the predicted values indicating a wide margin on panel fatigue life. Full panel test results confirmed the earlier margin in heat transfer characteristics and fatigue life shown by the earlier five-tube panel tests. Tube wall differential temperatures were well below the predicted values indicating a comfortable margin in the fatigue life of the pilot plant panel.

5.4.4 Absorber Surface Coating Test

The objective of the absorptivity surface coating test was to determine the effects, if any, of high-intensity solar radiation on candidate panel surface absorptivity coatings. The surface coating test article consisted of an Incoloy 800 bar, grit-blasted and painted on the heated surface with both S-31 and Pyromark paints. Four thermocouples were installed to measure the heated surface temperature.

The absorptive surface durability test article was installed in the White Sands solar furnace test facility in October, 1976. Cyclic solar exposure of the candidate absorptive coatings to concentrated sunlight at a level of 0.3 MW/m^2 was begun shortly thereafter.

An interim measurement of the absorptivity of the test article coatings was made in March 1977 after approximately four months of testing. No appreciable change was noted in the absorptivity of either coating.

5.4.5 Receiver SRE Conclusions

The results of the SRE testing conducted on the pilot plant receiver test articles lead to the following conclusions:

- A. The fabrication methods and materials selected for the pilot plant receiver are compatible with the production of a boiler acceptable to Section 1 of the ASME Boiler Code.
- B. Transportation and handling of the panel in both urban and business areas are practical and can be accomplished without specialized equipment.
- C. The devices which provide for thermal expansion of the panel function very satisfactorily.
- D. The receiver surface coating maintains its high absorptivity and structural integrity under extended exposure to cyclic and highly concentrated solar insolation.
- E. The panel can operate at maximum pilot plant heat fluxes with tube temperatures well below the predicted values. Differential tube temperatures were such as to ensure a 30-year panel life.
- F. During steady-state operation, flow is uniform from tube to tube and is stable, even under extreme variations in pressure, flow or solar insolation.
- G. The panel can survive flow cessations for significant periods without excessive temperatures or damage.
- H. The panel control loop will automatically maintain steam outlet conditions within specified limits under anticipated transient conditions of pressure flow or insolation.
- I. The panel can reach steady-state temperature conditions in 7 to 12 minutes (depending on the heat flux) after a constant heat load is applied.

In summary, the SRE tests have resulted in a high confidence in satisfactory operation of the pilot plant receiver segment and thus, in the commercial receiver design concept.

Section 6

THERMAL STORAGE SUBSYSTEM

6.1 INTRODUCTION

The primary function of the thermal storage subsystem (TSS) is to extend the solar plant's effectiveness into periods with little or no insolation. In addition, the TSS can act as a buffer between the receiver and the electrical power generation subsystems to protect the turbine from rapid variations in inlet conditions due to rapid- and short-term changes in solar energy caused by clouds passing over the collector field.

This section of the report summarizes the conception, design, and testing of the McDonnell Douglas/Rocketdyne solar thermal storage concept (Figure 6-1) during the period from July, 1975 through June, 1977.

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Figure 6-1. Thermal Storage Subsystem

6.2 HISTORICAL SUMMARY

The preliminary design baseline for the 10 MWe pilot plant thermal storage subsystem was completed during the first quarter of the contract period and accepted by DoE/Sandia at the preliminary design baseline review in October, 1975.

Compatibility testing of candidate heat transfer fluids and materials of construction was also begun in the first contract quarter. Tests to investigate the heat transfer fluids surface fouling characteristics were begun in December 1975.

The conceptual and detail designs of the procedures and hardware for the thermal storage subsystem research experiments were completed on schedule and approved by DoE/Sandia during the second and third quarters of the contract. Procurement and fabrication were begun immediately after approval of the detail design.

Following a procurement competition, the contract for the model subsystem thermal storage tank was awarded and fabrication was begun in April, 1976. Construction and refurbishment activities were concurrently begun at the subsystem test site at Rocketdyne's Santa Susana Field Test Laboratory.

