10/12 26. 11/15

SAN/1110-76/3

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

**Annual Progress Report (Final) for Period Ending
September 30, 1976**

February 1977

Work Performed Under Contract No. EY-76-C-03-1110

**Martin Marietta Corporation
Denver, Colorado**

**ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Division of Solar Energy**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

This report has been reproduced directly from the best available copy.

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161

Price: Paper Copy \$7.75 (domestic)
\$10.25 (foreign)
Microfiche \$3.00 (domestic)
\$4.50 (foreign)

CENTRAL RECEIVER SOLAR THERMAL
POWER SYSTEM, PHASE I

Annual Progress Report (Final)

PERIOD ENDING
SEPT 30, 1976

February 1977

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

MARTIN MARIETTA CORPORATION
P. O. Box 179
Denver, Colorado 80201

EP
DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

CONTENTS

	Page
I. INTRODUCTION	I-1
II. 10 MWe PILOT PLANT	II-1
A. INTRODUCTION	II-1
B. PILOT PLANT DESCRIPTION	II-2
1. Physical Description	II-2
a. Collector Subsystem	II-4
b. Receiver Subsystem	II-4
c. Thermal Storage Subsystem	II-4
d. Electrical Power Generation Subsystem	II-6
e. Master Control Subsystem	II-6
2. Pilot Plant System Schematic	II-6
a. Receiver Operation	II-7
b. Thermal Storage Subsystem Operation	II-7
c. Electrical Power Generation Subsystem	II-10
C. PERFORMANCE	II-10
1. General	II-10
2. Design Point Sizing for the Pilot Plant	II-11
3. Performance Calculations	II-12
4. Yearly Performance	II-14
D. PILOT PLANT OPERATION	II-20
1. Operational Modes	II-20
2. Pilot Plant Start-up Analysis	II-24
a. Plant Start-up from Hot Standby	II-24
b. Plant Cold Start	II-25
c. Plant Start-up from Thermal Storage	II-25
3. Pilot Plant Stand-by and Emergency Power Operations	II-25
E. SYSTEM EFFECTIVENESS ANALYSIS	II-31
F. 10 MWe PILOT PLANT COST ANALYSIS	II-31

III.	COLLECTOR SUBSYSTEM	III-1
	A. SUMMARY OF COLLECTOR SUBSYSTEM ACTIVITIES, JULY 1975 THROUGH JUNE 1976	III-1
	1. Preliminary Pilot Plant Definition	III-1
	2. Subsystem Research Experiment (SRE)	III-5
	a. Conceptual Design Phase	III-5
	1) Wind Tunnel Test	III-5
	2) Drive Mechanism Tradeoff Study	III-6
	3) Structural Tradeoff Study	III-6
	4) Control System Studies	III-6
	5) Heliostat Mirror Evaluation	III-7
	b. Detail Design Phase - October 1975 through March 1976	III-7
	c. Fabrication Phase - March through June 1976	III-8
	d. Test Phase - June 1976 through December 1976	III-12
	3. Solar Thermal Test Facility (STTF)	III-12
	B. SUMMARY OF COLLECTOR SUBSYSTEM ACTIVITIES, JULY 1976 THROUGH SEPTEMBER 1976	III-16
	1. Calibration of SRE Calorimeter/Radiometer Test Facility	III-16
	a. Calorimeter Test Facility	III-16
	b. Radiometer Test Facility	III-22
	c. Results of Calorimeter and Radiometer Tests	III-22
	2. Supplemental Tests	III-26
	a. Sheet vs Float Glass	III-26
	b. Effect of Pull Plate Size on Image	III-27
	c. Fixed Focus Mirror Development	III-27
IV.	RECEIVER SUBSYSTEM REQUIREMENTS	IV-1
	A. PILOT PLANT RECEIVER	IV-1
	1. Description of Concept and General Requirements	IV-1
	2. Input Thermal Energy Characteristics	IV-4
	B. RESEARCH EXPERIMENT TEST SYSTEM	IV-10
	1. Purpose	IV-10
	2. Test System	IV-10
	3. Test Support Equipment	IV-12
V.	RECEIVER SUBSYSTEM	V-1
	A. INTRODUCTION	V-1

B. PILOT PLANT RECEIVER	V-1
1. General Design Description	V-1
2. Thermal/Hydraulic Analysis and Design	V-4
a. Basis of Analysis and Design	V-4
b. Superheater Surface Arrangement	V-4
c. Boiler Circuitry Arrangement	V-5
d. Start-up Transient Analysis	V-6
3. Structural Analysis and Design	V-7
a. Applicable Codes and Standards	V-7
b. Superheater Analysis	V-7
c. Boiler (Waterwall) Analysis	V-9
d. Other Pressure Parts Calculations	V-9
4. Preliminary Seismic Analysis and Support Structure Design	V-9
a. Interface Background	V-9
b. Prediction of Natural Frequencies	V-10
c. Preliminary Seismic Analysis of Steel Supporting Structure	V-10
d. Trade-Off of Concrete vs. Steel Supporting Structure	V-14
C. RESEARCH EXPERIMENT RECEIVER	V-15
1. Objectives, Basic Requirements, and Approach	V-15
2. Conceptual Design and Related Analyses	V-15
a. Scaling Criteria	V-15
b. Mechanical Arrangement	V-16
c. Thermal/Hydraulic Analysis	V-17
d. Structural Analysis	V-18
3. Test Plan	V-18
a. Test Objectives and Summary	V-18
b. Test System and Facility	V-18
c. Description of the Receiver	V-18
d. Receiver Operation, Controls, and Safety	V-18
e. Instrumentation	V-19
f. Test Description	V-19
g. Data Acquisition and Logging	V-20
h. Test Personnel and Scheduling	V-20
4. Test Procedures	V-20
a. Pretest Requirements	V-20
b. Start-up and Shutdown Procedures	V-21
c. Data Acquisition	V-21
d. Boiler Water Quality Control	V-21
e. Description of the Scheduled Test Runs	V-21

	5. Detailed Design, Procurement, and Fabrication . . .	V-21
	6. Field Assembly and Erection	V-27
VI.	THERMAL STORAGE SUBSYSTEM	VI-1
	A. INTRODUCTION	VI-1
	1. Program Objectives	VI-1
	2. Summary of Progress Through Reporting Date	VI-2
	a. Pilot Plant Preliminary Design	VI-2
	b. Thermal Storage Subsystem Research Experiment	VI-3
	B. PILOT PLANT THERMAL STORAGE SUBSYSTEM	VI-3
	1. Requirements for Pilot Plant Thermal Storage Sub- system	VI-4
	2. Thermal Storage Feasibility Studies	VI-5
	3. Pilot Plant Thermal Storage Subsystem Baseline Design	VI-8
	4. Storage Media Selection	VI-16
	5. Pilot Plant Performance Studies	VI-18
	C. THE THERMAL STORAGE SUBSYSTEM RESEARCH EXPERIMENT	VI-18
	1. Research Experiment Requirements	VI-18
	2. Description of the Thermal Storage Subsystem Research Experiment	VI-20
	3. Research Experiment Test Plan and Procedures	VI-35
	4. Construction of Research Experiment	VI-36
	D. HYDROCARBON STORAGE FLUIDS	VI-38
	E. SCHEDULE FOR PROGRAM COMPLETION	VI-38
VII.	BALANCE OF PLANT INCLUDING ELECTRIC POWER GENERATION SUB- SYSTEM	VII-1
	A. INTRODUCTION	VII-1
	B. CONCEPTUAL DESIGN OF EPCS	VII-1
	1. Turbine Selection, Cycle Studies, and Heat Re- jection System Selection	VII-2
	a. Turbine Selection	VII-2

b. Cycle Studies	VII-3
c. Heat Rejection System	VII-4
2. Commercial Plant Applicability	VII-8
3. Startup and Modes of Operation	VII-8
a. Startup	VII-8
b. Modes of Operation	VII-9
4. EPGS Systems	VII-9
5. Auxiliary Power Requirements	VII-11
6. Failure Modes and Effects Analysis	VII-14
C. CONCEPTUAL PLOT PLAN DEVELOPMENT	VII-14
1. The Plot Plan	VII-14
2. General Plant Arrangement	VII-14
D. COLLECTOR FOUNDATION CONCEPTUAL DESIGN	VII-17
1. Design Criteria	VII-17
a. Wind Conditions	VII-17
b. Soil Type	VII-17
c. Site Topography	VII-18
2. Conceptual Design	VII-18
E. RECEIVER TOWER CONCEPTUAL DESIGN	VII-18
1. Receiver Tower	VII-18
2. Riser and Downcomer Piping	VII-21
F. PRELIMINARY ENVIRONMENTAL IMPACT IDENTIFICATION AND SITING CRITERIA	VII-22
1. Introduction	VII-22
2. Impacts to the Physical and Chemical Environment	VII-22
a. Topography	VII-22
b. Soils	VII-25
c. Microclimatology	VII-25
d. Other	VII-25
3. Impacts to the Biological Environment	VII-25
a. Terrestrial Ecology	VII-25
b. Aquatic Ecology	VII-26
4. Impacts to the Social Environment	VII-26
a. Land Use	VII-26
b. Aesthetics	VII-26
c. Scientific/Cultural	VII-26

G. EPGS OPTIMIZATION

1. Introduction	VII-26
2. Optimization Areas	VII-27
3. Summary of Recommendations	VII-31
a. Turbine Related Studies	VII-31
b. Condenser Studies	VII-32
c. Plant Control Studies	VII-33
d. Feedwater Heater Optimization	VII-34
e. Pump Selection Studies	VII-36
f. Pipe Sizing Studies	VII-38
g. Auxiliary Steam Supply Studies	VII-39

I. RECEIVER TOWER STRUCTURAL OPTIMIZATION STUDY VII-40

1. Introduction	VII-39
2. Description of Structural Systems	VII-40
3. Comparison of Structural Systems	VII-42
4. Receiver	VII-42
5. Receiver Seismic Loads	VII-42
6. Analysis and Results	VII-42

J. CONCEPTUAL PLANT COST VII-42

Tables

II-1	Collector Field Effective Cosine for Inyokern	II-8
II-2	Collector Field Heliostat Shading for Inyokern	II-8
II-3	Pilot Plant Performance: Solstice & Equinox	II-18
III-1	Error Budget for Collector Field and Heliostats	III-2
III-2	Characteristics of SRE Heliostats	III-11
III-3	Locations of SRE Heliostats with Respect to 10 MWe Collec- tor Field	III-14
III-4	Summary of Calorimeter Data	III-14
III-5	Summary of Radiometer Data	III-25
III-6	Comparison of Calorimeter and Radiometer Test Data	III-25
III-7	Calculated Uncertainty in Measurements	III-25
III-8	Results of Fixed Focus Mirror Testing	III-30
IV-1	Receiver Requirements	IV-1
IV-2	Pilot Plant Solar Concentration Summary	IV-2
IV-3	Incident Solar Energy on Pilot Plant Receiver Subsystem	IV-9
V-1	Summary Table of Control Adjustments for a Scheduled Cold Startup Test Run	V-22

VI-1	Pilot Plant Thermal Storage Steam Flows and State Points . .	VI-5
VI-2	Problem Areas in Solid Sensible Heat and Latent Heat Salt Thermal Storage Systems	VI-6
VI-3	Summary of Documents and Design Coverage Areas Applicable to the Design of the Pilot Plant and Research Experiment Thermal Storage Subsystems	VI-15
VI-4	Room Temperature Properties and Price for Bulk Quantities of Hydrocarbon Oil	VI-17
VI-5	Heat Exchanger Description and Specifications	VI-28
VI-6	Storage Tank Description and Specifications	VI-32
VI-7	Pump Specifications and Suppliers	VI-32
VI-8	Research Experiment Detailed Procedures	VI-37
VII-1	Base Heat Balance Values	VII-1
VII-2	EPGS Auxiliary Power Requirements, kWe	VII-13
VII-3	Summary of Environmental Impacts	VII-23
VII-4	Summary of Environmental Impact Siting Criteria	VII-24
VII-5	Performance Comparison - Conceptual Design vs. Optimized Design	VII-28
VII-6	Tower Top Receiver Support Concepts Comparative Features . .	VII-43

Figures

II-1	Artist's Concept, 10 MWe Central Receiver Solar Thermal Pilot Power Plant	II-3
II-2	Pilot Plant System Schematic	II-5
II-3	Tower Shadows for Winter Solstice	II-9
II-4	Pilot Plant Performance - Design Point	II-13
II-5	Pilot Plant Electrical Output vs Temperature	II-15
II-6	Spring Equinox - Midday Insolation	II-16
II-7	Summer Solstice - Midday Insolation	II-16
II-8	Fall Equinox - Midday Insolation	II-17
II-9	Winter Solstice - Midday Insolation	II-17
II-10	Energy Collection Performance for 1 m ² Mirror 137.2 m (450 ft) Tower, Flatfield Configuration	II-19
II-11	Combined Effects of Tower Height and Collector Field on Annual Energy Delivered by Solar Collector Field	II-21
II-12	Typical Day Insolation	II-22
II-13	Typical Day Performance Profile	II-22
II-14	Morning Startup Power to Receiver	II-24
II-15	Morning Startup, Receiver	II-26
II-16	Morning Startup, EPGS	II-26
II-17	Pilot Plant Cold Startup	II-27
II-18	Turbine Start-up From Admission Steam to Design Point . . .	II-28
II-19	Electrical System Block Diagram	II-29

III-1	Conceptual Collector Field Layout for 10 MWe Pilot Plant . .	III-3
III-2	25-Mirror Heliostat, Front View	III-9
III-3	25-Mirror Heliostat, Back View	III-9
III-4	Nine-Mirror Heliostat, Front View	III-10
III-5	Nine-Mirror Heliostat, Back View	III-10
III-6	Layout of SRE Test Facility	III-13
III-7	Schematic Diagram of SRE Heliostat Control System	III-15
III-8	SRE Calorimeter	III-17
III-9	Calorimeter Receiver Panel	III-17
III-10	Calorimeter Energy Balance	III-17
III-11	Atmospheric Attenuation	III-19
III-12	Spectral Attenuation through Atmosphere	III-19
III-13	Calorimeter Front-Surface Convection Losses	III-21
III-14	Cross Section of Calorimeter Used for Back Side Conduction Loss	III-21
III-15	SRE Radiometer Boom	III-23
III-16	Radiometer Energy Balance	III-23
III-17	Radiometer Data Reduction	III-24
III-18	Flux Distribution through Center of Solar Image	III-24
III-19	Distortion of Mirror Surface vs Pull Plate Size - Computer Model	III-28
III-20	Energy Distribution as a Function of Pull Plate Diameter . .	III-28
III-21	Image Quality Testing of Prototype Fixed-Focus Mirror . . .	III-30
IV-1	Cavity Dimensions	IV-3
IV-2	Pilot Plant Receiver Heat Flux Profile	IV-6
IV-3	Receiver Surface Node and Superheater Locations	IV-8
IV-4	Research Receiver Test System Schematic	IV-11
IV-5	Test Control Console	IV-13
IV-6	Mechanical Equipment Module	IV-14
V-1	Side Elevation View - Pilot Plant Receiver	V-2
V-2	Arrangement of Superheater Passes - Pilot Plant Receiver . .	V-3
V-3	Predicted Pressures and Flow Rates for a Hot Startup Transient - Pilot Plant Receiver	V-8
V-4	Predicted Feedwater and Steam Temperatures for a Hot Start Transient - Pilot Plant Receiver	V-8
V-5	Analytical Model for Predicting Natural Frequencies of the Receiver Support Structure	V-11
V-6	Preliminary Support Structure Design and Column Locations for the Pilot Plant Receiver	V-12
V-7	Predicted Operating Parameters of the Experimental Receiver for a Scheduled Cold Startup Test Run.	V-23
V-8	Side Elevation View - Experimental Receiver	V-24
V-9	General Arrangement of the Experimental Receiver - Superheater Elevation Sections and Details	V-25
V-10	General Arrangements of the Experimental Receiver - Sectional and Plan Views	V-26

V-11	General View of Experimental Receiver During Erection . . .	V-29
V-12	Positioning of Waterwall Panels During Erection.	V-30
V-13	Erection Schedule for the Experimental Receiver.	V-31
VI-1	Thermal Storage System Concepts.	VI-6
VI-2	Two-Stage Thermal Storage Subsystem Conceptual Schematic . .	VI-10
VI-3	Heat Balance for Two-Stage Thermal Storage Subsystem	VI-10
VI-4	Detailed Schematic for Two-Stage Pilot Plant Thermal Storage Subsystem	VI-13
VI-5	Preliminary Ground Layout for Two-Stage Pilot Plant Thermal Storage Subsystem	VI-14
VI-6	Predicted Steam and Water Outlet Temperatures for Pilot Plant Thermal Storage Subsystem during Charging	VI-19
VI-7	Predicted Steam and Water Outlet Temperatures for Research Experiment Thermal Storage Subsystem during Charging	VI-19
VI-8	Detailed Schematic of Thermal Storage Subsystem Research Experiment	VI-21
VI-9	Perspective View of Thermal Storage Subsystem Research Experiment, without Supporting Structures	VI-23
VI-10	View of Thermal Storage Subsystem Research Experiment under Construction	VI-23
VI-11	Ground Layout of Thermal Storage Subsystem Research Experiment	VI-24
VI-12	View of Pipe Trestle and Heat Exchangers in Thermal Storage Subsystem Research Experiment	VI-24
VI-13	Desuperheating-Superheating Heat Exchanger HE1 Drawing . . .	VI-25
VI-14	Desuperheating-Superheating Heat Exchanger HE1	VI-25
VI-15	Condensing-Evaporating Heat Exchanger HE2.	VI-26
VI-16	Condensing-Evaporating Heat Exchanger HE2 Drawing	VI-26
VI-17	Subcooling-Preheating Heat Exchanger HE3 Drawing	VI-27
VI-18	Subcooling-Preheating Heat Exchanger HE3	VI-27
VI-19	Drawing for Molten Salt Storage Tank	VI-30
VI-20	Molten Salt Storage Tanks and Heat Exchanger HE1	VI-30
VI-21	Hydrocarbon Oil Storage Tank	VI-31
VI-22	Drawing of Oil Storage Tank	VI-31
VI-23	Plan View of Oil Piping System	VI-33
VI-24	Desuperheating Drawing	VI-33
VI-25	Drawing of Research Experiment Control Panel	VI-34
VI-26	Research Experiment Control Panel	VI-34
VI-27	Total Oil Decomposition Rate as a Function of Temperature .	VI-39
VI-28	Thermal Storage Subsystem Master Schedule	VI-4
VII-1	Base Heat Balance Diagram	VII-5
VII-2	EPGS Mollier Diagram, 12.5 MWe Gross	VII-5
VII-3	Dry Cooling System Design Features	VII-7
VII-4	Typical Commercial Turbine Cycle Life Expenditure	VII-10

VII-5	Recommended Temperature Ramp vs. Temperature Change for GE 12.5 MWe Admission Turbine	VII-10
VII-6	EPGS Plant Layout	VII-12
VII-7	Plot Plan	VII-15
VII-8	General Plant Arrangement	VII-16
VII-9	Mat Foundation for Heliostat and Sensor	VII-19
VII-10	Receiver Tower and Major Piping	VII-20
VII-11	ERDA 10 MWe Design Point Heat Balance for Receiver Steam Operation	VII-29
VII-12	ERDA 7 MWe Design Point Heat Balance for Admission (Storage) Steam Operation	VII-30
VII-13	Tower Top Receiver Support Concepts	VII-41

INTRODUCTION

This report summarizes the program activities since the beginning of the program in July of 1975 through September of 1976.

The primary efforts during the first portion of this period were the preparation of the Pilot Plant Preliminary Design Baseline, and the Conceptual Design of the three subsystem research experiments. The Preliminary Design Baseline for the Pilot Plant was developed from the commercial plant conceptual design that had been defined prior to the start of this program. It definitized each of the basic Pilot Plant Subsystems and provided the basis for development of the conceptual designs of the research experiments.

For each of the solar peculiar subsystems, the Collector Subsystem, the Receiver Subsystem and the Thermal Storage Subsystem, a subsystem research experiment was planned. The Conceptual Design of each of these experiments was prepared and reviewed with ERDA and Sandia in order to obtain authorization for the design, build and test of these experiments.

Subsequent to completion of the Pilot Plant Preliminary Design Baseline and the Research Experiments Conceptual Design the experiments have been designed and are at this time in the final phase of being built and/or checked out.

In the Collector Subsystem experiment the design was completed, the four experimental heliostats have been fabricated and erected. Initial calorimeter and radiometer data have been obtained and have demonstrated good correlation with the projected performance.

The 5 megawatt thermal experiment receiver has been fabricated by Foster Wheeler Energy Corporation and is now completing erection at the Radiant Heat Test Facility at Sandia in Albuquerque, New Mexico. This testing is scheduled for the period from December 1976 through February 1977.

The Thermal Storage Research Experiment fabrication and erection is nearing completion by Georgia Institute of Technology. Checkout of this experiment is scheduled for the first of December 1976 with the test program to be complete by the end of January 1977.

In addition to the efforts on the three research experiments, Bechtel Corporation has conducted an optimization study on the Electrical Power Generation Subsystem. They have also provided all of the A&E assistance on the program and have accomplished such efforts as design of the receiver tower and the heliostat foundations.

The major remaining activities on the program are to complete the three research experiments, prepare the final Pilot Plant Preliminary Design, and develop cost estimates for both the Pilot Plant and a Commercial Solar Power Plant.

II. 10 MWe PILOT PLANT

A. INTRODUCTION

The 10 MWe Pilot Plant is one module of a full scale commercial plant, for the subsystems that are solar peculiar. The modular concept for the commercial plant provides higher performance, better economics, maximum flexibility and minimum scaling concerns. This concept extends from the full size commercial receiver and tower that will be used in the pilot plant, to the thermal storage subsystem and to heliostats which are presently full size in the research experiments. The turbine-generator and related process equipment for the commercial plant will not be modules of the pilot plant but will be of commercial size, with its related higher efficiency, rather than the industrial size used in the pilot plant.

The selected design and configuration of the plant provides maximum performance and economics consistent with the cyclic requirements of a solar plant. The location of the collector subsystem heliostats in a north field configuration provides maximum collection of energy throughout the year. The enclosed cavity receiver tilted down facing the field captures the total energy from the collector subsystem and provides the minimum loss of energy due to convection, reflection or radiation. The selection of a thermal storage subsystem which couples hydrocarbon oil with molten salt provides a working fluid that maximizes the thermodynamic cycle efficiency.