The thermal storage tank was delivered to the test site in early summer, 1976. Installation and hookup of the tank proceeded smoothly, with loading of the rock/sand bed completed in much less time than anticipated. All fabrication and installation of test hardware were completed by mid-September 1976, and facility activation and checkout were begun.

Following the completion of test facility activation in October, the model subsystem was in continuous operation until the completion of test activities on December 17, 1976. The data confirmed that the test program met or exceeded all program objectives and that the design of the dual-media thermal storage subsystem is more than capable of achieving pilot plant performance requirements at the lowest cost.

At customer direction, prequalification testing of the candidate heat transfer fluids was extended through June 1978 and was expanded to include an additional fluid (Mobiltherm 123), to construct and operate a laboratory scale flow loop for greater depth in the compatibility and fouling tests, and to develop criteria and procedures for monitoring the heat transfer fluid condition during plant operation. By June 1977, after over 13,000 hours of test time, samples of the original stability and fouling tests had developed no problems. The summary thermal storage program schedule is shown in Figure 6-2.

6.3 DESIGN SUMMARY

6.3.1 Concept Rationale

Thermal storage concepts can generally be classified into three categories: sensible heat, latent heat (phase change), and thermochemical (reversible chemical reactions). Of these, sensible heat systems are clearly within the current state-of-the-art and present the least technical risk and thus the

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Figure 6-2. Thermal Storage Schedule

least cost and development risk. Both latent heat and thermochemical storage systems will require considerable additional development before they can be evaluated in detail and considered for implementation on a large scale. Therefore, sensible heat thermal storage was selected for this application.

It would be uneconomical to store the steam produced by the central receiver directly in the quantities required for thermal storage, or to allow it to heat directly another storage medium, such as a rock bed, because of the very large, high-pressure tank which would be required. Therefore, some other fluid is required to transfer energy from the steam, and later to supply energy to produce steam for extended operation from storage. A variety of organic and inorganic liquids were considered. The large volumes of thermal storage fluid required in a commercial plant make fluid cost a driving parameter. The media cost for an all-liquid thermal storage subsystem can be of the order of 75% of the cost of the entire subsystem.

The thermal storage concept selected uses a low-cost stationary solid bed to store most of the energy, with a suitable liquid to transfer energy into and out of the bed (and to store part of the energy directly). This dual-medium type of system (patent pending to Rocketdyne and MDAC) combines advantages of a low-cost solid with the flexibility, low pumping power, and moderate heat exchanger requirements of a liquid energy storage system.

Conceptually, in its simplest form, the system uses a bed of an inexpensive solid (c.g., rock, ore, metal scraps, etc.). An appropriate high-temperature liquid fills the voids in the bed and circulates through the bed to deposit or withdraw energy (Figure 6-1).

In the cyclical operation, heating of the bed (charging) is achieved by removing lower temperature fluid from the bottom of the bed, heating it in a heat exchanger with steam from the receiver, and returning the fluid to the top of the tank. A fairly sharp temperature transition (a thermocline) is maintained naturally between hot and cold fluid because of the lower density of hot fluid. This thermocline moves downward through the bed during charging and upward during extraction. When the storage unit is completely

charged, all of the bed and the fluid are at the maximum temperature and the thermocline does not exist. The extraction loop utilizes the fluid to remove energy from the storage unit and produces steam for power plant operation or other plant functions such as equipment heating.

The large cost savings for this type of thermal storage results principally from two factors: (1) replacement of about 75% of the expensive storage liquid with virtually cost-free rock, and (2) use of the thermocline principle to reduce the tankage volume substantially, compared to a system with separate tanks for hot and cold storage.

One of the important decisions which required considerable analysis was selection of the maximum storage temperature, which fixes, within a small range, the maximum temperature of steam produced from thermal storage.

Commercially available organic heat transfer fluids which are liquid at ambient temperature have maximum operating temperatures of about 316°C (600°F). Storage temperatures above this region require alternatives which introduce some negative features. The principal alternatives are:

(1) organic fluids which are solid at ambient temperatures (with higher costs and the problems of initial startup and avoiding solidification during operation); (2) fluids with higher vapor pressures, e. g. , Dowtherm A (with higher fluid costs and much higher storage container costs); (3) inorganic salts and salt mixtures (with problems of initial startup and avoiding solidification during operations, plus usually higher medium costs).