Proven technology that provides low risk, simplicity of operation and high reliability has been utilized throughout to insure success in both the commercial plant and the pilot plant. These concepts include the tracking heliostats, a conventional boiler with a steam drum and related three element feedwater controller and attenuators and a sensible heat thermal storage subsystem with conventional heat exchangers and heat transfer medias that have been used extensively in the process industry.

All subsystem research experiments demonstrate the plant designs by providing what are essentially mini-commercial/pilot plant subsystems. Not only are the research experiments for the receiver and thermal storage scale models, they also generate the same state points and utilize the same controls as required for the commercial and pilot plants. The research experiment for the collector subsystem, as stated above, uses a full size commercial/pilot plant heliostat.

The plant performance optimizations and analysis that are performed for the pilot plant are almost entirely applicable to the commercial plant due to the modular concept. In performing intersubsystem optimization it has to be realized that solar plants have some different drivers than conventional plants. An example is the cost leverage that is provided by the collector subsystem with respect to the turbine plant. This permits additional expenditures in the turbine plant to increase efficiencies in order to provide a lower cost for the total plant. It is important to note that even though the pilot plant design point is specified as 10 MW electrical bus bar power at 2:00 PM on winter solstice, the plant will provide 10 MWe net throughout the year. All plant performance improvements that are gained at the specified design point through the system will of course be reflected in reduced heliostats required in the collector subsystem. Therefore, it is to be expected that the number of heliostats will be reduced as optimizations continue.

B. PILOT PLANT DESCRIPTION

1. Physical Description

An artist's conception of the Pilot Plant, which depicts the major subsystems of the plant, is shown in Figure II-1. The collector subsystem provides a field of heliostats that focus the insolation into the cavity of a receiver.

The receiver subsystem is located atop a tower at the south edge of the heliostat field to provide maximum collection of insolation on an annual basis. Superheated steam generated in the receiver subsystem is provided down the tower to either the electrical power generation subsystem, the thermal storage subsystem, or both. The electrical power generation subsystem accepts the steam in a dual admission turbine, and electrical power is provided to the electrical power transmission network or to a test load, as desired. Heat rejection is provided by an air cooled condenser which was selected in anticipation of a water availability problem for potential sites for solar thermal power plants. The thermal storage subsystem accepts steam from the receiver subsystem and removes the energy in heat exchangers for storage. As required, the stored energy is provided to the dual admission turbine in the electrical power generation subsystem. The electrical power generation subsystem conditions feedwater for return to both the receiver subsystem and the thermal storage subsystem.

Figure II-1
Artist's Concept, 10 MWe Central Receiver Solar Thermal Pilot Power Plant

The Pilot Plant is designed to produce 10 MWe net electrical power when operating from the receiver subsystem at the design point (2:00 PM, winter solstice, Inyokern, California) and can produce 10 MWe net throughout the year. In addition it is capable of producing 7 MWe net electrical power for 6 hours when operating from the thermal storage subsystem.

a. Collector Subsystem - The collector subsystem consists of heliostats with warped mirrors. The mirrors are second-surface silvered and are laminated using float glass. Each heliostat provides 37.16 m^2 (400 ft^2) of reflective surface. This configuration was selected for the relationship between cost, size, and the technique for structural support. The control of each heliostat combines independent fine-pointing coupled with a processor computer for executive control.

b. Receiver Subsystem - The receiver subsystem is configured as an insulated cavity with side by side tubes for maximum efficiency. To minimize heat loss during periods of shutdown, an insulated door covers the aperture of the cavity and the working fluid flows to and from the receiver are sealed off, thus providing a minimum rate of cool down and time required to bring up the receiver to operating temperature. The boiler and superheater tubes are located within the cavity to take advantage of the flux patterns provided by the collector subsystem. The receiver tower located at the south edge of the heliostat field, coupled with a cavity aperture at a height of 137.2 m (450 ft) and tilted at 0.35 rad (20 deg) provides optimum cosines for the collector subsystem, and therefore, maximum utilization of insolation throughout the year.

c. Thermal Storage Subsystem - The thermal storage subsystem utilizes a sensible heat storage system. The sensible heat is stored in insulated, fluid storage tanks. Two different storage media are used depending upon the temperature required for storage. HITEC is used for storage above 589 K (600°F) and a hydrocarbon oil for storage below 589 K. Energy is removed from the working fluid using three heat exchangers--one for superheat, one for latent heat, and one for subcooling the water. Additionally, desuperheaters are used to insure the storage media do not exceed specified temperatures. The same heat exchangers are used to provide superheated steam for use by the electrical power generation subsystem. In addition to providing steam for the generation of electrical power, energy is provided for thermal management of components in the electrical power generation subsystem and the thermal storage subsystem. The thermal storage subsystem provides stored energy for the generation of electrical power for periods of zero insolation or in combination with the receiver subsystem for periods of low insolation.

Figure II-2 Pilot Plant System Schematic

d. Electrical Power Generation Subsystem - The turbine in the electrical power generation subsystem is a dual admission machine that accepts high pressure and temperature main steam from the receiver subsystem and lower pressure and temperature admission steam from the thermal storage subsystem at a separate admission point in the turbine. The selected turbine can produce in excess of the design point power requirements for both the main steam and the admission steam. The selected concept of a dual admission turbine is compatible with the cyclic operation of a solar power plant and is also scalable to larger power plants. Dry cooling has been used in anticipation of water availability problems in the southwest United States. Performance penalties associated with dry cooling can be improved by using wet cooling should water be available at a given site.

e. Master Control Subsystem - The master control subsystem provides the necessary display and controls for control of the pilot plant in its various modes of operation and for transition from one operating mode to another. In addition, it provides for collection of data and computations to evaluate pilot plant performance, including subsystems. The pilot plant will include more instrumentation and data recording than a conventional plant of comparable size since the objective of the Pilot Plant is to obtain data on plant operation and performance.

2. Pilot Plant System Schematic

The system schematic for the pilot plant is shown in Figure II-2. This schematic shows the subsystems and how they are interconnected to provide an operating pilot plant. The arrows on the schematic indicate the direction of flow of the working fluid, steam or water, as applicable. Where arrows are shown in both directions, with an associated "C" or "D", the mode of operation for the thermal storage subsystem is indicated, i.e., "C" for charge and "D" for discharge. In the charge mode, energy is being extracted from the steam and stored; and on discharge, steam is being generated for conversion into electrical power.

As can be seen from Figure II-2, the system provides for generation of electrical power from the receiver alone, from the thermal storage alone, or from both the receiver and the thermal storage operating together. In addition, the receiver can charge thermal storage alone, or can charge thermal storage while providing steam for the generation of electrical power.

a. Receiver Operation - The boiler and superheat sections shown in Figure II-2, line the cavity walls of the receiver and accept the solar energy. The steam drum receives feedwater from the EPGS and/or the TSS via the high pressure boiler feedwater pump. The feedwater that enters the steam drum is distributed to the boiler section by gravity feed. The steam is separated in the steam drum and sent to the superheat to provide 789 K (960°F), 9137 k Pa (1325 psig) steam. This superheated steam is delivered down the tower for generation of electrical power, for conversion and storage in the thermal storage subsystem, or both. As required, feedwater coming up the tower is diverted and sent to the attemperator for temperature control of the superheated steam. The receiver provides the necessary controls for regulation of feedwater flow, attemperator flow, and signals to the master control subsystem for control of the heliostats, as required for meeting operational limits and out of limit conditions. Continuous blowdown is provided to maintain the steam purity requirements for the superheater and the turbine. For periods when the receiver is not operating shutoff valves seal the receiver and insulated doors cover the cavity aperture to minimize the rate of cool-down.

b. Thermal Storage Subsystem Operation - The thermal storage subsystem, as shown in Figure II-2, consists of two stages, both of which are used for either extracting heat from the steam generated in the receiver (charge) or for generating steam from the feedwater returned from the electrical power generation subsystem (discharge). The energy extracted from the steam is stored as sensible heat in either Hitec, a molten salt, or a hydrocarbon oil, depending on temperature. Hitec is used for high temperature storage, and the hydrocarbon oil for the lower temperature storage. The energy is stored in insulated tanks for later use to generate superheated steam for the generation of electrical power. The same heat exchanger stages are used for both the extraction of energy from the steam or for generation of superheated steam from feedwater. Each stage serves an opposite function, depending on the mode of operation. During the charging mode, superheated steam is accepted from the receiver at 783 K (950°F) and 8791 k Pa (1275 psig). Most of the superheat is removed in the first stage and stored as sensible heat in the HITEC. In the second stage, the latent heat is removed in a condensing heat exchanger and stored as sensible heat in the hydrocarbon oil. The water exiting the condenser is slightly subcooled and is further subcooled in the last heat exchanger. During discharge the flow direction and process is reversed. Feedwater is preheated and boiled in the second stage using the heat stored in the hydrocarbon oil. Superheat is added to the steam in the first stage, using the heat stored in the HITEC, to produce 672 K (750°F) output steam at 4137 k Pa (600 psig).

Table II-1 Collector Field Effective Cosine for Inyokern

Month/Day	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00
Dec 21				0.782	0.903	0.948	0.965	0.971	0.960	0.939	0.862	0.754			
Jan 21				0.796	0.902	0.941	0.961	0.969	0.963	0.944	0.909	0.811	0.710		
Feb 21			0.599	0.805	0.893	0.930	0.953	0.962	0.956	0.936	0.902	0.832	0.659		
Mar 21			0.541	0.777	0.875	0.915	0.938	0.945	0.938	0.915	0.876	0.779	0.545		
Apr 21		0.321	0.580	0.770	0.854	0.892	0.913	0.918	0.908	0.883	0.837	0.732	0.492	0.331	
May 21	0.502	0.476	0.624	0.770	0.833	0.868	0.888	0.893	0.882	0.857	0.815	0.727	0.557	0.477	
Jun 21	0.519	0.514	0.630	0.762	0.818	0.854	0.875	0.881	0.871	0.847	0.809	0.741	0.598	0.510	0.517
Jul 21	0.510	0.484	0.602	0.754	0.823	0.860	0.882	0.890	0.882	0.858	0.820	0.748	0.593	0.484	0.511
Aug 21		0.352	0.565	0.760	0.846	0.886	0.908	0.914	0.905	0.882	0.838	0.742	0.524	0.357	
Sep 21		0.203	0.608	0.802	0.883	0.918	0.937	0.941	0.930	0.904	0.854	0.735	0.450		
Oct 21			0.712	0.856	0.911	0.941	0.958	0.959	0.947	0.920	0.876	0.760	0.497		
Nov 21			0.748	0.852	0.922	0.951	0.966	0.968	0.956	0.931	0.879	0.772			

Note: Tower height 137.2 m (450 ft); flat field of 1718 heliostats.

Table II-2 Collector Field Heliostat Shading for Inyokern

Month/Day	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00
Dec 21				0.108	0.016	0	0	0	0	0	0	0			
Jan 21				0.075	0.005	0	0	0	0	0	0.002	0.065	0.116		
Feb 21			0.223	0.041	0	0	0	0	0	0	0	0.024	0.161		
Mar 21			0.280	0.055	0.003	0	0	0	0	0	0.003	0.054	0.275		
Apr 21		0.521	0.212	0.042	0.003	0	0	0	0	0	0.008	0.069	0.311	0.482	
May 21	0.099	0.267	0.131	0.019	0.001	0	0	0	0	0	0.003	0.046	0.196	0.225	
Jun 21	0.051	0.189	0.106	0.011	0	0	0	0	0	0	0.001	0.022	0.133	0.172	
Jul 21	0.061	0.236	0.146	0.025	0.001	0	0	0	0	0	0.001	0.028	0.154	0.230	
Aug 21		0.465	0.223	0.045	0.003	0	0	0	0	0	0.006	0.058	0.268	0.445	
Sep 21		0.704	0.205	0.037	0.001	0	0	0	0	0	0.010	0.085	0.383		
Oct 21			0.112	0.012	0	0	0	0	0	0	0.003	0.073	0.335		
Nov 21			0.092	0.032	0	0	0	0	0	0	0.015	0.083			

Note: Tower height 137.2 m (450 ft) flat field.

Figure II-3 Tower Shadows for Winter Solstice

c. Electrical Power Generation Subsystem - The electrical power generation subsystem consists of conventional power plant components. As shown in Figure II-2, the turbine is an admission type machine which accepts 783 K (950°F), 8791 k Pa (1275 psig) steam from the receiver, or 672 K (750°F), 4137 k Pa (600 psig) admission steam from thermal storage. The subsystem contains four stages of feedwater heating using uncontrolled extraction from the turbine. The turbine accepts superheated steam from the receiver via the main steam throttle valve and/or lower temperature and pressure steam from thermal storage via the admission throttle valve for conversion into electrical power. Steam leaving the turbine is condensed in the dry cooled condenser. The resultant condensate is pumped through the full flow demineralizer and the feedwater heaters, increased in pressure, and returned to the receiver and/or thermal storage subsystems, depending on the mode of operation.

C. PERFORMANCE

1. General

The design point for the pilot plant, when generating electrical power from the receiver, is a clear day at 2:00 PM on winter solstice at Inyokern, California. Since the 1963 Inyokern data shows winter solstice to be a cloudy day, the days around the solstice were analyzed to determine a representative insolation at 2:00 PM.. The value selected, and used for pilot plant design, was 0.8255 kW/m². With a solar plant, the insolation is a function of time of day as well as time of year. It is, therefore, necessary to insure that the total system as well as each subsystem is designed to accommodate these variations. In addition, weather conditions are a significant influence on the insolation as well as having direct effects on subsystem operation. The combined effects of the variation in insolation time and weather therefore must be evaluated.

Performance factors for each portion of the plant have been isolated and evaluated. This provides the visibility necessary to control those variables that have a major influence on pilot plant performance. Factors such as effective field cosine, which includes field heliostat shading, have a significant effect on overall plant performance. The effective field cosine, as well as heliostat shading, is shown in Tables II-1 and II-2 for Inyokern. Another factor to be considered is the effect of the receiver tower shadow on collector field performance. Figure II-3 shows the shadow projection on the field for winter solstice. These factors, and many others, have been evaluated and analyzed to insure optimum system sizing.

Since the design point for the pilot plant is not the time of maximum insolation, additional consideration must be given to each subsystem performance requirement to insure economical utilization of the extra insolation available above the design point. Therefore, separate analyses were performed to establish the maximum performance required for each subsystem. The collector subsystem was sized to meet the design point requirements. Based on this configuration, the receiver was sized to accept the maximum typical insolation over the year for any day in the southwestern United States. Since the receiver subsystem can accept the maximum insolation and its output can be utilized either by the electrical power generation subsystem or the thermal storage subsystem, it is not necessarily desirable to design these subsystems based on the maximum insolation throughout the year. The thermal storage subsystem must first be designed to accept sufficient energy during any given day such that it can deliver the required 7 MWe of electrical power for six hours. However, since there may be a day when it is not desirable to operate the electrical power generation subsystem while charging the thermal storage, the thermal storage has been sized to accept the maximum insolation that may be expected on a typical day in Inyokern, Calif. The electrical power generation subsystem design is based on the maximum utilization of commercially available hardware. Therefore, the turbine size selected in fact can provide more electrical power than is required at the design point when operating from the receiver alone. The selected turbine can also produce power in excess of the design requirement when operating from thermal storage alone. The designed capability of the overall pilot plant can utilize the maximum capability of each subsystem within the pilot plant.

2. Design Point Sizing for the Pilot Plant

The pilot plant is designed to deliver 10 MWe bus bar electricity at 2:00 PM on winter solstice at Inyokern, California. The design point defines the subsystem performance requirements and size of the collector subsystem but not the size of the receiver subsystem, the thermal storage subsystem, or the electrical power generation subsystem. The electrical power generation subsystem size must meet the performance requirement at the design point, but since existing hardware is being used to the maximum extent possible, the selected turbine has a capability of 12.5 MWe gross which is in excess of the design point requirements.

Figure II-4 shows an updated version of the system energy balance chart presented earlier in the program. The original chart started with the required bus bar electrical output of 10 MWe at the bottom

of the energy balance chart, 10 percent of the net electrical output was allowed for auxiliary power therefore requiring 11 MWe gross from the turbine/generator. The gross heat rate at this power output was 11,284 BTU/kWh, therefore requiring 36.369 MWth in steam provided to the turbine. Progressing up the energy balance chart to the top, the potential power required to the system was 57.702 MWth. The definition of potential power to the system is that amount of solar energy that would impinge on the mirror surfaces if they were oriented perpendicular to the sun's rays. Knowing this number and the solar insolation value, the required mirror area was determined. As discussed earlier, a solar insolation value of 0.8255 KW/m^2 was used. Based on the design size of the heliostat, the collector field required 1718 heliostats to support the pilot plant.

For comparative and analytical purposes the number of mirrors, and thus the potential power to the system, has been allowed to remain constant even though some gains have been made in plant efficiency. As can be seen in Figure II-4 the net electrical output has increased from 10.0 MWe to 10.29 MWe due to a gain in the turbine heat rate. When the plant optimization is complete the field size will be reduced based on the realized efficiency gains from all subsystems.

3. Performance Calculations

Determination of the pilot plant performance for a given set of conditions requires a detailed analysis of the performance factors involved. Some of these factors were mentioned in the previous paragraphs and a simplified summary of plant performance for design point conditions was shown in Figure II-4. Plant performance must be calculated as a function of many variables including insolation, time of day and time of year (sun position), ambient temperature, wind speed, plant location, etc. Since plant performance under various weather and insolation conditions as well as various times of the year and plant locations is important in evaluating the overall effectiveness of a solar plant, a computer program is being developed that will simulate pilot plant performance. The program is a quasi-steady state model of the pilot plant which will be capable of calculating plant performance at a single point or simulating daily, weekly, and even yearly performance.

Although the program is still under development, it has been used in several plant performance analyses. One of these was a study of the effect of ambient temperature on plant power output. Plant performance was calculated at design point conditions for ambient temperatures ranging from 272 K (30°F) to 317 K (110°F). The

Figure II-4 Pilot Plant Performance - Design Point

results of this study are shown in Figure II-5. The upper curve is the gross output of the turbine and the lower curve the net plant output. The discontinuity in the gross output is due to a change in condenser fan speed from full speed to half speed. For a plant utilizing dry cooling, the results indicate that ambient temperature is a significant driver in plant performance and that the temperature selected for the plant design point can significantly affect the size of the collector field. (Reduction of specified design point temperatures from 85°F to 57°F results in a field size reduction of approximately 55 heliostats).

The computer model was also used to evaluate plant performance at noon on winter solstice, spring and fall equinoxes as well as at the design point. The values used for ambient temperature and insolation were taken from the Aerospace tapes for Inyokern. The temperatures used for winter, spring, summer and fall were 287 K (57°F), 292 K (65.5°F), 301 K (82.5°F) and 298 K (77°F) respectively. The insolation values used are shown in Figures II-6 through II-9. It should be noted that except for fall equinox the noon insolation values used were not the peak for the day. For reference purposes the curves also show the insolation recorded at Denver which is representative of a clear day at the solstices and equinoxes. The plant electrical output was calculated using the Inyokern temperature and insolation values (and at the design point temperature 303 K (85°F) for reference). The results are summarized in Table II-3. This data shows consistent performance throughout the year and that equinox performance is slightly better than summer solstice and design point performance.

4. Yearly Performance

Determination of the annual energy performance for the collector subsystem and the solar power plant electrical output is shown in Figure II-10 and was based on the insolation data for Inyokern (Aerospace Tape) and the collector field performance pattern generated by a sequence of runs for the 21st of each month. Two performance parameters are necessary for the evaluation of varied configurations, the unit thermal energy from the collector, and the unit electrical energy from the power plant. The unit thermal energy curve shown is applicable to both the pilot plant and the commercial plant, since the commercial collector system is comprised of modules identical to the pilot plant collector system. Over the year, this integrates to 2567 KW-hrs/m² of mirror. The electrical energy curve shown was determined for the energy balances established for each month for the 10 MWe pilot plant and includes the variable effect of tower shadow on the collector as well as the seasonally variable geometry. As a reference point,

Figure II-5 Pilot Plant Electrical Output vs Temperature

Fig. II-6 Spring Equinox - Midday Insolation

Fig. II-7 Summer Solstice - Midday Insolation

Fig. 11-8 Fall Equinox - Midday Insolation

Fig. 11-9 Winter Solstice - Midday Insolation

Table II-3 Pilot Plant Performance: Solstice & Equinox

	Winter Solstice 2:00 pm		Spring Equinox Noon		Summer Solstice Noon		Fall Equinox Noon	
Direct Insolation, kW/m^2	0.8255		0.9150		0.9100		0.8750	
Ambient Temperature, $^{\circ}\text{K}$ ($^{\circ}\text{F}$)	302.6 (85)	287.0 (57.0)	302.6 (85)	291.8 (65.5)	302.6 (85)	301.2 (82.5)	302.6 (85)	298.2 (77.0)
Steam Flow Rate, kg/hr (lb/hr)	51963 (114556)		59003 (130077)*		55227 (121752)		56404 (124347)	
Auxiliary Load, MW (e)	1.05	0.57	1.09	0.61	1.07	1.07	1.07	1.08
Gross Electrical Output, MW (e)	11.34	11.19	12.75	12.40	11.99	12.01	12.23	12.28
Turbine Backpressure, kPa (in. Hg)	12.9 (3.8)	15.8 (4.7)	13.6 (4.0)	23.3 (6.9)	13.2 (3.9)	12.7 (3.7)	13.3 (3.9)	11.6 (3.4)
Backpressure Correction, MW(e)	+0.37	+0.22	+0.24	-0.12	+0.30	+0.32	+0.28	+0.33
Condenser Duty, MW(t)	24.64	24.79	27.75	28.10	26.08	26.06	26.59	26.54
Fan Speed	Full	Half	Full	Half	Full	Half	Full	Full
Feedwater Temperature, $^{\circ}\text{K}$ ($^{\circ}\text{F}$)	482.2 (408.3)		487.5 (417.8)		484.9 (413.1)		485.8 (414.7)	
Net Electrical Output, MW(e)	10.29	10.62	11.66	11.79	10.93	10.94	11.16	11.20

*Maximum Allowable Flow Rate, Actual Flow 130723

Figure JI-10 Energy Collection Performance for 1 m² Mirror 137.2 m (450 ft) Tower, Flatfield Configuration

the overall efficiency (electrical output/solar input) for daylight operation averaged 0.229.

For the commercial plant, daylight generation efficiency is 0.267, and storage operation efficiency is 0.227, resulting in a weighted efficiency for the plant of 0.250. To obtain the unit electrical energy curve for this 100 MWe plant with storage, the curve shown would be raised by a factor of 1.089. The output integral value would raise from 587 KW-hrs/m² to 639 KW-hrs/m².

Yearly performance of the pilot plant was calculated and is shown in Figure II-11. The calculations used the sequenced outputs of the University of Houston Solar Flux Program and the MMC Program which integrates annual energy based on collector field performance and insolation data from the Aerospace tapes. As shown in the figure the receiver aperture height effect on performance is a reduction in yield with reduced aperture height with the rate of reduction increasing as receiver aperture height decreases.

D. PILOT PLANT OPERATION

1. Operational Modes

The various operational modes of the plant as well as the interaction between subsystems can be best understood by describing a typical day of operation. It is obvious, depending on user requirements, that many combinations of operating modes are possible and may be desirable for a given day. Therefore, a set of operating conditions has been established which is representative of one approach that may be used. For this purpose, it is assumed that the thermal storage subsystem has been depleted during the previous night's operation. This assumption is made since it is believed that typically the user of the pilot plant electrical power will want to utilize all the energy available on a given day. In addition, it is believed that the user will want some assurance that the system is insensitive to minor cloud transients during the day before the system is allowed to go on line. Therefore, the first item to be accomplished on this typical day is to provide for approximately one hour's worth of operation during the day on the storage alone, if required. Starting with this premise, a typical day insolation curve was developed for purposes of establishing the system performance characteristics. These data were plotted in Figure II-12 for December 24, 1963, at Invokern as a representative day for insolation near winter solstice. Without further

Figure II-11 Combined Effects of Tower Height and Collector Field on Annual Energy Delivered by Solar Collector Field

Figure II-12 Typical Day Insolation

Figure II-13 Typical Day Performance Profile

meteorological data, it is not known why the curve dips, followed by an abrupt drop in the afternoon. However, it is characteristic of much of the Inyokern data and could be a problem with the way that the original data was recorded or recalculated for direct insolation. In addition to the insolation, the sun elevation angle for that day is plotted.

Based on this insolation data and calculations for the performance of the various subsystems the typical day performance profile shown in Figure II-13 was developed. Each mode of operation, which is keyed at the bottom of the page, refers to the operating modes shown in the inset. Based on the receiver flowrates generated by the insolation on this typical day and the selected mode of operation, the flowrates for the turbine and thermal storage were developed. In addition, the electrical power generated for transmission is shown. It is apparent from a review of this curve that many combinations of operating modes can be selected to provide the desired electrical output or thermal storage, while utilizing all of the steam generated in the receiver.

The first mode of operation shown on Figure II-13 is Receiver Start-up which brings the receiver up from overnight temperature and pressure to rated conditions. When rated conditions are attained steam is diverted to thermal storage to begin the Charge Thermal Storage mode. When thermal storage has been charged sufficiently to provide one hour's worth of discharge the Turbine Start-up To Synchronization mode is entered. During this mode steam is admitted to the turbine in the partial arc mode bringing it gradually from turning gear to rated speed, at which time the generator is synchronized with and tied to the electrical power network. Once the generator is synchronized the unit load is increased, within allowable rates, until it matches the flowrate capability of the receiver. At this point, when the turbine is accepting all steam available from the receiver, the system is in the Generating From Receiver Alone mode. At the point when the receiver output exceeds the maximum capability of the turbine the system enters the Generating From Receiver And Charging Storage mode. When the decreasing insolation in the afternoon causes the receiver output to decrease below the maximum turbine capability the system reenters the Generating From Receiver Alone mode. About 3:00 PM on the typical day shown the TSS is placed in the discharge mode to supplement the receiver output. This is the Generating From Receiver And Storage mode. When the receiver output goes to zero at sunset the Generation From Storage Alone mode is entered. In this mode the TSS supplies the steam flow required to maintain the net electrical output at 7 MWe. When

the TSS is nearly depleted the Turbine Planned Load To Trip mode is entered. In this mode the load is gradually decreased, according to a planned load-time profile, until the turbine trips off line and coasts to turning gear ending the day's operation. The Shutdown And Overnight mode maintains the system in readiness for startup the next morning.

2. Pilot Plant Start-up Analysis

Analysis have been performed to evaluate pilot plant performance when starting from a hot standby condition, a cold condition and when starting from thermal storage. The results of these analyses are summarized in this section.

a. Plant Start-up From Hot Standby - This type of a start-up, considered to be a normal morning start, was analyzed to determine the maximum time required to bring the plant to "full power", where full power is defined as the point at which all available insolation is used in the generation of electrical power. Insolation data taken at Martin Marietta, together with field cosines, show that the worse case startup conditions (fastest rate of increase of power to receiver) occur at winter solstice. This power input profile was calculated and is shown, together with the morning insolation, in Figure II-14.

Figure II-14 Morning Startup Power to Receiver

The resultant receiver start-up profile was calculated and is shown in Figure II-15. Finally, using the receiver start-up profile, the turbine start-up profile was calculated and is shown in Figure II-16. This analysis showed that, for a normal morning start-up, the receiver can accept all the available solar energy and the plant can reach "full power" in approximately 45 minutes.

b. Plant Cold Start - The pilot plant cold start-up is a start after the receiver and turbine have cooled to ambient temperature. This analysis was also performed to determine the plant start time but differs from the hot start analysis in that the receiver is not attempting to follow an anticipated power input profile. In this case the receiver is dictating the power input profile which it can accept. Figure II-17 summarizes the cold start-up and shows the turbine start-up profile as well as the receiver as a function of time after start. Although the analysis shows a start-up time of approximately 7.8 hours, it should be noted that this does not necessarily represent the fastest receiver start-up time.

c. Plant Start-up From Thermal Storage - This analysis was conducted to determine the time required to bring the plant to design point operating conditions for both a hot (0 to 12 hours) and warm (12 to 72 hours) start, and to identify the energy required for each. Start-up time was defined as beginning when the turbine rolls off turning gear and ending when the turbine-generator gross output reaches 7.7 MW_e . The resultant start-up curves, as well as a summary of the results, are shown in Figure II-18. The hot start from TSS required 37.5 minutes and required 7.24 MW-HR_{th} and the warm start required 64 minutes and required 14.04 MW-HR_{th} .

3. Pilot Plant Stand-by And Emergency Power Operations

In the event of loss of the normal source of power to the auxiliary electrical equipment in the power plant, harm to personnel and/or damage to equipment could occur unless backup power sources are provided. Two sources of backup power are provided to preclude this danger. A standby diesel generator is provided to prevent either harm to personnel or damage to equipment if power is lost. In addition, an emergency battery is provided to preclude harm to personnel during the startup period of the diesel generator or if there is a malfunction in the diesel generator. During normal operation, power is provided to the electrical equipment by the power plant itself or the utility grid, depending on the mode of operation. This power is drawn from the utility grid through the main transformer and circuit breaker as shown in Figure II-19. The 13.8 kV switchgear then distributes the power to the two load center units where it is stepped down to 480V and distributed to the required auxiliary loads.

Figure II-15 Morning Startup, Receiver

Figure II-16 Morning Startup, EPGS

Figure II-17 Pilot Plant Cold Startup

Figure II-18 Turbine Start-Up From Admission Steam to Design Point

Figure II-19 Electrical System Block Diagram

During morning startup, the main generator is brought up to rated voltage and speed. After it is synchronized with the 13.8 kV main bus, the generator circuit breaker is closed, thereby paralleling the unit with the utility power grid. The generator is then controlled to supply the pilot plant auxiliary loads and transmit net power to the utility grid.

In the event of loss of the utility grid power, the turbine-generator will remain in operation, supplying the pilot plant auxiliary loads until it is manually shutdown or down to a minimum of .12 MW, at which time the turbine will trip off line. If the 13.8 kV bus is lost due to interruption of both the utility grid and main generator, the 1600A load center circuit breakers and nonessential loads circuit breakers trip and automatic starting of the diesel generator will occur.

Any number of events could occur leading up to the starting of the diesel generator. Usage of the diesel generator is more likely to occur during plant startup, shutdown, or overnight operations, since loss of the utility grid will cause loss of the 13.8 kV switchgear bus. During daytime operations when the main generator is operating, loss of both the utility grid and main generator would be required to lose the 13.8 kV switchgear bus. Several possible causes for loss of the utility grid are storm, earthquake, fire, etc. The output of the generator may be lost for various reasons. Either a turbine trip or generator trip will shutdown the units and can be brought about by a number of reasons of mechanical or electrical irregularities.

Immediately upon loss of the 13.8 kV switchgear bus (loss of utility grid tie and main generator output), sensing circuitry detects the power loss; and within 0.5 seconds, signals operate the compressed air line solenoid to start the standby diesel generator. During the diesel generator startup period of about 10 seconds, the emergency battery will supply all loads on the 125 Vdc bus. As the output of the battery charger falls due to the loss of power, the battery immediately takes over and continues an uninterrupted supply of power to the emergency DC loads. Additionally, power to the DC/AC inverter serving the plant computer and public address system will be supplied by the battery. After the 10 seconds and the diesel generator has come up to speed, it will furnish the emergency loads previously supplied by the battery and will also start to recharge the battery. Should the diesel generator fail to start, or should its operation be interrupted, the emergency loads will be supplied by the battery for about two hours, after which time the battery will be completely discharged.

Once the diesel generator has started and has been brought on line, it will supply loads up to 375 kW for up to 48 hours. If it is determined that a longer run time will be required, additional fuel will need to be trucked in. Upon resumption of the utility grid power tie, it will be necessary to manually shutdown the diesel generator and reset the 1600 A load center circuit breakers, thereby restoring normally supplied power.

After the standby diesel generator has come up to speed and assumes its standby loads, plant safing actions will begin. If this occurs during normal daytime operating conditions, the turbine-generator, condenser, and feedwater pumps will stop. Using the residual water in the boiler drum, steam will continue to be generated until the heliostats are removed from the receiver cavity. This steam will be vented to the atmosphere since the condenser is not operating. Therefore, the initial sequence to be performed on standby power includes defocusing the heliostats off the receiver aperture and stowing, sealing off the thermal storage subsystem, closing the receiver doors, and sealing off the receiver subsystem.

E. SYSTEM EFFECTIVENESS ANALYSIS

System effectiveness for the pilot plant encompasses the two major areas of activity of availability allocations and failure modes and effects analysis. The availability model use is the inherent availability model which assumes that the system is capable of producing 100% load at the generator output, excluding any outage due to poor weather conditions. The failure modes and effects analysis identifies the equipment failure modes and the consequences of the failure modes along with the fail safe provisions and the failure warning provisions required.

F. 10 MW_e PILOT PLANT COST ANALYSIS

A report was prepared and delivered that provided a cost analysis of the Central Receiver Solar Thermal Power System 10 MW_e Pilot Plant. The design baseline for the plant is that presented in the Preliminary Design Baseline Report and Conceptual Design Report, Volumes I, II, and III, dated December 1975. The cost estimates were provided by a team lead by Martin Marietta Aerospace of Denver, Colorado, who is the integrator for the overall effort and the Collector Subsystem designer. Bechtel Corporation, San Francisco, California, is responsible for the Electric Power Generation Subsystem and the architect-engineer task of the Pilot Plant. Foster Wheeler Energy Corporation, Livingston, New Jersey, is responsible

for the Receiver Subsystem and Georgia Institute of Technology, Atlanta, Georgia, is responsible for the Thermal Storage Subsystem.

The format for presenting the cost estimates were provided by Sandia Laboratories, Livermore, California. For estimating purposes, the site location was assumed to be between Phoenix and Yuma, Arizona, and adjacent to the main highway system. The estimate is based on 1975 mid-year average dollars. A three-year program was assumed from go-ahead to plant operation.

III. COLLECTOR SUBSYSTEM

The contents of this section highlight two periods of activity. The first period, which covers from July 1975 through June 1976, includes the preliminary design of the 10 MWe pilot plant collector subsystem and the design, development, and fabrication of the Subsystem Research Experiment (SRE) heliostats and associated test facilities. The results of this first year have been described in detail in previous reports, therefore, only a summary of these activities is presented in this report.

The second period, extending from July through September 1976, includes the initial test results from the first SRE heliostat, results of component testing, and the checkout and calibration of the calorimeter/radiometer test facility. Since this test data has not been previously described, it will be addressed in considerable detail here.

A. SUMMARY OF COLLECTOR SUBSYSTEM ACTIVITIES, JULY 1975 THROUGH JUNE 1976

1. Preliminary Pilot Plant Definition

The major effort during this phase was directed toward the preliminary sizing and layout of the collector field for a 10 MWe pilot plant. First, the performance requirements for the Collector Subsystem were established based upon the 2 PM Winter Solstice design point. The purpose of these requirements was to establish interfaces with other subsystems and to provide an up-to-date basis for analyses, trade studies, and component designs.

An error budget was established for the Collector Subsystem which identified all known factors contributing to power loss. Values, based upon preliminary studies, tests and estimates, were assigned to each factor, as shown in Table III-1. Although conceptual at this point, these error allocations, in conjunction with pertinent environmental requirements, defined the design constraints under which the preliminary design of the 10 MWe pilot plant and the detailed design of the SRE experiment would proceed.

Figure III-1 shows the conceptual collector field layout for a 10 MWe pilot plant. The field contains a total of 1718 heliostats symmetrically arranged around a north-south centerline. The reflected sunlight from the heliostats is concentrated in the cavity receiver due south of the field.

Table III-1 Error Budget for Collector Field and Heliostats

Error of Loss Source	Tracking Error, arc-min	Efficiency Factor
Heliostat Availability	-	0.994
Tower Shading	-	0.974
Cosine Effect	-	0.939
Reflectivity	-	0.85
Pointing and Tracking		
Pointing	0.5	
Sensor Alignment	0.5	
Hysteresis	1.0	
Circuit Calibration	0.5	
Drive Mechanism Backlash	-	
Sensor Wind Loading	0.5	
Structural Deformation:		
Wind Velocity = 30 mph	2.5	
Wind Velocity = 5 mph	0.5	
Foundation * Pedestal Deflection		
Total Tracking:		
5 to 30 mph	5.5	0.982
0 to 5 mph	3.5	0.988
Optical Inaccuracies		
Aberration	-	
Tangential and Focusing Error	3.7	0.988
Total Collection Efficiency		
5 to 30 mph		0.750
0 to 5 mph		0.763

3-III

Figure III-1 Conceptual Collector Field Layout for 10 MWe Pilot Plant

The collector field was sized and the heliostats arranged to meet the criteria of: 1) providing 39.8 MW_{th} to the receiver cavity at 2:00 PM local sun time on winter solstice under an insolation of 0.8255 KW/m², 2) eliminating shading or blocking for a minimum sun angle of 30° above the horizon, and 3) utilizing as high a packing factor as possible.

The total area of the collector field is approximately 310.799 m² (0.12 square miles), exclusive of boundary roads, fences, support building, etc. The number of heliostats is 1718, giving a total reflecting area of 63843 m² (687,200 ft²). The shortest focal distance in the field is 151 m (497 feet) while the longest focal distance is 693 m (2275 feet).

The pilot plant field described above is intended to be a module in a commercial solar plant of 100 MWe capacity or greater. These modules would be grouped as dictated by the terrain and the power requirements of the plant. Once the pilot plant design is proved, the module could be readily used without additional scaling.

The control system currently envisioned for the pilot plant is a hybrid, consisting of a mini-computer operating in conjunction with individual heliostat control. In this scheme, the normal operating mode, i.e., where the reflected beams are concentrated in the receiver cavity, the pointing of the heliostats is controlled by a closed loop system. This system employs a dual axis sensor mounted on a centerline between the receiver aperture and the center of the heliostat. All other functions are handled with an executive control computer which addresses the heliostats either singly or in groups.

Each heliostat will achieve the following operating modes:

1. Fine-tracking - Tracking to within pointing requirements by automatic tracking system;
2. Synthetic tracking - Used to match the approximate rotational speed of the sun during cloudy periods when the automatic tracking system is inoperable, or to acquire the sun;
3. Synthetic tracking plus elevation or azimuth offset - Maintains approximate solar tracking in both axes, with a computer-commanded offset for emergency modes. Necessary for fast recovery of fine-tracking mode;
4. Fast slew - A high-speed slew to allow positioning of the mirror by the master control system. The collector shall be capable of slewing 3.14 rad (180 deg) in any axis of rotation within five minutes;

5. Stow - A face-down position to protect against wind or hail and for minimizing surface storage, as shown below, and achieved in fast slew mode;
6. Limit and fail-safe - Rotational limits of individual heliostats as restrained by electromechanical devices for safety reasons;
7. Automatic fail-safe - Master control of subsystem or individual heliostats in response to an alarm system connected to receiver/thermal storage subsystem. Heliostats go to fine-tracking plus elevation offset or stow position;
8. Calibration - To allow focusing and alignment of mirrors;
9. Manual control - For maintenance.

2. Subsystem Research Experiment (SRE)

The purpose of the Collector SRE is to support the preliminary design of the 10 MWe Pilot Plant. This purpose is achieved several ways. First, the basic solar thermal process is to be verified for ranges up to those representative of the 10 MWe module. Secondly, the SRE provides a vehicle for testing various components and assemblies in order to substantiate a heliostat design that can be commercially produced. Thirdly, the SRE results are to be used to either verify or modify the computer models used in the preliminary pilot plant and commercial plant designs.

During the SRE, a total of four heliostats are to be fabricated and tested to achieve the objectives stated above. The development process consists of four phases: 1) conceptual design, 2) detail design, 3) fabrication, and 4) test, as described below.

a. Conceptual Design Phase - July through October 1975 - The conceptual design phase consisted of a series of tests and analyses designed to establish the basic concepts to be investigated in the SRE. The most significant of these tests and analyses are described below.

- 1) Wind Tunnel Test - In order to adequately evaluate the effects of winds on the heliostat, models of several heliostat configurations were tested in a wind tunnel. The models were tested at different angles of elevation and rotated to different azimuth angles during the test. The tests also provided data on the loading caused by the mirror support structure since this was also modeled. Tests were made in

winds up to an equivalent of 22.35 m/s (50 mph). Pointing errors for each mirror as well as azimuth and elevation torque requirement were determined. From the data, a surface area of approximately 39.16 m² (400 ft²) was determined to be optimum.

- 2) Drive Mechanism Tradeoff Study - Design tradeoff studies were conducted to determine the most economical drive mechanism that would maintain the angular accuracy, including minimum backlash, would position the heliostat over the desired angle, and would maintain its requirements through 30 gears of continuous operation. A number of gear drives were evaluated as well as cable drives, chain drives, hydraulic drives and combinations involving braking and damping.

The study resulted in the choice of a direct gear drive mechanism as the optimum design for the given loads and environmental conditions.

- 3) Structural Tradeoff Study - The structure of the heliostat has been determined to be the highest cost component. Therefore, various combinations of structural components were analyzed to achieve the most cost-effective design that would meet the requirements of overall strength and rigidity.
- 4) Control System Studies - A series of studies were conducted to determine the characteristics of the hybrid control system. The basic pointing accuracies, rates of travel, gimbal angle ranges and desirable control modes were established.

The primary purpose of the remainder of the analyses was to evaluate the stability of the fine-tracking control system for heliostats operating in different parts of the collector field.

The results showed that the basic system operated satisfactorily for the majority of heliostats in the field. However, unacceptable operation occurs for some heliostats throughout the year. The problem is worst at summer solstice; at winter solstice only a few heliostats are affected. The most severe problem occurs just after sunrise and just before sunset. It continues until 9 a.m. on summer solstice and spring and autumn equinox and starts again before 3 p.m. during that part of the year. More than 300 heliostats are affected on the worst day at the worst time of day (summer solstice, 6 a.m. and 6 p.m.).

The recommended solution is to crossfeed the fine tracking error signals--sensed elevation error fed to the azimuth gimbal and vice versa. Simulation results show that in some cases, 100% crossfeed provides completely satisfactory operation. However, during the transition from crossfeed to straight feed, it is necessary to feed only half of the sensed error signal from each channel to each gimbal, as described below.

Switching from straight feed to crossfeed will be controlled by the executive control computer on the basis of heliostat location in the field and gimbal angles. The specific switching criteria to be programmed in the computer will be defined during the next quarter. The additional components required for each heliostat are switches, inverters, and summing amplifiers. Inverters are required because the signs of the crossfeed terms are, in some cases, opposite to the signs of the straight feed terms.

Crossfeed capability is not required for all heliostats in the 10 MWe field. The decision on whether to incorporate the crossfeed circuitry in all of the heliostat electronic packages will be based on a comparison of the cost of the additional electronics versus the cost of having two different designs.

- 5) Heliostat Mirror Evaluation - Studies and evaluations were made of the glass and mirrors that could be obtained in quantity through commercial production methods from a number of glass companies. The main test that was made specifically to evaluate mirror samples for heliostat use was reflectivity. It was found that sheet glass could be obtained with low-iron content and that the best reflectivity could be obtained with the sheet glass. Later, it was found that the sheet glass had too irregular a surface to provide good focusing. Details of the measurements on focusing and flatness are discussed in Section B.

b. Detail Design Phase - October 1975 through March 1976 - The detail design phase consisted largely of the detail design of the individual components and assemblies which comprised a heliostat. These designs, later translated into fabrication drawings and specifications, were direct results of the concepts established previously. Three design reviews were held at periodic intervals to review the drawings.

The heliostat control electronics were designed, breadboarded and tested during this period.

Two heliostat designs resulted from this phase. The first design incorporated twenty-five 1.2 x 1.2 m variable-focus mirror as shown in Figures III-2 and III-3. Each mirror consists of two 3 mm glass sheets laminated together. The front sheet is a second-surface, silvered glass mirror. Two types of glass are used, i.e., low-iron sheet glass with a specular reflectivity of 91% and a standard float glass with a specular reflectivity of 83%.

The purpose of the 25-mirror design was to establish the performance of the high-reflectivity mirrors. The largest laminated sheet glass mirrors available at the time were 1.2 m x 1.2 m (48 in. x 48 in.). The variable focus feature provided the ability to move mirrors to different locations for testing.

An "advanced concept" heliostat is designed to prove several features that are considered desirable for commercial plants. As shown in Figures III-4 and III-5, this heliostat incorporates nine mirrors of larger area than the 25-mirror heliostat. The overall total reflective area is identical for both designs.

Each of the nine mirrors is mounted on a substrate and has a fixed focal length. Since only a single layer second surface mirror is necessary, larger sheets of float glass can be obtained. Each mirror is 2.0 x 2.0 m (80 in. x 80 in.) in size. The specular reflectivity of these mirrors, however, is only 73%, due to the glass thickness (6 mm) and high iron content.

The advantages of this design are readily apparent. First, the pre-focused mirror will minimize field labor. Second, the smaller number of mirrors requires approximately 36% less attach points than required for the 25-mirror configuration, resulting in less components. Lastly, less steel is required in the 9-mirror yoke, i.e., the span is less than 60% of the 25-mirror heliostat.