The improvements in plant efficiency and energy output which can be obtained with higher storage temperatures must be great enough to justify the higher costs necessary to provide the higher storage temperature. Design analyses showed that the baseline system we used for this program i. e. , dual-medium, (low-cost organic fluid and rock with storage at about 316°C), provided lower-cost energy than any of the higher temperature alternatives. The only exception to this conclusion was the addition of a higher temperature molten salt/rock stage in addition to the baseline dual-medium stage; however, this option would require some additional technological development and demonstration.

6.3.2 Commercial Thermal Storage Subsystem Design Summary

The general requirement on the thermal storage subsystem is that it provide a means of transferring to stored thermal energy a portion of the thermal output from the receiver subsystem and, subsequently, transferring stored thermal energy to create steam in a form suitable for generating electrical power with the conventional turbine-generator in the EPGS.

The commercial thermal storage subsystem is required to have an extractable storage capacity of at least 1857 MWHt, which is comprised of 100 MWHt to provide a turbine hot start and 1757 MWHt to permit the turbine-generator to supply 70 MWe net (76 MWe gross) for six hours following turbine startup. The required charging rates are 12.5 MWt (rated steam operation) to 255 MWt (derated steam operation). Required discharging rates are 31.1 to 285 MWt. The maximum allowable heat loss is 2% of extractable capacity in 24 hours, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.42 MWt for approximately 12 hours.

The commercial plant TSS employs sensible heat storage using dual liquid and solid media for the heat storage in a parallel set of four tanks, each utilizing the thermocline principle to provide high and constant temperature extractable energy.

Figure 6-3 is a functional schematic of the thermal storage subsystem, showing all major types of components and major control concepts.

The TSS is divided into nine basic types of components which are: (1) thermal storage units, i. e., tanks which store and dispense thermal energy via the Caloria HT43 heat transfer fluid (Figure 6-4); (2) ullage maintenance units which provide an inert nitrogen gas cover over the fluid surface in the tanks; (3) fluid maintenance units which remove suspended and dissolved impurities from the fluid; (4) a desuperheater which limits incoming steam temperature; (5) thermal heat transfer fluid; (6) the thermal charging loop comprising the charging fluid pumps and associated equipment connecting the thermal storage heaters to the thermal storage unit; (7) steam generators which transfer heat from the fluid to make steam for the power plant; (8) the

Figure 6-3. Thermal Storage Subsystem Schematic

Figure 6-4. Typical 100-MWe Commercial Thermal Storage Unit (1 of 4)

extraction loop comprising the extraction fluid pumps and associated equipment connecting the steam generators to the thermal storage unit; and (9) the controls and instrumentation for the subsystem.

Table 6-1 summarizes the principal characteristics of the subsystem and major components. More detailed definition of the commercial thermal storage subsystem may be found in the McDonnell Douglas Preliminary Design Report, Volume V, MDC G6776, dated May, 1977.

6.3.3 Pilot Plant Thermal Storage Subsystem Design Summary

Since the pilot plant objective is to prove the viability of the commercial plant design, the same basic design concept was used for both commercial and pilot plant subsystems. The 10 MWe pilot plant TSS also employs sensible-heat storage using dual liquid and solid media and applying the thermocline principle to provide high-temperature, extractable energy. The functional schematics of the pilot and commercial TSS are also essentially identical (Ref: Figure 6-3) with the approximately tenfold difference in capacity accomplished by the use of modular components, with fewer numbers of components in the pilot plant, often of similar size to the more numerous modules in the commercial plant subsystem.

The pilot plant thermal storage subsystem is required to have an extractable storage capacity of at least 103.8 MWh_t, which is comprised of 7.5 MWh_t to provide a turbine hot start and 96.3 MWh_t to permit the turbine-generator to supply 7 MWe net (7.8 MWe gross) for three hours following turbine startup. The required charging rates are 1 MW_t to 30 MW_t. Required discharge rates are 3.1 MW_t to 32.1 MW_t. The maximum allowable heat loss is 3% of extractable capacity in 24 hours, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.33 MW_t for approximately 16 hours.

A major requirement is to accept steam from the pilot plant receiver at 343/510°C (650/950°F) and 10.1 MPa (1465 psia), where the two sets of temperatures correspond to derated and rated steam operation, respectively.

Table 6-1

COMMERCIAL PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Thermal storage unit	Four identical units; each a cylindrical tank, axis vertical, installed above ground, 27.6m (90.5 ft) diameter by 18.3m (60.0 ft) high; 10,900 m ³ (386,000 ft ³ , 2,890,000 gal.) volume; each containing 20.3x10 ⁶ kg (22,300 ton) of granite rock and coarse silica sand (approximately 2:1 rock:sand by volume) and 2.2x10 ⁶ liters (583,000 gal.) of Caloria HT43 heat transfer fluid. Fluid temperature range: 232 to 316°C (450 to 600°F). Fabricated of ASTM A537-70 Grade B structural steel by field-welded construction
Ullage maintenance unit	Storage and control of ullage gas with compressed gas storage at 1.20 MPa (175 psia); tank pressure control, venting, inert gas (nitrogen) control, volatile vapor recovery and control
Fluid maintenance unit	Full-flow, continuous filtration with dual 80-mesh filters in main fluid line upstream of pump; periodic distillation with vacuum distillation unit in side-stream to remove polymerized materials; periodic fluid makeup
Desuperheater	Direct-contact mixing chamber with water injected through multiple atomizing nozzles into superheated steam; single unit; ten nozzles
Thermal storage heater	Five identical exchangers in parallel; each is TEMA type DFU, with removable U-tube bundle, 2 shell passes, 6 tube passes; steam/water on tube side; 1672 m ² (18,000 ft ²) heat transfer area per exchanger; carbon steel
Steam generator	Five modules in parallel. Each module consists of three separate stages in series consisting of feedwater preheater, boiler, and superheater; steam/water on shell side; carbon steel. Preheater is straight tube, floating head, counterflow exchanger with 435 m ² (4684 ft ²) heat transfer area per exchanger Boiler is horizontal U-tube kettle boiler with 1204 m ² (12,948 ft ²) heat transfer area per exchanger Superheater is horizontal U-tube, crossflow exchanger with 594 m ² (6389 ft ²) heat transfer area per exchanger
Fluid charging loop Pump	Five identical pumps in parallel; centrifugal, high temperature type, with single-speed electric motors; each pump has flow of 260 kg/s (600 lb/s); and 0.19 MWe (260 hp) motor input at maximum charging rate (51 MWt)
Fluid extraction loop Pump	Five identical pumps in parallel; centrifugal, high temperature type, with single-speed electric motors; each pump has flow of 216 kg/s (490 lb/s), 0.19 MWe (250 hp) motor input at maximum extraction rate (57 MWt)

*Required input motor power; not full motor capacity

Another major requirement is to supply steam from the TSS steam generator for the turbine at 277°C (530°F) and 2.76 MPa (400 psia).

Table 6-2 summarizes the principal characteristics of the pilot plant TSS and its major components. Almost all of the components in the thermal storage subsystem are conventional, commercial items. Their fabrication and installation introduce no new requirements different from standard practices in the chemical processing and petroleum industries.

The thermal storage tank is conventional, using standard carbon steel structural materials. The tank fabrication and installation are combined, following standard practice, by field welding various sections, including plate sections previously rolled to the proper radius of curvature. The charging heat exchangers, steam generator units (separate units for pre-heating, boiling, and superheating), pumps, piping, valves, controls, and instrumentation are all commercial items. Their fabrication will follow standard vendor practices, and their installation will be standard.

The ullage maintenance unit and fluid maintenance unit are comprised of standard commercial components; tanks, valves, piping, compressors, filters, and a heat transfer fluid distillation unit. Although the combinations of components are unique to perform the particular functions of these two subsystem elements, the fabrication and installation of the individual components introduces nothing new to the construction industry.