Table III-2 presents a detailed comparison of the two configurations.

c. Fabrication Phase - March through June 1976 - This phase included the procurement and fabrication of components and assemblies for four heliostats. Three heliostats are of the 25-mirror configuration while the fourth is a 9-mirror configuration.

The yoke and beam structures were fabricated by Rio Grande Fabricators of Albuquerque, New Mexico. Four sets of drive mechanisms were designed and built to MMC requirements by the Morse Chain Division of Borg-Warner Corporation.

Figure III-2 25-Mirror Heliostat, Front View

Figure III-3 25-Mirror Heliostat, Back View

Figure III-4 Nine-Mirror Heliostat, Front View

Figure III-5 Nine-Mirror Heliostat, Back View

Table III-2 Characteristics of SRE Heliostats

Characteristic	Experimental Type Heliostats	
	25-Mirror	9-Mirror
Overall Heliostat		
Weight	2331 kg (5139 lb)	2381 kg (5249 lb)
Number of Mirrors	25	9
Envelope Dimensions	6.9 m x 6.7 m (22.5 x 22 ft)	6.7 m x 6.9 m (22 x 22.75 ft)
Reflecting Area	37.16 m ² (400 ft ²)	37.16 m ² (400 ft ²)
Mirror Assembly & Supports		
Size	1.2 x 1.2 m (4 x 4 ft)	2.03 x 2.03 m (6.7 x 6.7 ft)
Type	Sheet and Float	Float
Focusing	Variable	Fixed
Substrate Material	Tubular Steel	Honeycomb
Glass Thickness (Total)	6.3 mm (0.25 in.) 3 mm Face Sheet	6.3 mm (0.25 in.)
Spectral Reflectivity	91%, 83%	73%
Number of Support Struts/ Heliostat	10	6
Number of Pointing Adjustments	75	27
Number of Focusing Adjustments	25	None
Main Structure		
Yoke Width	4.1 m (13.5 ft)	2.4 m (7.8 ft)
Yoke Height	3.7 m (12.0 ft)	3.6 m (11.9 ft)
Yoke-Vertical Member	0.3 m (1 ft) I-Beam	0.25 m (0.85 ft) I-Beam
Yoke-Horizontal Member	Steel, 0.3 x 0.3 m (1 x 1 ft)	Steel, 0.25 x 0.25 m (0.8 x 0.8 ft)
Drive Mechanism, Bearings	Direct Gear Drive	Direct Gear Drive
Foundation Interface	Caisson	Caisson
Rotational Limits		
Azimuth	±135° Primary, ±140° Secondary ±135° Primary, ±140° Secondary	±135° Primary, ±140° Secondary ±135° Primary, ±140° Secondary

d. Test Phase - June 1976 through December 1976 - Four heliostats have been assembled and erected in the MMC Solar Thermal Test Facility. Checkout and alignment procedures have been completed for each heliostat. Although all heliostats have received partial testing, one 25-mirror heliostat has been tested extensively.

The results of the testing conducted through September 1976 is discussed in detail in Section B.

3. Solar Thermal Test Facility (STTF)

With company funding, MMC has completed an extensive facility to test the four SRE heliostats as well as to evaluate other heliostat concepts and designs.

Figure III-6 shows the layout of the MMC facility. The facility is laid out along a north-south line and is intended to simulate the north collector field concept. Eleven heliostat sites have been surveyed and core samples of the underlying rock taken. To date, only four sites are occupied, i.e., by SRE heliostats as shown in Figure III-6. Specific site data is given in Table III-3.

To simulate the tower-mounted receiver a target is mounted on a ridge directly south of the SRE collector field. Although the height of the target with each heliostat varies depending on the sites, it averages about 88.1 m (289 ft).

The target consists of a combination calorimeter and radiometer. The calorimeter provides a means of determining the total energy reflected from a heliostat. The radiometer provides both a measurement of total energy and of flux distribution in images. From the two instruments, image quality can be precisely determined as a function of mirror quality and slant range. A detailed description of these instruments is contained in Section B.

Calorimetric and radiometric measurements require precise control of the heliostats. Each site is equipped with its own fine tracking sensor.

Additionally, each heliostat can be controlled by an executive control computer located in a central station, thus simulating 10 MWe field operation, as well as providing maximum flexibility and safety during testing. The approximate location of the control center in the STTF is shown in Figure III-6. A schematic diagram of the SRE control system is shown in Figure III-7.

Figure III-6 Layout of SRE Test Facility

Table III-3 Locations of SRE Heliostats with Respect to 10 MWe Collector Field

<u>Test Heliostat No.</u>	<u>Simulated Position</u>	<u>Slant Range m(ft)</u>
1	Row 16 7W	270.81 m (888.5 ft)
4	Row 21 7W	342.52 m (1123.75 ft)
9	Row 24 3W	400.56 m (1313.84 ft)
11	Row 27 22W	505.75 m (1657.29 ft)

Table III-4 Summary of Calorimeter Data

<u>Date of Run</u>	<u>Run No.</u>	<u>Time, MST</u>	<u>Mirror Performance Factor, MPF</u>
8/6/76	3	10:05 A	0.8618
	4	10:10 A	0.8891
	5	10:19 A	0.8229
	7	10:43 A	0.8618
	8	10:50 A	0.8489
	9	10:57 A	0.8312
	11	11:39 A	0.8149
	12	11:45 A	0.8161
	13	11:52 A	0.8164
8/7/76	3	8:49 A	0.8851
	4	8:55 A	0.8859
	5	9:00 A	0.9034
	7	9:39 A	0.8399
	8	9:44 A	0.8131
	9	9:50 A	0.8618
8/12/76	9	11:14 A	0.8483
	10	11:20 A	0.8498
	11	11:38 A	0.8800
8/14/76	7	9:40 A	0.8181
	8	9:44 A	0.8148
	9	9:52 A	0.8618
			0.8488 (mean)

Figure III-7 Schematic Diagram of SRE Heliostat Control System

During a test, heliostats are controlled from the trailer by a test conductor. The test conductor has full communications by telephone and radio to all heliostat sites and the calorimeter and radiometer. During tests, all data, including environmental, atmospheric and insolation are recorded.

B. SUMMARY OF COLLECTOR SUBSYSTEM ACTIVITIES, JULY 1976 THROUGH SEPTEMBER 1976

1. Calibration of SRE Calorimeter/Radiometer Test Facility

The calorimeter is the major instrument used for the Solar Research Experiment because it measures the total energy received on its collector plate. Because all of the energy received is not due to the heliostats reflecting sunlight, calibration tests were run to determine all inputs and losses so that valid data could be obtained. One of the major tests performed to validate data was energy measurement by means of a bank of radiometers. The radiometer data gave immediate solar flux distribution data and after integration gave total energy for comparison with calorimeter data. The data from both instruments correlated well, substantiating credibility for both test measurement techniques. Details of the two test programs are discussed in the following paragraphs.

a. Calorimeter Test Facility

The calorimeter for the solar research experiment is shown in Figure III-8 which shows the doors open with the black collector panels ready to receive energy. Water is circulated through the panels which consist of 53 fin tubes shown in detail in Figure III-9. Each fin overlaps the next by 1.2 cm (1/2 in.) as indicated by the dotted section. The temperature rise in the water across the panel is measured and the mass flow rate is closely controlled so that the total energy in watts or Btu/hr can be determined.

The energies into and out of the calorimeter balance each other when steady state operation is achieved. Figure III-10 shows the energy and loss contributions which are evaluated in performing a calorimetric test for the SRE. These are discussed in detail as follows:

Incoming Solar: The energy available at the mirror for reflection to the calorimeter is measured by means of an Eppley pyrheliometer which is tracked to maintain its collimation tube on the sun line.

Figure III-8 SRE Calorimeter

Figure III-9 Calorimeter Receiver Panel

Figure III-10 Calorimeter Energy Balance

Cos θ : The angle between the sun line and the normal to the heliostat effectively diminishes the reflecting area of mirrors. The total area must therefore be multiplied by the cosine of this angle. The angle and cosine are determined by computer based on the location of the heliostat and calorimeter, the time and date of the test, and sun position from the Ephemeris.

Shadow of Tracking Device: The solar tracker which automatically positions the reflected beam on the target always casts its shadow on the target and sometimes casts a shadow on the mirrors of the heliostat in addition. The same inputs used for determining cosine are used with sensor tower height to determine the shadowing effect for each test.

Heliostat: The mirrors of the heliostat have a reflecting efficiency less than 100% since some of the incident energy is absorbed and some is scattered. This efficiency combined with any losses due to defocusing or pointing error is called mirror performance factor (MPF). This is the end result to be determined from the energy measurement.

Atmospheric Attenuation: The energy incident on the mirror has already suffered attenuation due to the atmosphere but a second attenuation occurs in the atmosphere between the heliostat and the calorimeter. Figure III-11 illustrates this attenuation in which Q = the incident energy and Q_R = the received energy. The relation is the factor $T(\lambda) \rho_m$ which is the atmospheric attenuation. Figure III-12 presents the spectrum of the incoming energy, the spectral attenuation factor and the spectrum of the received energy. These data are determined by a computer program made available to us from National Oceanographic and Atmospheric Administration (NOAA). The data presented were based upon the following conditions:

Rel. Air Mass	- 1	30% RH
Precipitable H ₂ O	= 1 cm	T = 24°C (75°F)
Aerosols	= 200 Parts/cm ³	R _s = 305 m (1000 ft)
Qzone	= 0.35 cm	Elev = 2 km (6562 ft)
		Rel Loss = $\frac{Q-Q_R}{Q} \approx 3\%$

Indirect Solar Energy: The energy from the sun that is reflected from particulate matter in the sky and also from the ground is measured by a pyrnometer mounted so that it faces North (the same direction as the calorimeter) and calibrated so that it reads the same as if it were mounted in the center of the calorimeter with the doors open.

Assumes Mirror Reflectivity Is Wavelength Independent

Figure III-11 Atmospheric Attenuation

Figure III-12 Spectral Attenuation through Atmosphere

Energy Absorbed in Water: The water circulated through the calorimeter panels absorbs energy by increasing in temperature. The temperature increase is measured and when the mass flow rate is known, the energy can be determined in either watts or Btu/hr.

Reflected Solar: The absorptance α of the black painted panels has been determined to be 0.974 so the reflectivity is 0.026. The loss is then 0.026 times the energy incident on the face of the calorimeter. The characteristics of the surface was measured on a sample taken from the calorimeter by means of a Beckman DK-2 spectrometer and a Gier-Dunkle integrating sphere.

Emitted IR: The infrared loss due to radiation is computed by means of a computer program which figures the average temperature of each 15 cm (6 in.) increment of the receiver panel shown in Figure III-9. This program assumes a radiation pattern on the face of the calorimeter characteristic of the heliostat being tested and defines incremental temperatures and gradients in accordance with the predicted flux distribution. When the average temperature for each increment is determined, the radiation loss is computed with respect to a sink temperature which is measured looking at the North Sky and ground in the same view as seen by the calorimeter. The radiation loss is then proportional to the difference of the 4th power of the two temperatures and the total loss is the sum of the losses of the 48 x 53 increments or thermal nodes.

Front Convection: The loss due to wind and circulating air across the face of the calorimeter is determined by making an auxiliary run in which warm water is pumped through the panels with no sun being reflected onto them. From flow rate and temperature loss, film coefficients are determined which can then be applied to the 2544 nodes of panel area at their temperatures present during a run to determine the loss incurred. Figure III-13 shows the results of these evaluations for incident fluxes from either one or four heliostats as used in the Solar Research Experiment.

Back Conduction and Convection: By measuring the temperature on the back side of the insulation of the calorimeter and knowing the temperatures on the face side, this conduction loss is computed. Figure III-14 shows a cross section view of the calorimeter as used to compute the back side loss.

Calorimeter Measurements - After making a number of test runs and performing the corrections described above, the calorimetric measurements of heliostat mirror performance factor gave an average value of 0.849. Statistically the deviation of the measurements gave a three sigma value of 0.018.

Figure III-13 Calorimeter Front-Surface Convection Losses

Figure III-14 Cross Section of Calorimeter Used for Back Side Conduction Loss

b. Radiometer Test Facility

In order to verify results obtained with the calorimeter a set of 13 radiometers was mounted on a boom that was oriented normal to the line of site from the calorimeter under test. Figure III-15 shows the boom mounted on the calorimeter door post and oriented on the site 4 heliostat. Each radiometer was mounted on a water tube to provide cooling and connected to a galvanometer recorder. In use the solar beam from the heliostat is passed across the radiometer boom by moving the heliostat in azimuth only. Data are recorded from all 13 radiometers with respect to time of passage.

The data recorded from the radiometers are affected by the energies and losses shown in the energy balance of Figure III-16. Each of these inputs and losses was described in detail earlier and is identical for use with the radiometers.

Radiometer data analysis - The radiometer data as recorded on the strip chart are integrated with respect to time which is convertible to distance. When conversion factors are applied the data are in the form of $\frac{\text{watt}}{\text{m}^2 \text{ sec}}$ $\left(\frac{\text{Btu/sec}}{\text{ft}^2\text{-hr}} \right)$. This is shown pictorially in Figure III-17. The thirteen values of integrated data are then plotted with respect to the spacing of the radiometers. This curve is then cross integrated and multiplied by the cosine of the angle at which the center of the beam crossed the line of the radiometer beam. This will then give the total flux in watts or Btu/hr.

c. Results of Calorimeter and Radiometer Tests

Data from calorimeter tests on a number of days are shown in Table III-4. These data were computed without consideration of backside losses and without correcting for atmospheric attenuation. The actual MPF will be 2 to 5% higher when the effects of atmospheric attenuation are accounted for.

Radiometer tests were made on two different days at different times during the day. Data from these tests are shown in the following table. Again these data have not been corrected for atmospheric attenuation so that the actual MPF is several percent higher than 0.889.

Figure III-15 SRE Radiometer Boom

Figure III-16 Radiometer Energy Balance

Figure III-1/ Radiometer Data Reduction

Figure III-18 Flux Distribution through Center of Solar Image

Table III-5 Summary of Radiometer Data

<u>Date of Run</u>	<u>Run No.</u>	<u>Time, MST</u>	<u>Mirror Performance Factor, MPF</u>
8/12	6	10:28 A	0.8963
8/12	14	12:08 P	0.9207
8/14	2	8:48 A	0.8970
8/14	3	8:51 A	0.8417
8/14	10	10:03 A	0.8852
8/14	11	10:07 A	0.8962
			0.889 (avg)

Table III-6 Comparison of Calorimeter and Radiometer Test Data

Calorimeter MPF_{avg} = 0.849 (+6.4%, -4.2%)

Radiometer MPF_{avg} = 0.886 (+3.9%, -5.0%)

Table III-7 Calculated Uncertainty in Measurements

Calorimeter $\pm 4.22\%$

Radiometer $\pm 4.53\%$

Comparison of the calorimeter and radiometer test data taken to date is shown in Table III-6. Both the mean and the spread of the MPF are given. These values will approach the predicted 91% MPF when corrections for atmospheric attenuation are considered. A computer program to speed and simplify the computation of the attenuation based upon atmospheric data taken during tests is being developed.

Uncertainty analyses were performed on both the calorimeter and radiometer systems as shown in Table III-7. Calibration or experimental tolerances were assigned to each measured parameter. When the uncertainties are compared with the range of MPF values in Table III-6, close correlation is obtained.

The conclusion to be drawn from this comparison is that by using two completely different methods of measurement, mirror performance factors can be measured within a few percent. The correlation of results indicates no loss in energy that is unaccountable and confirms the validity of the measurements.

Flux distribution through solar image, i.e., the data from a single radiometer, can be used to plot the flux distribution of energy in the beam from a heliostat. Figure III-18 shows the plot of data from radiometer No. 8 which was near the center of the beam on the particular test illustrated.

2. Supplemental Tests

a. Sheet vs Float Glass - It is desirable to use low-iron float glass for heliostat mirrors since it gives a correspondingly higher reflectivity of the order of 91%. Low-iron float, however, is not available at this time. Therefore, this low-iron sheet glass was substituted for float glass in the 25-mirror SRE heliostats.

When mirrors were made with sheet glass, it was noted that fine-tracking could not be satisfactorily achieved using the sensing of the reflected beam from the center mirror as a control. The response was one of good tracking for a few minutes, followed by an unacceptable offset for a few minutes, then followed by a return to stable on-target tracking. It was found that this unstable tracking condition was due to the irregular reflections due to ripples in the sheet glass formed during the manufacturing process. Subsequent tests using a laser and calibrated target indicated that the ripples were causing beam deviations or effective scattering of the beam of the order of 10 arc-minutes. It was therefore determined that present day sheet glass cannot be used for the center mirror which provides the beam for fine-track control. Float glass does not create these

problems. Two heliostats are currently being refitted with float glass with specular reflectivity of 83%. Tests will be conducted in the future to determine if the integrated loss due to scatter will make sheet glass mirrors acceptable for close-in heliostats so that advantage can be taken of the higher reflectivity. At the same time, efforts are underway to obtain low iron float glass.

b. Effect of Pull Plate Size on Image - The diameter of the center pull plate of a variable focus mirror controls the shape of the warped surface. If the diameter is small, the warping tends to approach a cone. As the diameter increases, the warping approaches a parabola of revolution.

In order to more fully evaluate this condition, a computer program was designed to simulate the glass stresses and distortions. Figure III-19 shows the results of this analysis. It does not apply any judgment factor relating material cost or weight to the problem. Since these constraints lead toward smaller pull plates, a number of actual mirrors were made using five sizes of pull plates. Tests were then made to determine image size relative to pull plate diameter. As a result of both the analytical evaluation and the confirming tests, it was decided that 53.3 cm (21 in.) was the optimum size for use with the 1.2 meter (4 ft) mirrors. Figure III-20 shows curves of the tests made using the different diameter pull plates. The tests indicate that larger pull plates increase image quality. The two heliostats being refitted with float glass are being refitted with 55.9 cm (22 in.) diameter plates.

c. Fixed Focus Mirror Development - Looking toward minimizing costs and complexity, a heliostat consisting of only nine mirrors instead of 25 has been built and is undergoing test. The mirrors on this heliostat provide the same reflective area as does the 25 mirror with only 27 adjustments as contrasted to 75.

These larger mirrors are pre-focused during fabrication and do not have focusing adjustments or require focusing adjustment on installation. Figures III-4 and III-5 show the front and back views of the nine mirror heliostat with one mirror still to be installed.

To date, 10 of the 2 m (80 in.) square mirrors have been fabricated. The first mirror was a prototype and utilized fibreglass back sheets, subsequent mirrors were made using a combined fibreglass and aluminum back sheet for increased stiffness. One of the aluminum backed mirrors was made using aluminum honeycomb while all the remaining eight used paper honeycomb. Combined steel and fibreglass brackets were cemented to the backing sheet to permit mounting and pointing adjustment. The mirrors are now all installed and undergoing preliminary testing.

Figure III-19 Distortion of Mirror Surface vs Pull Plate Size - Computer Model

1.2x1.2 m (4x4 ft) Laminated Sheet Glass Mirrors, 2.5 Face Sheet

Focal Distance: 106.7 m (350 ft)

Measured by Normal Incidence Pyrheliometer, 5.7° FOV

Figure III-20 Energy Distribution as a Function of Pull Plate Diameter

Tests have been made on the prototype and on one of the SRE, aluminum backed mirrors to determine the image size and shape when the beam was reflected the design distance of 400 meters. Figure III-21 shows the image of the sun projected on the doors of the calorimeter. It should be noted that the face of the calorimeter is vertical so that the image when projected upward at an angle appears to be oval. The results of the test are summarized in Table III-8.

Figure III-21 Image Quality Testing of Prototype Fixed-Focus Mirror

Table III-8 Results of Fixed Focus Mirror Testing

Test

Samples: Prototype, 1st SRE
 Size: 2.0 x 2.0 m (80 x 80 in.)
 Mirror: 6 mm, Float, Back-Surface
 Silver, ASG Glass
 Slant
 Range: 400.2 m (1313 ft)

Test Results

Measured Image Quality

Mirror	Major Dia, m (ft)	Minor Dia, m (ft)
Proto	5.1 (17)	3.7 (12)
SRE 1	4.9 (16)	3.7 (12)

Calculated Image Quality

Theoretical Sun Dia:	3.72 m (12.2 ft)
$d_s = 0.012 R_s$	4.80 m (15.8 ft)

IV. RECEIVER SUBSYSTEM REQUIREMENTS

A. PILOT PLANT RECEIVER

1. Description of Concept and General Requirements

The receiver subsystem provides the necessary function within the Central Receiver Solar Thermal Power System of final collection of the directed solar energy and transfer of that energy into a working fluid for subsequent conversion within a turbine/generator unit to electrical power. The cavity concept has been determined to be the most efficient configuration for receiving focused solar energy since it minimizes thermal losses both during operation and during overnight shutdowns.

The working fluid used in the pilot plant receiver is water delivered in the form of superheated steam at 9,136 kPa (1325 psig) and 789 K (960°F), to either the EPGS or the thermal storage subsystem. The steam generating concept selected uses a natural circulation boiler.

The major requirements for preliminary design of the pilot plant receiver are tabulated in Table IV-1. The aperture size was established to maximize the net energy collected. A yearly average of better than 98.5% of the energy projected toward the receiver is captured by the aperture. The receiver tilt angle was established by the geometric relationship of the heliostat field to the receiver. The energy capacity and flow rate capability were based upon the maximum probable energy reflected by the heliostat field. The steam conditions were established as a compromise between attaining a high thermodynamic efficiency and obtaining a high cyclic lifetime turbine.

The selection of a conventional natural circulation boiler concept for the receiver placed a high degree of importance upon the development of a suitable cavity configuration. The major factors which influenced the size, shape, and orientation of the cavity were the heliostat field layout, tower height, cavity efficiency and heat flux profiles.