More detailed information on the design may be found in the McDonnell Douglas Preliminary Design Report, Volume V, document number MDC G6776, dated May, 1977.

6.4 THERMAL STORAGE SUBSYSTEM RESEARCH EXPERIMENTS

The SRE were designed to minimize any risk inherent in the commercial and pilot plant designs by establishing a firm technical data base in each design area. Fortunately, the TSS concept has a sound base of commercial experience. Most of the equipment, materials, flow loops, and controls include commercial components that are available; therefore, unique features of this program requiring development were minimized.

Table 6-2

PILOT PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Thermal storage unit	Single cylindrical tank, axis vertical, installed above ground, 15.2 m (50.0 ft) diameter by 13.4 m (44.0 ft) high; 2450 m ³ (86,400 ft ³ , 646,000 gal.) volume; contains 4.53 x 10 ⁶ kg (4990 ton) of granite rock and coarse silica sand (approximately 2:1 rock: sand by volume) and 525,000 liters (139,000 gal) of Caloria HT43 heat transfer fluid. Fluid temperature range: 218 to 302°C (425 to 575°F). Fabricated of ASTM A537-70 Grade B structural steel by field-welded construction.
Ullage maintenance unit	Storage and control of ullage gas with compressed gas storage at 1.20 MPa (175 psia); tank pressure control, venting, inert gas (nitrogen) control, volatile vapor recovery and control.
Fluid maintenance unit	Full-flow, continuous filtration with dual 80-mesh filters upstream of pump; periodic distillation with vacuum distillation unit in side-stream to remove polymerized materials; periodic fluid makeup.
Desuperheater	Direct contact mixing chamber with water injected through multiple atomizing nozzles into superheated steam; single unit; three nozzles.
Thermal storage heater	Two identical exchangers in parallel; each is TEMA type DFU, with removable U-tube bundle, 2 shell passes, 6 tube passes; steam/water on tube side; 464 m ² (5000 ft ²) heat transfer area per exchanger; carbon steel.
Steam generator	Three-stage (series) modules each with separate feed-water preheater, boiler, and superheater; 2 modules in parallel; steam/water on shell side; carbon steel. Preheater is straight tube, floating head counterflow exchanger with 196 m ² (2106 ft ²) heat transfer area per exchanger. Boiler is horizontal U-tube kettle boiler with 791 m ² (8513 ft ²) heat transfer area per exchanger. Superheater is horizontal U-tube, crossflow exchanger with 84 m ² (904 ft ²) heat transfer area per exchanger.
Fluid charging loop pump	Two identical pumps in parallel; centrifugal, high temperature type, with dual-speed electric motors; each pump has flow of 64 kg/s (141 lb/s), and 0.060 MWe (80 hp) *motor input at maximum charging rate (15 MWt).
Fluid extraction loop pump	Two identical pumps in parallel; centrifugal, high temperature type, with single-speed electric motors; each pump has flow of 70 kg/s (155 lb/s), and 0.052 MWe (70 hp) *motor input at maximum extraction rate (16.1 MWt).

*Required input power; not full motor capacity

The SRE program consisted of two series of tests: (1) prequalification tests and (2) model subsystem tests. The major test objectives were:

- Evaluate heat transfer fluid thermal stability, compatibility, and surface fouling characteristics.
- Evaluate charging and extraction capabilities of a scalable thermal storage unit (TSU).
- Obtain performance of the TSU over all ranges of equivalent operating conditions in the pilot plant.
- Demonstrate stable operation for high, low, intermittent, and no insolation conditions.
- Evaluate the fluid under operational conditions.
- Determine the strain from rock/wall interactions.

All tests were completed successfully, and all test objectives were met.

6.4.1 Prequalification Tests

Thermal stability and compatibility tests were designed on a laboratory scale to demonstrate candidate heat transfer fluid characteristics when subjected to long-term exposure to operating temperatures. The tests were initially conducted on three commercially available fluids which were believed to be capable of meeting the requirements of the thermal storage system: Exxon Caloria HT43, Monsanto Therminol 55, and Monsanto Therminol 66. At customer direction, prequalification testing has been expanded to include Mobiltherm 123 and extended into 1978.