Figure IV-1 depicts the selected cavity configuration. Dimensions shown are for inside surfaces at ambient temperature. Also shown are the projections of heliostat solar beams which bound the directly insulated surfaces of the cavity. Active steam generating tubing must occupy most of the two 6.706 m (22 ft) wide side walls

Table IV-2 Receiver Requirements

IV-2

Aperture Opening	7.498 m (24.6 ft)		
Tilt of Receiver Centerline	0.349 $\pm$ 0.087 rad (20 $\pm$ 5 deg)		
Solar Energy to Receiver			
Receiver Design Point	52 MWth		
Feedwater Input			
Flow Rate	69,400 kg/h (153,000 lb/hr)		
Pressure	10,860 kPa (1,575 psig)		
Temperature	479 K (403°F)		
Receiver Output			
Flow Rate	69,400 kg/h (153,000 lb/hr)		
Temperature	789 K (950°F)		
Pressure	9136 kPa (1,325 psig)		
Minimum Receiver Efficiency		Wind Speed	
	<u>m/s</u>	<u>(mph)</u>	<u>Efficiency</u>
	0	0	95
	4.47	10	94
	6.71	15	93
Cycle Life			
Diurnal Cycles	13,000		
Cold Starts	1,300		
Startup Time			
Over Night	45 min		
Cold	6 hr		
Winds	33.5 m/s (75 mph) Max Operating		
	100.6 m/s (225 mph) Max

IV-3

Figure IV-1 Cavity Dimensions

and the entire 6.096 m (20 ft) wide rear wall. All remaining walls will be inactive with reflective surfaces. Allocation of specific zones for boiler walls and superheater panels will be discussed in Chapter V.

The aperture was sized to accommodate two divergent considerations. The opening should be large enough to capture most of the reflected energy from all heliostats. However, increasing the aperture size also increases thermal losses from the cavity by radiation and convection. Therefore, the optimum aperture size is that which minimizes the total of energy excluded from entering, plus radiation and convection losses from the cavity. The 7.498 m (24.6 ft) square aperture selected for the pilot plant receiver excludes an average of approximately 2% of the energy projected from the heliostat field, and loses about 5% by radiation and convection from the cavity.

Input Thermal Energy Characteristics

An accurate prediction of heat flux patterns produced by the heliostat field within the cavity is vital to the proper design of the receiver steam generating elements. First of all, the available active wall area (that which receives direct solar flux from the field) must be divided into boiler and superheater zones so that the total energy into each of those zones is compatible with specified steam generation rates, pressures and temperatures. The second factor of importance is knowing the magnitude and location of peak heat fluxes, which can induce destructive asymmetric thermal stresses within individual steam tubes.

The thermal radiation analysis system (TRASYS) program was used to determine cavity internal heat flux profiles for several combinations of days during the year and times of day. The days selected were the spring equinox (which produces very nearly the same results as the autumnal equinox), the summer solstice, and the winter solstice. Heat flux predictions were run for solar times of 8:00 a.m., 10:00 a.m., and noon for each of the three days; and also at 6:00 a.m. for the summer solstice. No afternoon runs were made in this study since the symmetry of the problem on the summer and winter solstice afternoons produce mirror images of the morning results. This is not exactly true for the case of the equinox days, but is close enough to preclude the necessity of expending additional computer run time. Insolation values for these cases were determined by calculating an atmospheric extinction factor which produced an insolation of 0.969 kw/m^2 (90 watts/ft^2) at noon on a given day, then using that extinction factor to calculate insolation values at the other times for that day.

Two additional cases of significant interest were run. One was for the pilot plant system design point conditions which are an insolation of 0.8255 kw/m^2 at 2:00 p.m. on winter solstice. The other case of interest is that which produces the maximum design heat flux and total energy rate within the receiver, which was taken as noon on the winter solstice at an insolation of 1.02 kw/m^2 (95 watts/ft^2). The heat flux profile incident upon the three active receiver interior surfaces for the maximum flux case is shown in Figure IV-2.

The results of these TRASYS analyses are summarized in Table IV-2. The field average cosine is defined as the arithmetic average of the angles of incidence for all heliostat mirrors in the field. The aperture efficiency is defined as the ratio of the net energy entering the receiver cavity to the total energy directed toward the receiver from the heliostat field. The combined concentration efficiency is the product of the field average cosine, mirror reflectivity, and aperture efficiency. This combined efficiency represents that percentage of solar energy striking a surface normal to the solar vector (and which has an area equal to the total mirror area in the heliostat field), which actually enters the aperture of the receiver cavity.

A very important consideration in establishing the size and locations for superheater surfaces, is the proper proportioning of total energy between boiler and superheater surfaces. At the nominal feedwater inlet and superheated steam outlet conditions specified in section IV-A-1, 72% of the energy added to the working fluid produces saturated steam in the boiler tubes, and 28% must be absorbed in the superheater tubes. Some imbalance in those proportions can be tolerated, but must be minimized for an efficient system. The superheater surface area is designed by the minimum anticipated proportion of energy incident upon it. The spray attemperator is then sized to accommodate the maximum anticipated proportion of energy on the superheaters. Figure IV-3 illustrates the locations of the six superheater panels in the pilot plant receiver. Table IV-3 summarizes the direct incident energy on the panels for several points in time during the year. The significant factor is that although the incident energy upon any given superheater panel varies considerably from time to time, the total proportion of energy incident upon all six panels varies only by a small amount--from a minimum of 27.5% to a maximum of 31.6%.

Figure IV-2 Pilot Plant Receiver Heat Flux Profile

Table IV-2 Pilot Plant Solar Concentration Summary

Day of Year	Solar Time, hr	Ambient Insolation, kW/m ²	Field Average Cosine	Mirror Reflectivity	Aperture Efficiency	Combined Concentration Efficiency	Energy into Receiver, MW
Spring Equinox	0800	0.692	0.807	0.85	0.982	0.674	29.75
	1000	0.920	0.905	↑	0.988	0.760	44.60
	1200	0.969	0.942		0.990	0.792	49.01
Summer Solstice	0600	0.304	0.624		0.966	0.512	9.95
	0800	0.784	0.750		0.977	0.623	31.17
	1000	0.932	0.844		0.983	0.705	41.94
	1200	0.969	0.874		0.985	0.732	45.28
Winter Solstice	0800	0.441	0.870		0.98.6	0.729	20.51
	1000	0.897	0.927		0.991	0.780	44.68
	1200	0.969	0.954		0.992	0.805	48.78
(Max Flux Case)	1200	1.024	0.954		0.992	0.805	52.55
(System Design Case)	1400	0.826	0.927	0.85	0.991	0.780	41.09

Figure IV-3 Receiver Surface Node and Superheater Locations

Table IV-3 Incident Solar Energy on Pilot Plant Receiver Superheaters

Day of Year	Solar Time, hr	Incident Energy, MW							Total Re-ceiver	Energy Percent on S/H
		On Superheaters								
		1	2	3	4	5	6	Total		
Spring Equinox (80)	0800	1.413	1.840	2.045	1.841	1.343	0.684	9.166	29.75	30.8
	1000	2.368	2.736	2.453	2.586	2.205	1.036	13.384	44.64	30.0
	1200	2.856	2.994	2.856	2.615	2.615	1.228	15.164	49.01	30.9
Summer Solstice (172)	0600	0.362	0.564	0.729	0.712	0.374	0.268	3.009	9.95	30.2
	0800	1.403	1.817	2.130	1.958	1.415	0.855	9.578	31.17	30.7
	1000	1.845	2.448	2.247	2.496	2.111	1.113	12.287	41.94	29.3
	1200	2.218	2.650	2.218	2.484	2.484	1.031	13.085	45.28	28.9
Winter Solstice (355)	0800	1.168	1.287	1.388	1.220	0.913	0.498	6.474	20.51	31.6
	1000	2.084	2.862	2.490	2.536	2.105	1.001	13.078	44.68	29.3
	1200	2.265	3.184	2.265	2.492	2.492	1.007	13.705	49.78	27.5

B. RESEARCH EXPERIMENT TEST SYSTEM

1. Purpose

A reduced scale model research experiment receiver is to be designed, fabricated, and tested for the purpose of verification of the Central Receiver Solar Thermal Power System receiver concept and to investigate operational performance characteristics. An infra-red (I.R.) heat source will be used to simulate solar energy input. This test program will be conducted at the Sandia Laboratories Radiant Heat Facility, Albuquerque, New Mexico. The receiver design requirements, test support functions, and test system interfaces have been defined in detail in the "Design Criteria and Interface Requirements for the 5 MWth Research Experiment Receiver and Test System" document, released by Martin Marietta Corporation on 25 November 1975. Therefore, only the key elements of the research experiment program will be summarized herein.

2. Test System

The I.R. test program will be conducted with the receiver mounted at ground level on a specially constructed test pad at the Sandia Laboratories Radiant Heat Facility. Electrically energized radiant lamps will provide the heat source. The radiant heater arrays have been designed and fabricated to mount within the receiver cavity, parallel to and at a distance of approximately one foot from the three active steam generating surfaces.

The test receiver has been designed to generate superheated steam at the same exit pressure (9,136 kPa) and temperature (789 K) specified for the pilot plant receiver. Steam flowrate at a steady state 5 MW thermal input will be 6,561 kg/h (14,460 lbm/h) as compared to 51,963 kg/h (114,560 lbm/h) for the pilot plant receiver at the system design solar energy input of 39.6 MW.

The closed loop steam/water cycle to be used is shown schematically in Figure IV-4. Superheated steam exiting the receiver flows through a heat exchanger which reduces steam temperatures to very near the saturation point, while heating the boiler feedwater to the required receiver inlet temperature. The steam is then condensed in an air cooled condenser, where treated makeup water is added as required. The saturated condensate at 438 K (328°F) flows through a subcooler to reduce its temperature to below 380 K (225°F), which is the maximum operating temperature for

the feedwater pump and heater units to attain specified pressure and temperature, and is again converted to superheated steam in the receiver. Proper feedwater chemistry is maintained by a treatment module located just upstream of the feedwater pump.

Data from approximately 450 test points (temperatures, pressures, flowrates, strain, power levels, and others) will be monitored and recorded at intervals for subsequent test analysis and evaluation. Certain of the test points which are critical to test system operation will be displayed continuously on a central console, which will also contain necessary controls for system operation in normal and emergency modes.

Various test runs will be performed to simulate receiver steady state operation under several different conditions of heat flux distribution and to evaluate startup and shutdown transient characteristics (see section V).

3. Test Support Equipment

Two major subassemblies for the Research Receiver Test System have been designed and fabricated by Martin Marietta Corporation. They are the test control console and the Mechanical Equipment Module. The Test Control Console provides remote control and monitoring functions for the receiver, condenser and mechanical equipment module. The mechanical equipment module performs feedwater pumping, steam and water pressure regulation, and thermal conditioning functions for the test system.

The test control console is pictured in Figure IV-5. The console has been completely assembled, checked out, and delivered to the Sandia Laboratories Albuquerque test site.

The mechanical equipment module is shown in Figure IV-6. It has successfully completed checkouts, hydrostatic testing, and has been delivered to the test site.

Figure IV-5 Test Control Console

Figure IV-6 Mechanical Equipment Module

SECTION V
RECEIVER SUBSYSTEM

BY :

S. F. WU

R. J. ZOSCHAK

FOSTER WHEELER ENERGY
CORPORATION

JOHN BLIZARD RESEARCH CENTER
12 PEACH TREE HILL ROAD
LIVINGSTON, NEW JERSEY 07039

7. RECEIVER SUBSYSTEM DESIGN

A. INTRODUCTION

This report covers the period from the start of this project through September 1976. During this period, three Quarterly Reports have been issued consecutively in January, April, and July 1976. This report briefly summarizes the progress reported in these previous Quarterly Reports and describes in greater detail the activities during the last quarter, from July 1 to September 30, 1976. These activities include a preliminary seismic analysis and a supporting structure trade-off study for the pilot plant receiver and preparation of the Test Procedures; completion of the detailed design task; and continuation of procurement, fabrication, and field erection for the experimental receiver.

B. PILOT PLANT RECEIVER

1. General Design Description

The basic approach to the design of the pilot plant receiver was to make maximum use of existing boiler design and fabrication technology consistent with the special conditions imposed by the operating requirements of the solar plant.

The boiler (receiver) is of the natural-circulation type, as opposed to a forced-recirculation or once-through type. Natural circulation was selected because it is simple, easily adapted to the given cavity configuration, and particularly suited to the 10 MWe receiver module because of its long history of reliable operation in utility, industrial, and marine boilers of comparable capacity operating at similar pressure and temperature conditions.

Simplified sectional side elevation and plan views of the receiver are shown in Figure V-1. The rear wall and a large portion of both side walls of the cavity are covered with 38.1-mm (1-1/2 in.) O.D. carbon steel boiler tubes on 44.5-mm (1-3/4 in.) centers using MonowallTM construction in which the tubes are joined together along their length by continuous welded integral fins to form flat panels. The superheater consists of 6 passes in series, 2 passes on each of the side walls and 2 on the rear wall. Each pass is made from a number of parallel 25.4-mm (1-in.) O.D. austenitic stainless steel tubes arranged side by side on 28.6-mm (1-1/8 in.) centers to form one or more platens. These tube platens are placed in front (on the cavity side) of the waterwall panels, as shown in Figure V-2. All superheater headers and transfer lines between passes are located behind the waterwall panels. Superheater

Figure V-1 Side Elevation View - Pilot Plant Receiver

Figure V-2 Arrangement of Superheater Passes - Pilot Plant Receiver

tubes penetrate through openings in the waterwall panels at the inlet and outlet of each pass and at intervals along each pass where expansion loops and intermediate supports are located.

Water flows from the steam drum through 6 external downcomer pipes and 28 branching feeders to the lower side and rear wall headers. Here the flow is divided among the various upflow circuits in the waterwalls (boiler tubes). As the water flows upward through the tubes, a portion of it is converted into steam by the absorbed heat. The resulting mixture of water and steam leaving the tubes is collected in the upper side and rear wall headers and carried to the steam drum through 40 riser tubes. In the drum the mixture passes through centrifugal separators which separate the steam from the water. The water, after mixing with incoming feedwater, enters the downcomers for another trip around the circuit while the steam passes through chevron-type mist eliminators, or driers, for removal of any entrained water droplets. From the drum the dry saturated steam flows through a number of pipes to the superheater, where it is heated to its specified outlet temperature.

Feedwater flow to the receiver is controlled by a conventional 3-element feedwater regulator which matches feedwater flow to steam flow, with a trimming override in response to drum water level. Superheated steam temperature is controlled by attemperators in which feedwater is sprayed into the superheated steam and evaporated, thus lowering the steam temperature. Two attemperators are used, one between the second and third passes and the other between the fourth and fifth passes.

2. Thermal/Hydraulic Analysis and Design

a. Basis of Analysis and Design - The applicable correlations used in thermal/hydraulic analysis and design of the receiver were all taken from well-established sources. The preliminary pilot plant receiver design was based on the maximum insolation condition which corresponds to the heat input to the receiver at 1200 on winter solstice at an insolation of 1.023 kW/m^2 . The receiver was designed for a superheater outlet condition of 789 K (960°F), 9,136 kPa (1325 psig) and 69,400 kg/h (153,000 lb/h). The physical dimensions of the receiver were built up around the cavity internal dimensions established by Martin Marietta. Using this fundamental cavity size and geometry, boiler and superheater surfaces were arranged on the active walls, flow connections and circulating components were placed outside the cavity, and an enclosure and insulation were then added to box up the receiver.

b. Superheater Surface Arrangement - The heating surface in the receiver cavity has to be correctly proportioned between superheater and boiler sections so that the designed superheater outlet conditions

can be obtained. Because of the uneven heat flux distribution on the heating surfaces, surface arrangement became very complicated, and a cut-and-try process had to be followed. For each proposed scheme, the procedure was first to assure that heat energy absorbed by the superheater surface as a whole met its requirement. The superheater surface was then divided into a suitable number of passes and connected in a desired order so that sufficiently high steam mass velocity could be obtained without exceeding allowable pressure drop across the whole superheater. For each superheater pass, the average fluid bulk temperature and film coefficient were calculated. With this information and heat flux values taken from the distribution curves, thermal stresses were spot-checked for the critical locations. If the stresses were too high, a different scheme was substituted for the previous scheme, and the same procedure was repeated.

The final selected superheater surface arrangement as shown in Figure V-2 had the superheater passes located outside high flux zones of both rear and side walls. In this arrangement each side wall pass was divided into two subpasses, and steam flow was split into one upper and one lower subpass as shown in this figure. By so doing, the regions of high fluxes were covered by the initial superheater passes which have lower steam temperatures. In addition, the vertical heat flux variation in these passes was reduced by approximately half of what it would be were these passes not divided. This reduces variation in flow distribution and steam temperature rise among the tubes.

For this preliminary baseline design, pressure drop and temperatures of steam and tube wall were calculated for the entire superheater using Foster Wheeler computer programs. The pressure drop across the superheater was predicted as 1620 kPa (235 psi), not including the loss through two attemperators which would amount to approximately 138 kPa (20 psi).

c. Boiler Circuitry Arrangement - The boiler section of the receiver uses the natural-circulation principle. In a natural-circulation system, the rate of flow that can be produced is governed by flow resistances and differences in density between the downcomer circuits and the upward heated circuits. Control of these resistances will enable the designer to apportion an adequate flow of water to parallel circuits. Criteria imposed on the design of boiler circuitry are the entrance velocity to and the exit steam quality from a heated tube. At the drum operating pressure of 10,860 kPa (1575 psig), the entrance velocity should be no lower than 0.61 m/s (2 ft/s) and the exit steam quality no greater than 15 percent by weight.

The boiler circuitry design and circulation calculation were also repetitive in nature. The calculation was done using a Foster Wheeler computer program, and results were interpreted with the aid of an

automatic plotter. Results of the circulation analysis for the boiler circuitry selected for the preliminary baseline design indicated that all circuits have satisfied the circulation design criteria. The overall circulation ratio for this arrangement was calculated as 10.1, which was equivalent to an average exit steam quality of approximately 10 percent by weight.

d. Start-up Transient Analysis - The receiver transient operation modes analyzed included hot start-up after overnight shutdown, restart after cloud passage, and cold start-up from ambient condition. Hot start-up is a diurnal morning start-up of the receiver after overnight shutdown. Since the receiver accepts all the insolation available during the start-up period in this type of start-up, the governing parameter is the rate of increase of drum pressure. Restart after cloud passage requires the receiver, which was operating at the normal load condition before being shut down by cloud cover, to restart as soon as the cloud disappears. Because the receiver is expected to be started up from its full pressure condition, the rate of increase of incident energy becomes the dominant parameter for this type of restart. Cold start-up is anticipated when the receiver is first put into operation or is started after a duration of shutdown longer than 72 hours. Starting from ambient condition, the receiver cannot absorb the fully concentrated solar energy without overheating its components, especially during the early period of start-up when there is no steam being generated. Therefore, the rates of increase of both incident energy and drum pressure play the governing roles in this type of start-up.

The study of start-up transients involved the development of a computer program for transient overall receiver heat balance calculation and the use of an existing program for local tube temperature and stress evaluation. The overall heat balance program uses a group of lump-mass nodes as a receiver model, approximates a transient period by a sequence of steady-state intervals, and performs simplified heat balance calculation for each time interval. The output of this program gives predicted pressures, steam flow, atomizer spray flows, steam temperatures, average tube wall temperatures, and average superheater steam film coefficients during a transient period.

For each case of transient analysis, the overall heat balance program was first run with an assumed drum pressure ramping rate, or an increasing rate of incident energy, or both, depending on the type of start-up. After the computer output of this program was examined, the critical locations on the superheater passes were selected for more detailed analysis. This detailed temperature distribution analysis was performed using the program ANSYS with a two-dimensional tube wall model. The program output of this tube model was then used to check whether or not, at any instant of the start-up period, the wall temperature and front-to-rear face temperature difference of the superheater tubes

exceeded their respective design limits. If so, a slower start-up rate was programmed in the next run.

The start-up cases analyzed were summarized along with their pertinent results in Section V.A of Quarterly Report No. 2 of this project. In this report, only the results of a typical hot start-up transient are presented in Figures V-3 and V-4.

3. Structural Analysis and Design

To ensure the structural integrity and lifetime requirements of the pilot plant receiver, detailed analyses of the superheater, boiler, and other pressure parts were performed. The methods used in these analyses and the important results were presented in Section V.A.3 of Quarterly Report No. 1. A brief summary follows.

a. Applicable Codes and Standards - The primary document used in sizing the components of the receiver is the ASME Boiler and Pressure Vessel Code, Section I. In design of solar receivers, creep and fatigue are important modes of failure to be considered because of the thermal cycling caused by diurnal start-up and shutdown. Section I does not have explicit criteria to evaluate these failure modes. Hence it was decided to supplement Section I with ASME Code, Section VIII, Division 2. However, Section VIII, Division 2 also does not have criteria to evaluate fatigue at temperatures in the creep regime, i.e., above 700 K (800°F). The only document which contains such criteria is Code Case 1592. Because this is a Code for Nuclear Components, Code Case 1592 has, of necessity, very stringent requirements, and its use will result in overly expensive solar receivers. However, since no solar boiler code or design rules have been developed, the Code Case 1592 approach was used in evaluating the structural integrity of the superheater tubes.

b. Superheater Analysis - The superheater is one of the most critical components of the receiver because it is subjected to high and uneven metal temperatures and stresses, and hence much structural analysis effort was concentrated on the superheater. Two types of superheater tube-support schemes were considered. The first scheme, which uses the stringer-and-sleeve-support concept, is better from structural rigidity and vibration points of view. However, sleeves prevent the tubes from bowing out at the support points and consequently introduce unacceptably high thermal stresses. The second scheme, which employs the "floating-tube" concept, allows the tubes to bow out and reduces the thermal stresses considerably. The latter scheme was chosen for the preliminary baseline design.

Figure V-3 Predicted Pressures and Flow Rates for a Hot Startup Transient - Pilot Plant Receiver

Figure V-4 Predicted Feedwater and Steam Temperatures for a Hot Start Transient - Pilot Plant Receiver

The thermal stress calculation was performed using the finite element program ANSYS. A plane strain analysis, i.e., assuming that the tube is not allowed to bend or expand axially, was first made using a two-dimensional tube model. The axial force and moment corrections were evaluated separately to reflect the actual tube support conditions. A continuous beam analysis was then done, and the stresses were added algebraically to the plane strain solution. From these results further calculations were made to check the Section VIII Code compliance and to estimate the fatigue and creep life based on the Code Case 1592 approach. To evaluate the fatigue life more realistically, inelastic analysis was also performed using a different computer program, MARC.

Superheater Passes 2, 3A, and 5A were identified as the most critical, judging from their anticipated heat flux, metal temperature, and stress levels. Results of stress calculations indicated that all three passes satisfied the Section I and Section VIII Code requirements. The allowable number of cycles for each of these passes as predicted by the inelastic fatigue analysis was greater than that specified in the design criteria. For the most critical pass, the creep life corresponding to the peak heat flux condition was found approximately equal to the estimated operating time at that condition, while at lower heat flux conditions, the corresponding creep life was for longer than the estimated operating time.

c. Boiler (Waterwall) Analysis - The determination of the boiler tube wall thickness was done according to the Code Section I formula. The thermal stress analysis was done using a finite element model which simulated the MonowallTM construction. Since the boiler tubes operate at temperatures below the creep regime, Code Section VIII was used for stress calculations and fatigue life evaluation. The results indicated that the boiler section has satisfied all design requirements.

d. Other Pressure Parts Calculations - The thicknesses of the downcomer, waterwall headers, feeders, risers, superheater transfer pipes and headers, and steam drum were calculated according to Code Section I. These pipes and headers, which are located outside the heated zones, were also evaluated for transient response and cyclic effects. The allowable heat-up or cooldown rates for pipes and headers of different thicknesses were defined for 10^5 cycles of fatigue life.