The objectives of the fluid thermal stability and compatibility tests are to (1) determine the ability of the heat transfer fluids Caloria HT43 (formerly Humbletherm 500), Therminol 55, and Therminol 66 to function stably at 288 to 316°C (500 to 600°F) for extended periods of time; (2) assess the high-temperature, long-term compatibility of these heat transfer fluids with rock and materials of construction (stainless steel, carbon steel) which will be in contact with fluid in the thermal storage unit; and (3) determine the rate and extent of fouling of heat transfer surfaces by Caloria HT43 and Therminol 66.

Stability tests of the heat transfer fluids are being conducted at 288°, 302°, and 316°C (550°, 575°, and 600°F) in the presence and absence of rock aggregate, stainless steel, and carbon steel. The tests are being performed at atmospheric pressure under a nitrogen gas atmosphere (Figure 6-5). The loss of fluid from each of the flasks over the heating time is determined by periodic weighings. Kinematic viscosity of the fluids is measured with a viscosimeter to further assess the effect of heating on the fluid.

Using the test data and the temperature dependence derived from the data, estimates were made of the expected weight loss for each of the three fluids with both pilot plant and commercial plant cycles. These values are given in the top half of Table 6-3. The bottom half of this table translates these makeup requirements into relative costs, using the latest costs supplied by the manufacturers (\$1.02/gal for Caloria HT43, \$1.60/gal for Therminol 55, and \$5.70/gal for Therminol 66). It can be seen that both weight loss and cost of makeup fluid makes Therminol 55 unattractive. Therminol 66 has the lowest pilot plant weight loss of the three fluids, but its higher price would result in higher fluid makeup cost than for the baseline fluid, Caloria HT43. Pilot plant replenishment costs for HT43 have been calculated to be no more than 0.014% of pilot plant capital cost.

Table 6-3
COMPARISON OF FLUID MAKEUP COSTS

Fluid	Caloria HT43	Therminol 55	Therminol 66
Estimated weight loss (percent/year)			
Pilot plant (218 to 302°C)	7.0	19.7	6.07
Commercial plant (232 to 316°C)	12.9	47.6	30.9
Relative cost of makeup fluid			
Pilot plant (218 to 302°C)	1.0	4.4	5.35
Commercial plant (232 to 316°C)	1.84	10.7	24.7

Surface fouling from degradation of the heat transfer fluid is most likely to occur at the points of highest surface temperatures. Tests are being performed to determine the extent and rate of fouling of electric heaters immersed in the heat transfer fluids as a function of the surface temperature of the heater for two fluids: Caloria HT43 and Therminol 66 (Figure 6-6).

Figure 6-5. Fluid Thermal Stability and Compatibility Setup

Figure 6-6. Surface Fouling Test Setup

Six fouling tests are being conducted, three with Caloria HT43 and three with Therminol 66. The surface temperatures of the electrical heaters are controlled at 316°C , 329°C , and 343°C (600°F , 625°F , and 650°F). The manufacturer's recommended maximum film temperatures are 360°C (680°F) for Caloria HT43 and 374°C (705°F) for Therminol 66.

Heat exchanger surface fouling tests have exceeded 13,000 hours on the longest-lived test samples. Some minor surface deposit has continued to be visible near the bottom (no flow zone) of the HT43 heaters; however, no detectable change in either extent of deposits or heat transfer rate has occurred during the report period. No visible fouling deposits of any kind have formed on the Therminol 66 samples. Apparatus for three additional fouling tests with Mobiltherm 123 has been assembled and the tests should begin in the third quarter of 1977.

In summary, the SRE prequalification testing has fully confirmed the choice of Caloria HT43 as the baseline heat transfer fluid, and demonstrated that it can fulfill the requirements of this solar power application. This fluid was found to have excellent stability and compatibility with rocks and materials of construction in tests up to 316°C (600°F) and for extended test durations. In addition, Caloria HT43 was found to develop no problems in fouling of heat transfer surfaces.

6.4.2 SRE Model Subsystem Tests

The basic objectives of the thermal storage SRE model system tests were to:

- A. Evaluate charging and extraction capabilities of a scalable TSU.
- B. Obtain performance of the TSU over all ranges of equivalent operating conditions in the pilot plant.
- C. Demonstrate stable operation for high, low, intermittent, and no insolation conditions.
- D. Demonstrate changeover capabilities from one operating mode to another and emergency response modes.
- E. Develop requirements and techniques for treating the heat transfer fluid to permit a 30-year lifetime with makeup.
- F. Determine the additional strain on container walls from tank/rock interaction during repeated thermal cycling.

The thermal storage model subsystem consists of a heat transfer fluid heater, steam generator, thermal storage unit, two fluid circulation pumps, ullage maintenance unit, fluid maintenance unit, and associated controls, valves, and sensors (Figure 6-7).

A basic flow schematic of the subsystem for the SRE is given in Figure 6-8. The SRE model subsystem has the same three major constituents as the pilot plant and commercial TSS. They are the central TSU, the charging loop, and the energy extraction loop.

Fabrication of the SRE thermal storage tank was begun by a commercial tank fabricator in April 1976 in accordance with previously approved detailed drawings. The tank was constructed by rolling and welding courses of the same structural steel (ASTM 537-70) planned for the pilot plant. Tensile pull tests of weld specimens using various weld rods indicated that all welds would meet or exceed design strength properties of the steel.

The basic TSU tank was completed and delivered to the Santa Susana test site in early June 1976. The TSU was installed, pressure checked, and connected to the fluid circulation pumps and the remaining portions of the system by the piping contractor.

The rock/sand bed, with its numerous thermocouples, was installed in the TSU with no difficulties, and was completed in much less time than originally estimated.

Completion of the fabrication, procurement, facility buildup, and installation of all other equipment for testing the SRE model subsystem, permitted test facility activation and checkout to begin in mid-September 1976. Hot testing began as planned by October 15.

All test objectives were met or exceeded in a two-month test program completed in December 1976. A large variety of tests were conducted with thermal charging and discharging rates from 0.1 to 2 MWt and with hold periods up to 144 hours. The performance of the subsystem was excellent. Sharp, stable thermoclines were present, even during partial charging and discharging, with variable rates, and under repeated cycling.

Figure 6-7. Thermal Storage SRE Test Site

Figure 6-8. Thermal Storage SRE Subsystem Schematic

Figure 6-9 illustrates thermocline shape and travel during a 100 percent extraction compared with pretest predictions. Figure 6-10 shows the heat transfer fluid temperature leaving the top and entering the bottom of the TSU. The extraction fluid temperature was maintained within 2°C (3.6°F) of the target temperature of 302°C (575°F) for more than three hours and within the minimum desired temperature of 293°C (560°F) for almost four hours at an extraction rate of 1.34 MW. The performance of the TSU was even better than expected for a unit of this size. The unit delivered over 5.1 MWh of energy, which exceeded the design goal of a minimum of 4 MWh.

Other conclusions from the SRE subsystem tests:

1. The practicality and high performance of the dual-medium thermal storage concept have been demonstrated on a significantly large scale. Scale up to the 10-MWe pilot plant can be made with high confidence.
2. The development and vertical movement of a sharp thermal gradient or thermocline in a dual-medium (liquid-solid) system is a predictable and reproducible phenomenon.

Figure 6-9. Typical Extraction Thermoclines, SRE Subsystem Tests and Design Predictions

Figure 6-10. Typical SRE Performance Data During Extraction

3. Thermocline integrity and stability are high enough to provide high-energy recovery performance for daily operation.
4. Partial charging and extraction, and repeated cycling, do not significantly degrade the thermocline.
5. Fluid flow and temperature uniformity are very high across the TSU cross-section.

In summary, the detailed analysis of the test data has confirmed that the SRE testing met or exceeded all program objectives and has conclusively demonstrated the integrity of the dual media/thermocline thermal storage design concept.

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