4. Preliminary Seismic Analysis and Support Structure Design

a. Interface Background - Seismic induced accelerations were calculated by Bechtel for an 0.3 g and an 0.2 g ground acceleration for both a 132-m (432-ft) and a 101-m (332-ft) tower. The results, shown as vertical and horizontal response spectra, were presented in Section VII.D.1 of Quarterly Report No. 3. These driving forces will induce significant

loads into the receiver support structure mounted on the top of the tower. To determine the domain in the response spectra in which the receiver and its support structure would fall, the natural frequencies of the receiver support structure were first calculated. The preliminary design and analysis of the support structure were then made using the equivalent static "g" method. Since a massive steel structure was found necessary to take the high seismic loads, the concept of supporting the receiver inside the concrete tower was considered. A trade-off study of concrete vs. steel supporting structure was also made to sustain this receiver-in-tower approach. The methods and results of the above-mentioned analyses are reported in greater detail as follows.

b. Prediction of Natural Frequencies - The natural frequencies of the receiver support structure were predicted using the computer program STRDUL. The geometry of the supporting structure assumed in the calculation is shown in Figure V-5. Since this structure has about 600 degrees of freedom, a three-dimensional analysis would be too costly to conduct at this preliminary design stage. Two plane frame analyses were consequently made in the YZ (side) and XY (front) planes as defined in Figure V-5. The three-dimensional effect, however, was approximated by using equivalent stiffnesses and masses. Heavy steel sections were assumed for the columns, beams, and braces to stiffen the supporting structure.

The first four natural frequencies were calculated as 7.1, 22.6, 27.4, and 39.5 Hz in the YZ plane, and 4.8, 20.8, 21.1, and 25.7 Hz in the XY plane. The modes corresponding to these frequencies were found to be predominantly horizontal. The vertical frequencies were postulated to be much higher. In an attempt to obtain higher natural frequencies so that they may fall outside the frequency ranges of high spectral accelerations, the stiffness of the braces was increased by a factor of 3 in the extensional mode and a factor of 10 in the bending mode. The results indicated that this increased the first fundamental frequency in the YZ plane from 7.1 to 8.9 Hz. A few other arrangements of the braces were also considered. However, the frequency did not increase substantially. From the above finding it is apparent that higher natural frequencies are not achievable unless an inordinate amount of steel is used. Hence the receiver supporting structure has to be designed for these frequencies and the associated high accelerations.

c. Preliminary Seismic Analysis of Steel Supporting Structure - The preliminary steel supporting structure design is shown schematically in Figure V-6 along with the column locations at the interface. The structural analysis was made based on the following assumptions:

- The equivalent static "g" method is used for seismic analysis. The equivalent static acceleration in the horizontal direction is 4.5 g. This number corresponds to a tower height of 101 m (332 ft),

Figure V-5 Analytical Model for Predicting Natural Frequencies of the Receiver Support Structure.

Figure V-6 Preliminary Support Structure Design and Column Locations for the Pilot Plant Receiver.

ground acceleration of 0.2 g, and damping of 2 percent and also represents approximately 1.16 times the peak acceleration anticipated for a tower height of 132 m (432 ft) at the same ground acceleration and damping. Since the vertical natural frequencies are high, the equivalent static acceleration in the vertical direction is taken as 1 g.

- The lateral seismic forces act nonconcurrently in the directions of the main axes (X and Z). This is according to the requirement of the Uniform Building Code.
- The weight of the receiver is 3.4×10^5 kg (7.5×10^5 lb). For the purpose of this analysis, this weight is assumed to be lumped equally at nine middle nodes on the top grid.
- For the first iteration, the weight of the steel frame excluding ties is assumed to be 5.67×10^5 kg (1.25×10^6 lb). This weight is lumped equally at all nodes of the frame.
- The receiver is connected by 12 horizontal members to the front and back of the steel frame. The lateral seismic forces on the receiver ($4.5 \times 3.4 \times 10^5$ kg) are transmitted to the steel frame by these members. The lateral seismic forces are assumed to be shared equally by all ties.
- The steel frame is assumed to have rigid joints and is also assumed to be connected rigidly to the tower platform.
- Solar beam clearance for supports is 0.61 m (2 ft) in the plane of the aperture.
- Receiver tilt angle remains at 20 degrees.
- Cavity dimensions remain the same as those on which the preliminary baseline design was based.
- The receiver will be located at the same elevation above the tower platform as shown on the latest issue of the pilot plant receiver general arrangement drawing.

A three-dimensional stress analysis was done using the computer program STRUDL. The results showed that the preliminary steel structure design as depicted in Figure V-6 was adequate for the seismic loads considered.

d. Trade-Off of Concrete vs. Steel Supporting Structure - The foregoing analyses showed that the natural frequency of the receiver supporting structure would coincide with a high spectral acceleration as predicted by Bechtel's analysis of the tower and that a massive steel structure would be required to withstand the high seismic loads. Accordingly, it was generally agreed among Martin Marietta, Bechtel, and Foster Wheeler that it would be better to suspend the receiver from a steel gridwork supported by an upward extension of the concrete tower wall. The design information was then developed for this approach. It consisted of the minimum tower inside diameter required to enclose the receiver and estimate of mass, center of gravity, snubber point locations, and loads of the receiver. A cost trade-off was also done to support this contention which showed that the cost delta for receiver within tower, including the top support grid, as compared with the structural steel approach was approximately minus \$1 million.

The advantages of concrete over steel supporting structure, from the receiver point of view, are listed below:

- Concrete has a higher damping coefficient (7 percent) compared with steel (2 percent). This reduces the lateral seismic forces considerably. For example, for the 132-m (432-ft) tower, the lateral seismic force in the concrete supporting structure would be only 50 percent of that in the steel structure. If the concrete supporting structure is monolithic with the tower, this force may even be less.
- If the receiver is supported inside the tower, it need not be designed for the peak wind loads, provided the top of the tower is roofed over.
- Concrete supporting structure provides more flexibility in locating the connections from receiver to support structure.
- Analysis and design of the concrete supporting structure should be simpler.
- There would be less hazard to personnel working on a receiver located within the tower.
- The receiver-in-tower approach has higher aesthetic value, better appearance.
- It should be easier to maintain a receiver located within the tower. Painting requirements should be reduced.

- It will probably be simpler to protect concrete from stray flux around the aperture.
- Erection of the concrete support will probably be easier because it is essentially a continuation of the process used on the lower portion of the tower.

C. RESEARCH EXPERIMENT RECEIVER

1. Objectives, Basic Requirements, and Approach

The objective of the receiver experiment is to test a scaled-down version of the pilot plant receiver while simulating, as closely as possible, the operating conditions anticipated for the pilot plant receiver, to prove that the design is satisfactory, and to confirm the analytical procedures used in design.

The need for simulating the pilot plant receiver operating conditions sets certain basic requirements for the receiver experiment. Steam must be generated at the same pressure and temperature. The structural arrangement must be similar to that of the pilot plant receiver in critical zones, and the same materials of construction should be used. The distribution of absorbed heat flux over the heating surfaces must have the same pattern. Finally, the experimental receiver must be subjected to start-up and other transients typical of those predicted for the pilot plant.

The basic test approach will be to operate the receiver, using electrical radiant heaters as a heat source, through various modes simulating anticipated operation of the pilot plant receiver to check thermal/hydraulic performance, structural design, and operation of controls. The operational modes will simulate:

- Steady-state operation at the pilot plant receiver design point and at reduced flow rates
- Steady-state operation at the pilot plant receiver peak load
- Pilot plant receiver start-up and other operating transients.

2. Conceptual Design and Related Analyses

a. Scaling Criteria - The scaling criteria selected as most important in relating the experimental receiver to the pilot plant receiver are:

- Heat flux densities should be the same at corresponding points.
- The diameter and thickness of the steam drum should be the same.

- Tube diameters and thicknesses should be the same at corresponding points.
- The same tube support scheme and spans between supports should be used in both receivers.
- The loading on the drum internals (steam purification equipment) should be the same.
- Circulation of steam/water in the generating tubes should be identical in corresponding circuits.
- Mass velocity of steam should be the same at corresponding points in the superheater.

These criteria are ideal, and it would be desirable to meet them all exactly. However, it is impossible to do this in practice. In designing the experimental receiver, an attempt has been made to minimize the compromises that must be made in attempting to satisfy these ideal scaling criteria.

Identical heat flux densities are maintained at corresponding points so that no change need be made in heating surface arrangement when scaling down. Doing so requires that the internal dimensions of the experimental receiver cavity be obtained as follows:

$$\frac{\text{Experimental Receiver Dimension}}{\text{Pilot Plant Receiver Dimension}} = \sqrt{\frac{5 \text{ MW}}{39.613 \text{ MW}}} = 0.3553,$$

where 5 MW and 39.613 MW are the total heat fluxes incident on the surfaces of the experimental and pilot plant receivers respectively.

b. Mechanical Arrangement - The arrangement of the experimental receiver is geometrically and structurally similar to the pilot plant receiver in most respects. However, some dimensions of the pilot plant receiver, such as the diameters and thicknesses of the drum and tubes, are preserved in the experimental receiver according to the scaling criteria.

The boiler circuitry arrangement resembles many features of the pilot receiver boiler circuits. Differences, related to natural circulation requirements, are the number and size of downcomers, feeders, and risers; the unheated length of the waterwall panels; and the elevation of the drum.

To approximate the same steam mass velocity as in the pilot plant receiver, 16 passes, rather than 6, are used in the superheater. These 16 passes are arranged as 6 panels on the rear and side walls, thus resembling the 6 superheater panels of the pilot plant receiver. The superheater tubes span the complete width of the heated walls, which makes the spans nearly equal to those in the pilot plant receiver. The two lower passes across the lower right side wall of the experimental receiver have an expansion loop and intermediate support at the midpoint of the wall to check the functioning of the expansion loop and to compare the effectiveness of different span lengths. Since there is some concern that superheater tubes on 28.6-mm (1-1/8 in.) centers may rub against each other and wear excessively at some points because of the thermal cycling, tubes in the lower left side wall passes are installed on 31.8-mm (1-1/4 in.) centers in the experimental receiver.

Other special features to be incorporated in the experimental receiver are supports and cable access ports for the electrical radiant heaters. An insulated cavity door (not shown in the figures) will be provided for the aperture, which is 2.67-m (8-ft 9-in.) square.

The experimental receiver uses the same control concept for the feedwater flow and superheater steam temperature as does the pilot plant receiver. In an effort to cut costs of the experimental receiver, only the first-stage attemperator is installed for the ground test using electric heaters. In testing, the lack of the second-stage attemperator can be compensated for by stretching the start-up time, if necessary.

c. Thermal/Hydraulic Analysis - The methods and procedures used in the thermal/hydraulic design and analyses of the experimental receiver are the same as described previously for the pilot plant receiver. The rated heat flux condition was used as the basis for calculating the steady-state tube-wall and steam bulk temperatures. The superheater pressure drop, not including attemperators, was calculated as 896 kPa (130 psi) at the rated steam flow of 6559 kg/h (14,460 lb/h) and 1482 kPa (215 psi) at the maximum steam flow of 8760 kg/h (19,312 lb/h). The boiler circulation analysis was also performed at the rated heat flux condition. The results showed that the circulation characteristics of the two boilers were quite different because of the scaling criteria. The exit steam qualities predicted for the experimental receiver were much lower than for the pilot plant receiver. Therefore, biasing of heat-flux distribution to obtain zones of high heat flux during receiver tests is required so that the extreme steam quality vs. heat-flux-intensity condition expected for the pilot receiver can be simulated and tested in the experimental receiver. The outer panel of the left hand side boiler wall was chosen and designed for the purpose of this biased heat-flux test.

d. Structural Analysis - The structural design and analysis of the experimental receiver are similar to those of the pilot plant receiver. Since the diameters and thicknesses of the superheater and boiler tubes of the two receivers are identical, the same finite element tube models used in calculating the temperature distribution and thermal stresses for the pilot plant receiver were employed for the experimental receiver. The results indicated that the maximum tube-wall temperatures and thermal stresses of the two receivers were comparable.

3. Test Plan

The test plan for the 5 MWt experimental receiver has been issued. The content of this test plan is outlined below.

a. Test Objectives and Summary - This section describes the test objectives and summarizes the approaches to achieve these objectives.

b. Test System and Facility - The test system involves three subsystems, i.e., the receiver, the support equipment, and the test facility. Foster Wheeler will fabricate and erect the receiver and furnish the associated instrumentation and controls needed to operate the unit and to monitor its performance. Martin Marietta will supply the support equipment, which includes the steam pressure control valve, feedwater pump, feedwater heater, steam desuperheater, condensate subcooler, and main test control console. Sandia Laboratories will supply the test facility site and accommodations, which include buildings and support structure, radiant electrical heaters with associated controls, condenser, water and water treatment, and data acquisition and logging facilities.

c. Description of the Receiver - A detailed description of the 5 MWt experimental receiver is given in this section.

d. Receiver Operation, Controls, and Safety - The receiver will be operated according to the Receiver Operating Instructions, a detailed manual to be issued by Foster Wheeler. These instructions will specify the initial system checks and necessary pretest service requirements to make the unit operational and will describe the steps to be taken during start-up and shutdown of the unit.

The receiver will be controlled from a single console located in a control room. To achieve the correct operating mode, adjustments to the control functions will be based on the real-time data display of the key parameters on the control console. Other selected parameters displayed on a television screen on a close-to-real-time basis will be referenced in making those control adjustments.

The receiver has been designed to the ASME Boiler Code and other standards to ensure reliable and safe operation. After erection and prior to operation, a careful inspection will be made to assure that all components are properly assembled and conform to the design details. Before each test a check will be performed by the operator in accordance with a safety precaution list provided in the Receiver Operating Instructions. Limits will be established in the Test Procedures document on the critical parameters such as the drum pressure and the tube-wall temperatures. If these values are exceeded during the test, an orderly shutdown procedure will be followed.

e. Instrumentation - This section describes the instruments to be used in testing the experimental receiver and the sensing locations on the receiver. The instrumentation includes the measurements of temperature, pressure, flow, heat flux, deflection, residual strain, and strain. The temperature readings will be sensed using chromel-alumel thermocouples. Pressures will be sensed using pressure-to-electrical signal transducers. The feedwater and spray flow rates will be measured by orifice plates; the exit steam flow rate, by a flow nozzle. Flow meters for downcomers and selected feeders will be of two types: venturi and Ravese pitot tube. Two kinds of radiant heat flux meters, either independently water cooled or directly cooled by contact with the waterwall, will be used to sense the incident heat flux on the boiler walls. The deflections of superheater tubes will be measured by LVDT's (Linear Variable Differential Transformers). The residual strain measurements will be made by an Image Transfer Technique developed at Foster Wheeler. Weldable high-temperature, resistance strain gages will be used to measure the strains in the superheater and boiler tubes. An instrumentation list and figures showing the key instrumentation locations were previously presented in Section V.B.2 of Quarterly Report No. 2.

f. Test Description - The tests proposed in this Test Plan are classified and defined as follows:

- Low temperature and pressure trial runs to check out overall system operating characteristics by running the receiver at reduced pressure, temperature, and incident heat energy
- High temperature and pressure trial runs to obtain a full-load operating condition by starting the receiver from the ambient condition and stepping up the heat input in small increments
- Cold start-up to demonstrate the receiver cold start-up performance by a sequence of progressively shorter periods of start-up

- Hot start-up to demonstrate the receiver hot start-up performance by a sequence of progressively faster rates of increase of drum pressure
- Restart after cloud passage to simulate a cloud's obscuring the incident radiation and then a restart reaching the peak heat flux within a minimum time span
- Steady-state and reduced-flow runs to demonstrate the receiver operating characteristics at 100, 75, 50, and 25 percent of the rated flow rates at the full pressure and temperature
- Biased heat flux superheater test to simulate the peak heat flux condition locally on the superheater to obtain the temperature, stress, and deflection data at this condition
- Biased heat flux boiler test to simulate the extreme boiler circulation characteristic of the pilot plant receiver by exposing one particular boiler panel to the desired higher heat input.

g. Data Acquisition and Logging - The data acquisition and logging system will be programmed and operated by Sandia personnel. Signals from all the sensing locations will be scanned by multiplexers and transmitted to a central data processing system, where a computer will be used to process, store, and print out the test data. The signals required for control functions and real-time visual display will be directly wired from the receiver to the control panel. Provision is also being made to extract certain important test data from the computer on a close-to-real-time basis. These data can be displayed on a television screen and printed out, if desired, with a hard-copy unit.

h. Test Personnel and Scheduling - Foster Wheeler will provide two engineers and one technician to supervise the operation and testing of the receiver. The control of the test conditions will be originated by Foster Wheeler personnel as detailed written instructions to the other parties. The testing period will encompass an eight week period.

4. Test Procedures

A preliminary draft of the Test Procedures for the 5 MWt Experimental Receiver was completed. The purpose of these procedures is to formulate a set of pretest system checkouts and define the specific tests to be performed, i.e., to cover detailed receiver operating procedures to be followed to achieve the test objectives and operate the receiver safely. The content of this document covers the following areas:

a. Pretest Requirements - This section describes the necessary system checkouts to be made after the receiver has been erected and the control equipment and instrumentation have been installed.

b. Start-up and Shutdown Procedures - This section explains, in general, how the receiver will be filled with water, started up from an ambient condition to full load, shut down, and restarted at an elevated temperature and pressure. Also included is a set of conditions which warrant an emergency shutdown of the receiver along with the steps needed to achieve such a shutdown.

c. Data Acquisition - This section explains which readings will be recorded and how often they will be recorded. The locations of the data sensing points are also included along with any necessary signal conditioning.

d. Boiler Water Quality Control - In this section the limits of impurities in the boiler water are given along with the steps to be taken, if necessary, to reduce the impurities to the acceptable levels.

e. Description of the Scheduled Test Runs - The operating conditions, the control functions, and the procedures needed to conduct the scheduled test runs are described in this section. A description of each test is contained in a table which lists the adjustments that must be made to the receiver's controls and valving as a function of time. Step-by-step procedures were also developed for each test based on this table. As an example, a summary control table for a typical cold start-up test is shown in Table V-1. This table was derived from the analytical results obtained using the receiver transient analysis computer program. The predicted operating conditions of this sample test run are shown in Figure V-7.

This summary table serves only as a reference to follow. Each line represents changes in the receiver control settings as a function of time. The increase in settings between lines will be accomplished stepwise at the convenience of the operators.

5. Detailed Design, Procurement, and Fabrication

The detailed design of the experimental receiver began in October 1975 with work on the development of general arrangement drawings. These drawings, shown in Figures V-8, V-9, and V-10, were essentially complete, except for minor revisions, at the end of March 1976. Work on the design of the pressure parts and internal structural members began in November 1975, followed by design of structural supporting steel and the cavity door in January 1976. The receiver detailed design was completed early in June 1976, although some shop detailing of the cavity door and enclosure continued into August.

Figure V-7 Predicted Operating Parameters of the Experimental Receiver for a Scheduled Cold Startup Test Run.

Figure V-8 Side Elevation View - Experimental Receiver

(From DWG. No. 59-6026-5-21 Rev. D)

Figure V-9 General Arrangement of the Experimental Receiver - Superheater Elevation Sections and Details

(From Dwg. No. 59-6026-5-22 Rev. G)

Figure V-10 General Arrangement of the Experimental Receiver - Sectional and Plan Views

Procurement began in September 1975 when orders were placed for the safety valves, which were long-lead items, for delivery in September 1976. Steam drum materials were ordered in December 1975 to enable shop fabrication of the drum to begin in April 1976. Pressure parts and auxiliary equipment were ordered during January, February, and March of 1976. Structural steel, grating, and railing were ordered during April. The initial order for the feedwater and steam temperature control systems was also placed during April, but because of changes and modifications, the final order was not released until June. The following items remain to be delivered as of the date of this report:

- Superheater stop valve
- Remote level indicator
- Miscellaneous valves
- Superheater safety valve
- Steam temperature control
- Feedwater regulator

Shop fabrication covered the period from mid-April 1976 until the end of September, with most of the work concentrated during July and August. Listed below are the starting and shipping dates for major subassemblies of the receiver.

Began fabrication:

- Steam drum - April 12
- Structural steel - May 24
- Waterwall panels, circulation system piping - June 21
- Superheater - July 6
- Cavity door and enclosures - August 30

Shipped:

- Structural steel - June 21
- Steam drum - July 13
- Circulation system piping - July 30
- Waterwall panels and most of superheater - August 14
- Balance of superheater plus buckstays - August 27
- Cavity door and enclosures - To be shipped October 1

6. Field Assembly and Erection

Foundations for the experimental receiver were poured at the Sandia Radiant Heat Facility in Albuquerque in mid-June, and the supporting steel for the receiver was erected on these foundations during the first week of July. The steam drum arrived on site on July 19 and was immediately raised into position and secured. The waterwall panels

were lifted into position after black radiation-absorbing paint had been applied to their heat-absorbing surfaces. Figures V-11 and V-12 show the left side wall and rear wall panels being brought into their proper sloped positions. The outside surfaces of the superheater tubes were also painted black prior to assembly. By the end of this reporting period, all pressure parts (drum, waterwalls, superheater panels, downcomers, feeders, risers, and superheater transfer lines) were in place and about 95 percent welded. Erection was proceeding on schedule. An erection schedule is given in Figure V-13.

Figure V-11 General View of Experimental Receiver During Erection.

Figure V-12 Positioning of Waterwall Panels During Erection.

Figure V-13 Erection Schedule for the Experimental Receiver.

CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM (PHASE 1)

Annual Progress Report Number 1

SECTION VI

THERMAL STORAGE SUBSYSTEM

Authors:

Steve H. Bomar, Jr.

Ralph F. Altman

Charles A. Murphy

John E. Myers (MMC)

**Engineering Experiment Station
Georgia Institute of Technology
Atlanta, Georgia 30332**

VI. THERMAL STORAGE SUBSYSTEM

A. INTRODUCTION

The Martin Marietta Central Receiver Solar Thermal Power System team is structured so that program responsibilities are divided among the members of the team, under the overall direction of the Martin Marietta Corporation. The Thermal Storage Subsystem is the responsibility of the Engineering Experiment Station of the Georgia Institute of Technology. The major activities in this area of the program include development of a preliminary design for a 10 MWe Pilot Plant Thermal Storage Subsystem and the design, construction and testing of a Thermal Storage Subsystem Research Experiment; various other tasks such as cost estimating and engineering support are also underway.

1. Program Objectives

The objective of the program presently under contract is to complete a preliminary design of a solar thermal electric power generation system which will have the ability to feed electricity into a commercial power distribution grid. If this design, or parts thereof, are selected for further development, subsequent contracts would cover detailed design and construction of the 10 MWe Pilot Plant and a test program to permit acquisition of operating experience and demonstration of feasibility.

In its request for proposal, ERDA specified that the Pilot Plant preliminary design would incorporate thermal storage and that its capacity would be sized to maintain the electrical output of the plant at 70 percent of its nominal value for six hours. In its proposal to ERDA, the Martin Marietta team selected a storage system design concept based on storage of sensible heat in liquid media. The reasons for this choice were: (1) storage in liquid media has a high probability of technical success because much of the necessary technology exists, and (2) the possibility of program delays caused by technical difficulties would be reduced by selecting this approach because thermal storage was the least well-defined of the plant subsystems and unexpected delays in this area could impact the schedule of the entire program.

The Thermal Storage Subsystem is an integral part of the solar thermal Pilot Plant. It must supply steam to the turbine in the Electrical Power Generation Subsystem at temperatures and pressures acceptable to the turbine and it must accept steam from the Receiver Subsystem at the receiver's output conditions. Its controls must be integrated with the remainder of the plant and it must serve as a buffering device to smooth steam flow rates in some operating modes.

The specific objectives of the Georgia Tech portion of the Martin Marietta Pilot Plant program are: (1) to analyze the requirements of the Pilot Plant Thermal Storage Subsystem and develop a preliminary design, (2) to develop a conceptual design of the Thermal Storage Subsystem Research Experiment based on the Pilot Plant preliminary design, (3) to develop a detailed design for the Thermal Storage

Subsystem Research Experiment, (4) to build and test the Thermal Storage Subsystem Research Experiment, and (5) to revise the preliminary design of the Pilot Plant Thermal Storage Subsystem using the results of the Research Experiment activity. Other subsidiary objectives such as program management, participation in coordination and design review activities, preparation of technical reports, etc. are included within the technical objectives.

2. Summary of Progress Through Reporting Date

This report describes the technical progress during the first 15 months of the program. Details of the program accomplishments are described in the following sections of the report, but a broad summary is appropriate at this point to illustrate major milestones:

a. Pilot Plant Preliminary Design

- o Subsystem analyses have been conducted to define Thermal Storage Subsystem requirements and assure compatibility with other subsystems.
- o Alternate thermal storage approaches have been evaluated to determine whether a change in concept should be recommended.
- o An analytical performance model based on the Martin Interactive Thermal Analysis System (MITAS) heat transfer computer program has been developed.
- o Heat exchangers, piping, tanks and other process equipment have been sized and preliminary detailed schematics and ground layout drawings have been prepared.
- o Users of candidate storage fluids have been visited, a heat transfer salt has been selected as the higher temperature fluid and a hydrocarbon oil has been selected as the lower temperature fluid.
- o Subsystem specifications have been updated and preliminary reliability allocations established.
- o A Preliminary Design Baseline Review has been conducted and ERDA approval received.
- o A preliminary cost estimate for the 10 MWe Pilot Plant has been prepared and submitted.

b. Thermal Storage Subsystem Research Experiment

- o A Research Experiment has been defined based on the Pilot Plant preliminary design baseline; the charging and discharging rates are 2 MWth and the storage capacity is 1.6 MW-hr.
- o Arrangements have been made for construction of the Research Experiment at Plant Yates of the Georgia Power Company, near Newnan, Georgia; the Georgia Power Company will supply superheated steam and feedwater for test operations.
- o An analytical performance model based on the MITAS computer program has been developed for the Research Experiment.
- o Conceptual and detailed designs for the Research Experiment have been completed; Design Reviews have been conducted and ERDA approval received.
- o Subcontracts and Purchase Orders have been issued for equipment and components of the Research Experiment; all materials have arrived except the storage media which are scheduled for delivery.
- o Subcontracts have been issued for construction and cleaning; construction is in progress.
- o A Test Plan for the Research Experiment has been prepared and approved.
- o Operating Procedures for the Research Experiment have been prepared; approvals and changes are in progress.

B. PILOT PLANT THERMAL STORAGE SUBSYSTEM

The preliminary baseline design of the Pilot Plant Thermal Storage Subsystem has been completed. As part of this effort, the storage concept was reviewed and finalized, major components were identified and sized and a ground layout of the system was completed. In addition, two basic operation modes were defined: charging of energy into the storage system and discharging of energy from the storage system.

In the charging mode, superheated steam from the solar central receiver will be passed through a heat exchanger where it will be cooled to a temperature near its saturation temperature. Simultaneously, a molten eutectic salt mixture, heat transfer salt (HTS), will be circulated through the heat exchanger countercurrently to the steam. This process raises the temperature of the heat transfer salt to approximately 728 K (850°F) and it is stored at that temperature for subsequent use during discharging. The steam continues to another heat exchanger where it is condensed to remove its latent heat. In this heat exchanger a hydrocarbon oil circulates countercurrently to the steam and is thereby

heated to approximately 567 K (560°F); the oil storage medium is also stored for subsequent use during discharging. Finally, the liquid water resulting from condensation of the steam passes to a third heat exchanger where additional heat is removed and stored in oil. During discharging of the storage system, all fluid flow directions are reversed and liquid water is converted to superheated steam. This approach to thermal energy storage results in a highly reliable system capable of meeting all of the requirements for Pilot Plant Thermal Storage.

1. Requirements for Pilot Plant Thermal Storage Subsystem

The Thermal Storage Subsystem has three basic requirements:

- o To store the thermal energy in any portion of the output from the Receiver Subsystem, from full steam flow to very low flow.
- o To provide steam to the Electrical Power Generation Subsystem (EPGS) for generation of at least 7 MWe net for a continuous period of up to 6 hours or lower rates for correspondingly longer times.
- o To provide steam with acceptable conditions for maintaining operation of the turbine during cloud transients and for protection of plant equipment during shutdown periods.

In order to satisfy the first requirement, the Thermal Storage Subsystem was designed to extract heat from steam at flow rates up to 61,236 kg/h (135,000 lb/h). Steam will enter the Storage Subsystem at 8825 kPa (1280 psig) and 784 K (950°F). To meet the second requirement, 4137 kPa (600 psig), 700 K (800°F) steam can be discharged from storage at a maximum rate of 57,380 kg/h (126,000 lb/h). The minimum design flow rate is 2268 kg/h (5000 lb/h) for both the charging and the discharging operations. The storage subsystem also contains equipment to generate sealing steam for the turbine and condenser in the EPGS when the turbine is operating in the standby mode. The sealing steam, which satisfies the second part of the third requirement above, will be delivered at 1034 kPa (150 psig), 459 K (366°F) at a rate of 454 kg/h (1000 lb/h). These flow rates and steam states are summarized in Table VI-1:

TABLE VI-1

PILOT PLANT THERMAL STORAGE STEAM FLOWS AND STATE POINTS

Parameter	Charging	Discharging
Inlet or Outlet Temperature	784 K (950° F)	700 K (800° F)
Inlet or Outlet Pressure	8,825 kPa (1250 psig)	4137 kPa (600 psig)
Max Flow Rate	61,236 kg/h (135,000 lb/h)	57,380 kg/h (126,000 lb/h)
Min Flow Rate	2268 kg/h (5000 lb/h)	2268 kg/h (5000 lb/h)

2. Thermal Storage Feasibility Studies

During the early stages of the Thermal Storage design, three approaches to thermal storage were considered. The details of this concept study were published in the First Quarterly Report for the Central Receiver Solar Thermal Power System, Phase One Program; a description of the concepts and summary of the conclusions drawn from the study will be presented here.

Three storage concepts are illustrated in Figure VI-1. In the simplest concept heat would be stored as sensible heat in a pumped liquid storage medium. The principle of operation of a liquid sensible heat system is depicted in the top diagram of Figure VI-1. When heat is being stored in the system (charging operation), liquid is pumped from a low temperature reservoir through a heat exchanger. The storage medium is heated in the heat exchanger by cooling or condensing steam or liquid condensate and flows into a high temperature reservoir. At some later time, heat can be extracted from the hot liquid (discharging operation) by passing it back through the same or a different heat exchanger where it heats or evaporates water or steam. The storage medium then flows back into the low temperature reservoir. This process of heating and cooling a liquid in a heat exchanger is a common industrial practice. All of the fluids that were considered for use in the Thermal Storage Subsystem have a long record of industrial usage in heat exchange application. Building a liquid sensible heat storage device is largely a matter of adapting existing technology to a new usage.

In the second approach studied, heat would be stored as sensible heat in a solid storage medium. The storage medium would be an inexpensive

Sensible Heat Fluid Storage System

Sensible Heat Solid Storage System

Encapsulated Salt Latent Heat Storage System

material, either rocks or scrap metal. The principle of operation of the solid sensible heat storage concept is independent of the storage medium and this principle is illustrated in the middle diagram of Figure VI-1. The solid storage medium, in this case rocks, is contained in a tank. A heat transport fluid, such as a hydrocarbon oil, is circulated in a loop through the tank and through a heat exchanger. When heat is being charged into the medium, the heat transport fluid will absorb heat in the heat exchanger and then flow into the top of the storage tank. As the fluid flows down through the tank, it gives up heat to the colder rocks. The fluid does not just flow through the tank one time. It is continuously circulated through the tank and back through the heat exchanger. As the charging process continues, a thermal gradient is established along the length of the tank. If charging continues long enough, the rocks at the top of the tank reach a temperature very close to the temperature of the incoming oil, and the oil leaving the bottom of the tank has a temperature close to the temperature of the cold rocks at the bottom of the bed.

In a well designed storage system of this kind, the thermal gradient will be relatively sharp and most of the solid medium will undergo a large temperature change as the gradient travels down the length of the tank. The volumetric efficiency for a sharp thermal gradient is high. If the thermal gradient is long compared to the length of the tank, a portion of the solid medium will undergo only a small temperature change, and the volumetric efficiency will be low. The size of the tanks and amount of storage medium increases under these conditions, and as a consequence the cost of the storage system goes up. The character of the thermal gradient depends on the thermal parameters of the heat transfer fluid and the solid medium, as well as the size, shape and packing of the solid medium. MITAS, the thermal analysis program described elsewhere in this report, has been used to model a stone bed. The results indicate that an economical stone storage system can be built if relatively small stones are used.

To extract heat from a solid-filled heat storage tank, the flow of the heat transfer fluid is reversed. The cold fluid flows in the bottom of the tank, is heated as it passes up through the solid storage medium and then flows back through the heat exchanger where it gives up heat. During this process, the movement of the thermal gradient is also reversed. The gradient now moves from the bottom of the tank towards the top.

A latent heat thermal storage system was the third concept investigated at the beginning of the program. The system would use an inorganic salt encapsulated in thin wall tubes. These tubes would be placed in a tank in bundles. A heat transfer fluid would circulate through the tank to either melt the salt in the tubes by giving up heat to it or solidify the salt by absorbing heat from it. The dead space in the tubes would be filled with an inert gas such as nitrogen or helium to enhance heat

conduction from the tube wall to the solid mass at the beginning of the melting process. The latent heat concept is diagrammed in the bottom illustration of Figure VI-1.

The operation of the system is similar in many ways to the operation of the solid sensible heat system just described. When heat is being stored in the system, hot oil would flow into the top of the tank and down through the tube bundle giving up heat to the salt in the tubes. The thin tube wall would have little thermal resistance, and the tube diameter would be small so that the heat conduction path into the salt would be short. These two design considerations would allow high charge and discharge rates to be maintained throughout the entire melting and solidification processes. A thermal gradient and melting gradient would form in the tank and move down the axis of the tank as charging continued. During discharge, the oil flow through the tank would be reversed. The direction of the thermal/melting gradient would also reverse. Again, the thermal and melting gradients should be sharp to minimize the size of the storage system.

At the end of the storage concept study, it was concluded that a storage system based on any of the three concepts was feasible. The last two systems, the solid sensible heat and latent heat system, were eliminated from further consideration because it was concluded that the development effort required to produce a highly reliable storage system based on either of these concepts was inappropriate for this program. Table VI-2 contains a summary of problems that would have to be understood before a reliable plant with a thirty-year lifetime could be proposed with confidence. Some of the problems listed in Table VI-2 would require long-term studies in order to document reliability and lifetime of the proposed solutions.

On the other hand, a Thermal Storage System based on the liquid sensible heat principle that is reliable and has a predictable lifetime requires only a modest development effort. Low cost heat transfer liquids with a long record of industrial usage in heat exchange applications are readily available. Building a liquid sensible heat storage device is largely a matter of adapting existing technology to a new usage. For these reasons, the Baseline Design incorporates a Thermal Storage Subsystem based on the liquid sensible heat approach.

3. Pilot Plant Thermal Storage Subsystem Baseline Design

A conceptual schematic for the Pilot Plant Thermal Storage Subsystem Baseline Design is shown in Figure VI-2. The design is based on a two-stage liquid sensible heat concept. The high temperature stage employs a molten salt for the storage medium. A hydrocarbon oil is used as the storage medium in the low temperature stage.

TABLE VI-2

PROBLEM AREAS IN SOLID SENSIBLE HEAT AND LATENT HEAT SALT THERMAL STORAGE SYSTEMS

-
-
- (1) The solid sensible heat system requires that oil be pumped through the tank. The tanks are large and must operate at low pressure to minimize their cost. This system therefore requires that the oil be pumped into and out of the tank with sufficient controls to ensure that the tanks are not overpressurized.
 - (2) The solid media will have a tendency to settle in the tanks. The settling will produce an effective pressure on the tank walls and the thickness of the tank side must be increased in order to accommodate this effective pressure.
 - (3) If rocks are used, they will be a self-generating contamination source. Filters and settling tanks may be required to protect heat exchanger and flow control equipment.
 - (4) The long term chemical compatibility of oil and rocks at high temperature is unknown, and the chemical composition of the rocks will vary depending on the source of the stone.
 - (5) Flow may tend to channelize in the solid sensible heat system. If channelization occurs, baffles that will increase the cost of the system will be required.
 - (6) The long-term chemical stability of salts at high temperatures is not well known. As most salts degrade, they tend to evolve gas which can cause pressure build up in the tubes. A salt composition change may shift the melting point or degrade the heat of fusion.
 - (7) The long term compatibility of the salt and the tube metal at high temperatures must be understood.
 - (8) As the salt thaws and freezes its volume will change putting some stress on the tubes. The tubes must be designed to stand this cyclic stress.
 - (9) A manufacturing process would need to be developed to encapsulate the large volume of salt into tubes economically.
-
-

Figure VI-2. Two-Stage Thermal Storage Subsystem Conceptual Schematic.

Figure VI-3. Heat Balance for Two-Stage Thermal Storage Subsystem.

During charging, the system operates as follows: Superheated steam at 734 K (950°F), 9925 kPa (1280 psia) enters the tube side of the molten salt heat exchanger where it is partially desuperheated by heating the salt flowing in the shell side. The salt flow is adjusted to maintain a constant steam outlet temperature. The steam then enters a spray desuperheater where its temperature is further reduced by mixing with and evaporation of feedwater. The feedwater flow rate is regulated to maintain a constant steam temperature of 600 K (620°F) into the vertical, hydrocarbon oil heat exchanger is protected from excessive film temperatures that would result in a high rate of thermal degradation. In the vertical heat exchanger, the steam is condensed in the tubes and heats the oil. The oil flow rate is controlled so that a constant pressure is maintained in the top head of the heat exchanger. The condensate collects in the bottom head of the vertical heat exchanger and then flows into a third exchanger where it is subcooled by the oil. The subcooling heat exchanger is sized so that the maximum temperature of the condensate exiting from thermal storage will be 539 K (510°F). Flow of condensate out of thermal storage is adjusted to maintain a constant liquid level in the condensing heat exchanger.

During discharge of thermal storage, the water, steam and storage media flows are reversed. The same heat exchangers are used during both the charging and discharging operations. In this mode of operation, feedwater enters the horizontal oil heat exchanger and is preheated by the oil to just below the boiling point. The inlet feedwater temperature into thermal storage will vary as a function of the power output level of Electric Power Generation Subsystem, but a constant feedwater temperature into the vertical oil heat exchanger is maintained by regulating oil flow through the preheater with a bypass valve and a throttle valve in the preheater oil supply line. The feedwater is evaporated in the tubes of the vertical heat exchanger by hot oil in the shell. The oil flow rate, which determines the rate of evaporation, is adjusted to maintain a constant turbine inlet pressure. Feedwater will be admitted into thermal storage to maintain a constant water level in the top head of the evaporator. Steam generated in the evaporator passes through a dryer in the top of the head and then into the salt heat exchanger where it is superheated. The salt flow is adjusted to maintain a constant steam outlet temperature of 700 K (800°F).

Some of the control circuits are analogous to circuits used in the operation of conventional power plants. During charging, the condensing steam pressure is controlled by the oil flow rate; analogous to the operation of a turbine condenser in which the cooling water flow rate is regulated to maintain a constant turbine back-pressure. During discharging, the outlet pressure from thermal storage is controlled by oil flow rate. This control mechanism is equivalent to adjusting the fuel burning rate to control outlet pressure in a conventional boiler. A three-element feedwater regulator which is used in conventional boilers to control the water level will be used in the Thermal Storage Subsystem during discharge for the same purpose.

Figure VI-3 shows a heat balance for the two-stage thermal storage subsystem. The uppermost line represents the temperature potential of the charging steam as a function of the energy removed from it; likewise the lowest line represents the temperature of the steam discharged as a function of the energy added. The temperatures along these lines are established by the physical properties of water and the pressures selected for system operation. If the pressures are fixed, the Thermal Storage Subsystem must extract energy from steam represented by the upper line and supply energy to steam represented by the lower line. This diagram therefore defines the temperature limits that are imposed on the heat storage media by heat transfer and thermodynamic considerations.

The diagram resembles a temperature vs. enthalpy diagram. In fact, the graph for the discharging steam and the oil and molten salt lines do represent temperature-enthalpy curves. The charging steam curve has been slightly modified to account for the fact that the enthalpy change per kilogram of steam is slightly different during charging and discharging. To obtain the correct enthalpy change per kilogram of charging steam, it is necessary to divide the enthalpy change read from the graph by the ratio of kilograms of charging steam required for each kilogram of discharge steam. Since the ratio is 1.034 and very close to 1.0, it is safe to neglect the difference unless a very accurate heat balance is required.

The detailed schematic of the Thermal Storage Subsystem, shown in Figure VI-4, illustrates the scope of the equipment required more accurately than the conceptual schematic given earlier. Each of the stages incorporates multiple tanks, with one empty tank. The fluid in a full tank is pumped through a heat exchanger into the empty tank. The tank that has just been emptied then serves as the receiving reservoir for the next full tank. This system minimized the empty tank capacity in each stage, but required a flexible manifold and valve arrangement. A nitrogen blanket is incorporated in each stage in order to extend the life of the thermal storage fluids.

The preliminary layout in Figure VI-5 of the Pilot Plant Thermal Storage Subsystem includes all major components properly sized and all piping and valves that are shown on the detailed schematic. It has been partially optimized with respect to placement of equipment, although it is recognized that further optimization can be accomplished.

The molten salt and hydrocarbon storage media have been segregated to comply with safety codes and the energy transfer modules, including heat exchangers, pumps and associated equipment, have been grouped separately from the tanks. The land area required for this design is a rectangular plot approximately 275 by 250 feet (2.2 acres). The codes that apply to the Pilot Plant and Research Experiment Storage Systems are listed in Table VI-3. These codes affect not only the general layout of the storage systems but the details of design as well. The extent of the coverage is indicated in the table. To date no deviations from these codes are foreseen in either the Pilot Plant or Research Experiment.

Figure VI-5. Preliminary Ground Layout for Two-Stage Pilot Plant Thermal Storage Subsystem.

TABLE VI-3

SUMMARY OF DOCUMENTS AND DESIGN COVERAGE AREAS APPLICABLE TO THE DESIGN OF THE PILOT PLANT
AND RESEARCH EXPERIMENT THERMAL STORAGE SUBSYSTEMS

Document	Coverage
(1) American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (1974)	
SECTION II Material Specifications	Material specifications for materials used in the construction of the pressure vessels and storage tanks and all materials used as welding rods, electrodes and filler materials
SECTION V Nondestructive Examination	Nondestructive examination methods specified and utilized on all equipment and components
SECTION VIII Pressure Vessels - Division I	Heat exchangers and storage tanks
SECTION IX Welding & Brazing Qualifications	Welding and brazing requirements
(2) American National Standards Institute	
ANSI A58.1-1972 Building Code Requirements for Minimum Design Loads in Buildings and other Structures	Wind and earthquake loading
ANSI B31.1-1971 Power Piping Code	Steam and condensate/water piping. Molten salt and petroleum oil piping (all loading designs).
ANSI B31.3-1973 Petroleum Refinery Piping	Molten salt and petroleum oil piping (construction only)
(3) National Fire Protection Assn.	
Volume 2 (15-73) Water Spray Fixed Systems	Fire protection systems
Volume 3 (30-73) Flammable & Combustible Liquids Code	All elements of the thermal storage tankage design and layout not covered by ASME and ANSI
Volume 5 (69-73) Explosion Prevention Systems	Storage tankage explosion prevention system (inerting)
Volume 6 (70-75) National Electrical Code	All electrical circuits
(4) Standards of American Institute of Steel Construction and American Concrete Inst.	All structural steel design not provided for otherwise above and all structural concrete
(5) Department of Labor, Occupational Safety and Health Standards, FR-39-125 (June 27, 1974)	All aspects of personnel safety and health as applicable
(6) Uniform Building Code	All designs not provided for otherwise above
(7) Interstate Commerce Commission Shipping Standards and Regulations	As applicable

Three tanks for molten salt have been provided, any one of which is capable of holding the entire salt inventory; the rationale for this selection was to allow two working tanks and one spare which would be used if repairs were necessary. Three large tanks for hydrocarbon oil are shown, two of which are needed to hold the oil inventory; because of the large volume of fluid required in the condensing/evaporating stage, the tanks must be used to contain both hot and cool oil at different times, depending on the level of system charge. The nitrogen cover gas systems are incorporated in the ground layout.

4. Storage Media Selection

The storage media used in the Thermal Storage Subsystem had to have thermal properties that were compatible with the requirements for storage already outlined. In selecting the storage fluids, the following criteria were considered:

- o The operating temperature range of the media,
- o The heat capacity or specific heat,
- o The cost effectiveness of the material as a heat storage medium,
- o Hazards associated with handling and use, and
- o Performance of the material in industrial applications.

Hydrocarbon oils proved to be the most attractive candidates for the low temperature stage storage medium. Both synthetic oils and heat transfer oil obtained from the distillation of crude oil were considered.

Table VI-4 contains a partial listing of the oils that were investigated along with some of their pertinent properties. The narrow-cut distillation fraction oils appear to be more cost effective than the synthetic oils. However, the synthetic oils are more stable chemically. It is presently anticipated that one of the cheaper oils will be used in the Pilot Plant, but the final decision depends on the results of the oil studies that are now being conducted. A detailed description of this study is found in a later section of this chapter.

An inorganic salt was chosen as the storage medium for the high temperature stage. The salt is a eutectic mixture of NaNO_2 , NaNO_3 and KNO_3 with a melting point of 415 K (288°F). The salt is variously called HITEC, Dupont's tradename for the mixture, and Heat Transfer Salt (HTS), the generic name for the same salt mixture. Heat Transfer Salt has a density of 1746 kg/m³ (109 lb/ft³) and a specific heat of 1562 J/kg K (0.373 Btu/lb-°F). The cost of the salt ranges from 84¢/kg (38¢/lb) to 46¢/kg (21¢/lb). The lowest cost results from buying the individual chemical components and mixing the salt at the plant site. HTS has been used in industrial applications since the 1930's, and its physical and chemical properties are well documented. The salt has been operated at temperatures of up to 727 K (850°F) in low carbon steel systems and up to 810 K (1000°F) in stainless steel process equipment.

TABLE VI-4

ROOM TEMPERATURE PROPERTIES AND PRICE FOR BULK
QUANTITIES OF HYDROCARBON OIL

<u>Oil Trade Name</u>	<u>Density</u>	<u>Capacity</u>	<u>Cost</u>	
	<u>kg/m³</u>	<u>J/kg K</u>	<u>Dollars/m³</u>	<u>Dollars/Gallon</u>
Amoco Oil 4199	880	1943	266.81	1.01
Gulf Security 53	878	1943	250.96	.95
Shell Thermia 33	865	1926	219.26	.83
EXXON Caloria HT43	855	2177	258.89	.98
Therminol 55	887	1884	396.26	1.50
Therminol 66	1001	1449	1505.77	5.70

5. Pilot Plant Performance Studies

A computer model of the Thermal Storage Subsystem was developed in order to:

- 1) Better understand the operating characteristics of the Thermal Storage Subsystem, and
- 2) Simplify scaling of Research Experiments results to Pilot Plant performance predictions.

The partial differential equations describing the transient flow of heat and fluids in the thermal storage subsystem were solved using finite difference routines contained in MITAS. MITAS (Martin Marietta Interactive Thermal Analysis System) is a multipurpose computing program.

Models were made of thermal storage for both the Pilot Plant and the Research Experiment. The graphs in Figures VI-6 and VI-7 are typical examples of results obtained from the MITAS simulations. The steam and feedwater outlet temperatures from the spray desuperheater and heat exchangers for a charging operation are shown in the figures; Figure VI-6 is for the Pilot Plant and Figure VI-7 is for the Research Experiment. In both simulations, a step increase in steam flow from zero to the nominal flow rate is imposed on the model. The temperatures in the desuperheater and subcooler behave as expected and after initial damped oscillation level off to a steady temperature. The temperature reaches steady state after about 10 minutes following a sudden startup. The outlet temperature from the condenser, in both models, displays a periodic oscillation but this oscillation has been traced to a deficiency in the analytical model.

C. THE THERMAL STORAGE SUBSYSTEM RESEARCH EXPERIMENT

The Research Experiment is designed to demonstrate the technical feasibility of the selected thermal storage concept. It is a miniature Pilot Plant Thermal Storage Subsystem in all important respects except flow rate and capacity, which will require scaling to the Pilot Plant levels. It matches the Pilot Plant with respect to steam and water state points, storage materials, operating temperatures and handling methods, control systems, and operating strategy.

1. Research Experiment Requirements

The Research Experiment is designed to meet the following requirements:

- o Accept superheated steam in the CHARGE MODE at 8619 kPa (1250 psig) and 783 K (950°F); produce superheated steam in the DISCHARGE MODE at 4137 kPa (600 psig) at 672 K (750°F).

Figure VI-6. Predicted Steam and Water Outlet Temperatures for Pilot Plant Thermal Storage Subsystem during Charging.

Figure VI-7. Predicted Steam and Water Outlet Temperatures for Research Experiment Thermal Storage Subsystem during Charging.

- o Function in the CHARGE AND DISCHARGE MODES at
 - A maximum rate of 2.0 MWth and a minimum rate of 0.4 MWth
 - Any intermediate rate between the maximum and minimum
 - Varying rates which simulate the anticipated characteristics of the Pilot Plant Receiver and Electric Power Generation Subsystems.
- o Demonstrate the suitability of commercially available control systems for control of the Thermal Storage Subsystem.
- o Store in the CHARGE MODE sufficient energy to allow 1.6 MWth (thermal) to be recovered in the steam supplied in the DISCHARGE MODE.
- o Provide performance data suitable for correlation to design predictions.
- o Establish design criteria and performance expectancies for the Pilot Plant Thermal Storage Subsystem.

2. Description of the Thermal Storage Subsystem Research Experiment

A detailed schematic of the Thermal Storage Subsystem Research Experiment is shown in Figure VI-8. It is analogous to the conceptual schematic shown in Figure VI-2, but includes control loops and additional piping details which were omitted from Figure VI-2; the heat balance diagram shown in Figure VI-3 applies also to the system defined by Figure VI-8.

During charging, steam enters the Research Experiment at 783 K (950°F) and 8619 kPa (1250 psig) and is cooled to 626 K (667°F) in the heat transfer salt stage (heat exchanger HE1 and associated hardware). The remaining superheat in the steam is removed in the desuperheater DS2 and the steam passes to the condensing stage (heat exchanger HE2) where it is condensed at 574 K (574°F). The condensate is passed to the sub-cooling stage (heat exchanger HE3) where it is cooled to 539 K (510°F) and discharged to silencer S1. Desuperheaters DS1 and DS2 are water injection units provided to allow the steam temperatures to be reduced in order to protect the storage media from overheating under certain flow conditions. During operation in the charging mode the molten salt storage medium is pumped from the cold reservoir (tank T2) at 577 K (580°F) through heat exchanger HE1 to the hot reservoir (tank T1) at 727 K (850°F); the hydrocarbon oil storage medium is pumped from the cold reservoir (tank T4) at 531 K (497°F) through heat exchangers HE3 and HE2 to the hot reservoir (tank T3) at 570 K (567°F).

During discharging, all flow directions are reversed and the pressure in the steam-water sections of the system is changed. Feedwater enters the Research Experiment at 505 K (450°F) and 4137 kPa (600 psig), is preheated in the heat exchanger HE3, evaporated in the heat exchanger HE2 at 530 K (495°F), and superheated in heat exchanger HE1 to an

LEGEND	BDV	BLOW DOWN VALVE	CP	CIRCULATION PUMP	CP3	CHARGING PRESSURE SENSOR	CT3	CHARGING TEMPERATURE SENSOR	EDV	EMERGENCY DRAIN VALVE	CWV	COLD WATER VALVE	DP3	DISCHARGING PRESSURE SENSOR	DST3	DESUPERHEATER TEMPERATURE SENSOR	DT3	DISCHARGING TEMPERATURE SENSOR	FV	FEEDWATER VALVE	HE	HEAT EXCHANGER	HT	HOLDING TANK	LP	LINE FILTER	FCV	FLOW CONTROL VALVE	FR3	FLOW RATE SENSOR	FRV	FLOW REVERSING VALVE	LLS	LIQUID LEVEL SENSOR	HST	NITROGEN BLANKET STORAGE TANK	PR	PRESSURE REGULATOR	PRD	PRESSURE RUPTURE DISC	PS	PRESSURE SENSOR	FVRD	PRESSURE/VACUUM RUPTURE DISC SILENCER	SDV	STEAM BYPASS VALVE	SCV	STEAM CUTOFF VALVE	SR	SAFETY RECEIVER	T	TANK	TEFVC	THREE ELEMENT FEEDWATER CONTROL	T3	TEMPERATURE SENSOR
--------	-----	-----------------	----	------------------	-----	--------------------------	-----	-----------------------------	-----	-----------------------	-----	------------------	-----	-----------------------------	------	----------------------------------	-----	--------------------------------	----	-----------------	----	----------------	----	--------------	----	-------------	-----	--------------------	-----	------------------	-----	----------------------	-----	---------------------	-----	-------------------------------	----	--------------------	-----	-----------------------	----	-----------------	------	---------------------------------------	-----	--------------------	-----	--------------------	----	-----------------	---	------	-------	---------------------------------	----	--------------------

Figure VI-8. Detailed Schematic of Thermal Storage Subsystem Research Experiment.

output temperature of 672 K (750°F). The desuperheaters DS1 and DS2 are not used in the discharge mode. A shunting loop around heat exchanger HE3 is provided for the hydrocarbon oil through FRV 15 in order to balance the heat input and prevent boiling in HE3. The specified discharging rate of 2.0 MWth corresponds to a steam flow rate of 3267 kg/hr (7187 lb/hr).

The Research Experiment operation will virtually duplicate the Pilot Plant operation except in flow rates and energy storage capacity. The exception is the use of feedwater at a temperature of 505 K (450°F) in the Research Experiment rather than the temperature of 477 K (500°F) anticipated in the Pilot Plant.

The Research Experiment is currently being constructed at a site owned by the Georgia Power Company at its Plant Yates, a coal-fired steam electric generating installation near Newnan, Georgia. This location was selected because superheated steam could be obtained at conditions matching the Pilot Plant receiver output state points, and the temperature levels throughout the Research Experiment could thus match very closely those expected in the Pilot Plant. The Georgia Power Company is supplying superheated steam, treated feedwater, other utilities and land; the program will pay for these services under a subcontract with the company.

A perspective drawing of the Thermal Storage Subsystem Research Experiment is shown in Figure VI-9 and a recent photograph of the experiment under construction is shown in Figure VI-10. The two smaller tanks are the molten salt storage reservoirs; the heat exchanger assembly mounted between these tanks is the desuperheating-superheating exchanger HE1. The large vertically-oriented heat exchanger is the condensing-evaporating exchanger HE2, and the small horizontally-oriented heat exchanger behind HE2 is the subcooling-preheating exchanger HE3. The two larger tanks in the background are the oil storage reservoirs and the control trailer can be seen on the left in Figure VI-10. A ground layout of the Research Experiment is shown in Figure VI-11, illustrating the placement of the major items of equipment.

Figure VI-12 shows the Research Experiment heat exchangers and pipe trestle on which the steam and water lines are supported. The superheated steam and treated feedwater are supplied from a nearby generating unit and are carried along the trestle to a point adjacent to the desuperheating-superheating heat exchanger.

The heat exchangers are shown in Figures VI-13 through VI-18 and described in Table VI-5. All heat exchangers are tube-in-shell designs and perform dual functions; heat is transferred to the storage fluids during charging and from the storage fluids during discharging. In order to provide the thermal driving force needed for system operation, the pressure on the steam-water side of the exchangers is 8619 kPa (1250 psig) during charging and 4137 kPa (600 psig) during discharging.

Figure VI-9. Perspective View of Thermal Storage Subsystem Research Experiment, without Supporting Structures.

Figure VI-10. View of Thermal Storage Subsystem Research Experiment under Construction.

Figure VI-11. Ground Layout of Thermal Storage Subsystem Research Experiment.

Figure VI-12. View of Pipe Trestle and Heat Exchangers in Thermal Storage Subsystem Research Experiment.

Figure VI-13. Desuperheating-Superheating Heat Exchanger HE1 Drawing.

Figure VI-14. Desuperheating-Superheating Heat Exchanger HE1.

Figure VI-15. Condensing-Evaporating Heat Exchanger HE2.

Figure VI-16. Condensing-Evaporating Heat Exchanger HE2 Drawing.

TABLE VI-5

HEAT EXCHANGER DESCRIPTION AND SPECIFICATIONS

Parameter	HE1	HE2	HE3
Description	Two U-tube, tube-in-shell bundles; mounted horizontally and operated in series	One fixed tube sheet, tube-in-shell bundle; mounted vertically and operated as a condenser during charging and as an evaporator during discharging	One U-tube, tube-in-shell bundle; mounted horizontally
Heat exchange area m ² (ft ²)	43.8 (471)	321.8 (3464)	33.2 (357)
Material	alloy steel	carbon steel	carbon steel
Shell diameter m (in.)	0.356 (14)	0.508 (20)	0.254 (10)
Shell length m (ft)	3.048 (10)	7.315 (24)	3.658 (12)
Number of tubes	60 per unit	301	30
Tube description	0.01905 m (0.75 in.) diameter; seamless	0.01905 m (0.75 in.) diameter; low fin, seamless	0.01905 m (0.75 in.) diameter; low-fin, seamless
Supplier	THERMXCHANGER, Inc., Oakland, California		

During charging, the water level in the condensing-evaporating exchanger HE2 is maintained near the bottom of the unit and steam condensation occurs; during discharging, this water level is maintained near the top of the unit and boiling occurs. The desuperheating-superheating heat exchanger HE1 was constructed as two series-operated units to accommodate the large changes in fluid temperatures between the respective inlets and outlets.

The storage media tanks are shown in Figure VI-19 through VI-22 and described in Table VI-6. The salt tanks are provided with submerged steam coils to permit melting of the heat transfer salt after it is placed in the tanks and to permit the salt to be held above its melting temperature during prolonged shutdown periods. The salt tanks in the Research Experiment were fabricated from carbon steel because their operating lifetimes are expected to be only a few months; stainless steel has tentatively been selected for use in a Pilot Plant.

The storage media pumps are described in Table VI-7. The oil pump is a conventional design and is mounted outside the tanks. However, the salt pumps are a submerged design with the electric motor mounted on a flange at the top of the tank and the pumping head submerged in the molten salt near the bottom of the tank. Submerged pumps are required for pumping the molten salt at the temperature levels in the Research Experiment because reliable shaft seals are not available. In a Pilot Plant Thermal Storage Subsystem the salt pumps would be a cantilevered design without submerged bearings or seals. In this case, the pumps might be installed in sumps rather than in the tanks themselves because tank heights would require prohibitively long cantilevered pump shafts.

Figure VI-23 shows a plan view of the oil piping system to illustrate the scope of the piping designs. After the pipe lines were sized and components located, the piping design procedure called for tentative location of lines using hand-calculation routines, followed by computation of stresses using a commercial computer program known as PIPESD; the tentative designs were then iterated until stresses fell within allowable limits.

Figure VI-24 shows a drawing of the spray desuperheaters DS1 and DS2. These units spray feedwater into the superheated steam in order to reduce its temperature. The desuperheaters are unusual in that they must operate over a wide range of flow rates and must allow flow in both directions with acceptable pressure drops; they are used while steam is flowing in only one direction, during operation in the charging mode. The desuperheaters are mounted vertically with steam flowing up during desuperheating and the water is injected countercurrent to the direction of steam flow.

The Research Experiment control panel is shown in Figures VI-25 and VI-26. This equipment is mounted in the control trailer adjacent to the Research Experiment and provides process control and the capacity for simulation of flow rates from the Receiver Subsystem and to the Electric Power Generation Subsystem.

Figure VI-19. Drawing for Molten Salt Storage Tank.

Figure VI-20. Molten Salt Storage Tanks and Heat Exchanger HE1.

Figure VI-21. Hydrocarbon Oil Storage Tank.

Figure VI-22. Drawing of Oil Storage Tank.

TABLE VI-6
STORAGE TANK DESCRIPTION AND SPECIFICATIONS

Parameter	Salt Tanks	Oil Tanks
Description	Two cylindrical tanks; vertically mounted with submerged pumps placed near bottom of tanks; ASME flanged and dished heads	Two cylindrical tanks; vertically mounted; ASME flanged and dished heads
Liquid and nominal capacity kg (lbs)	Heat transfer salt; 6,636 (14,629)	Heat transfer oil (hydrocarbon oil); 59,380 (130,910)
Nominal and maximum operating temps. K (°F)	722 (840); 755 (900)	570 (567); 580 (585)
Material	carbon steel, welded construction	carbon steel, welded construction
Tank diameter m (ft)	1.83 (6)	3.66 (12)
Over-the-heads dimension m (ft)	2.06 (6.75)	9.30 (30.5)
Total tank volume m ³ (ft ³)	4.6 (162)	90.6 (3,200)
Supplier	J. J. Finnigan Industries, Inc., Duluth, Georgia	

TABLE VI-7
PUMP SPECIFICATIONS AND SUPPLIERS

Parameter	CP-1 & 2	CP-3
Storage Fluid Circulated	Heat transfer salt (molten)	Heat transfer oil (hydrocarbon oil)
Pump Type	Centrifugal, single stage	Centrifugal, single stage
Mounting	Vertical, tank mounted	Horizontal, base mounted
Casing Material	Stainless steel	Cast steel
Impeller Material	Stainless steel	Cast iron
Rated Drive Power, kW (hp)	3.73 (5)	22.37 (30)
Speed, rad/s (rpm)	183 (1750)	366 (3500)
Rated Capacity, liter/s (gpm)	1.14 (18)	25.23 (400)
Total Developed Head, m (ft)	21.34 (70)	76.20 (250)
Pump Size, Discharge x Suction mm (in.)	38.1 x 50.8 (1.5 x 2)	76.2 x 101.6 (3 x 4)
Supplier	Lawrence Pump and Engine Co. Lawrence, Massachusetts	Dean Brothers Pumps, Inc. Indianapolis, Indiana

3. Research Experiment Test Plan and Procedures

The Test Plan for the Research Experiment was published on June 28, 1976. The Test Plan describes the test objectives, the planned test sequence, the test site, instrumentation and controls, crew organization and safety requirements.

The objectives, which are grouped into primary and secondary categories in the Test Plan, can be summarized as follows:

Primary Objectives -

- 1) Demonstrate that the Research Experiment hardware is capable of charging and discharging thermal energy into and out of the liquid storage media at the desired system temperatures, pressures and rates.
- 2) Demonstrate that the Research Experiment controls can maintain the required operating parameters at different steam flow rates and during transients.
- 3) Demonstrate that the process equipment designs (heat exchangers, tanks and interconnecting piping) can tolerate anticipated thermal and pressure transients.
- 4) Develop and demonstrate procedures for Pilot Plant initial charging and operation.

Secondary Objectives -

- 1) Develop control and operating experience with a Thermal Storage Subsystem of the planned design.
- 2) Experimentally establish system heat losses and their effect on the storage system efficiency.
- 3) Identify design deficiencies and corrections to be incorporated into the Pilot Plant Preliminary Design.

The test sequence starts with a series of preliminary tests. These tests are primarily for the purpose of system checkout and trouble-shooting, but also include the initial loading and heating of the storage media. After successful completion of the preliminary tests, the performance test sequence will begin. A brief outline of the performance tests is:

- 1) Constant rate charging and discharging operations in which the entire Thermal Storage Subsystem Research Experiment is completely charged and discharged at 50, 100, and 20 percent of the design steam flow rates.

- 2) Transient charging and discharging operations in which the Research Experiment is run in modes which simulate steady changes in power level and power level changes which follow anticipated Pilot Plant conditions.

During the performance test sequence, operating mode reversals and repetition of runs will be accomplished in order to demonstrate the capability of the system to respond to the demands of other Pilot Plant subsystems.

To ensure that the test operations are safe and productive, detailed test procedures have been written. The procedures are grouped into nine function-related chapters as indicated in Table VI-8. It was intended that the procedures include every action that is required to accomplish the specified task and any cautions or unusual circumstances that operators should be aware of during execution of the procedure. The procedures will be updated during the tests to incorporate changes suggested by operating experience.

4. Construction of Research Experiment

All of the components required for construction of the Research Experiment have been ordered and received. The major components have been described in this chapter. Broyles and Broyles, Inc., Atlanta Division, was selected to build the Research Experiment on a competitive bid basis and began work the week of July 12, 1976. To date, all major components have been set in place and the oil piping system has been completed and hydrotested. The molten salt piping and steam and feedwater piping are about 50 percent complete, and the contractor is maintaining the original construction schedule.

At the end of the construction phase, the steam and feedwater piping will be chemically cleaned. The cleaning operation will be performed by Halliburton Services through their Huntsville, Alabama office. The operation will consist of:

- 1) Degreasing with a 2 percent NaOH solution.
- 2) Descaling with an inhibited 3 percent hydroxyacetic-formic acid solution.
- 3) Rinsing and passivating with a NaNO_2 solution.

Cleaning is scheduled for the week of October 25, 1976. Immediately following the cleaning operation, the steam and feedwater piping will be hydrotested.

TABLE VI-8

RESEARCH EXPERIMENT DETAILED PROCEDURES

Chapter	Title	Scope of Procedures
1	Systems Descriptions	Standard System Operating Configurations
2	Initial System Preparation	Functional checks, loading of media, N ₂ blankets
3	Startup Tests	Preheating, steam trace & heating, salt melting, blowdown, initial charge & discharge, shutdown
4	Component Handling	Water sampling & testing, changing N ₂ tanks
5	Normal Operations	Steady state & transient charging & discharging operations; mode reversal, daily checks, manual operation, steam consumption
6	Emergency Conditions	Loss of controls, loss of steam, manual override, spillage, line rupture, fire, other emergencies
7	Disaster Plans	
8	Maintenance	
9	Electrical & Mechanical Tests	
10	Instrumentation Calibration	

D. HYDROCARBON STORAGE FLUIDS

During this program we have been studying the thermal stability of commercially available hydrocarbon heat transfer oils. The intent of this study is to determine the best oil for the commercial plant, the Pilot Plant, and the Research Experiment.

All hydrocarbon oils decompose. The rate of decomposition is an exponentially increasing function of increasing temperature. The products of decomposition are classified into two groups:

- 1) High boilers (H.B.), those compounds that boil at temperatures higher than the fresh fluid, and
- 2) Low boilers (L.B.), those compounds that boil at temperatures lower than the fresh fluid.

The H.B. concentration in a heat transfer fluid tends to increase the viscosity of the fluid. The low boilers tend to vent off and must be replaced with make up fluid.

The hydrocarbon heat transfer fluids commercially available are relatively stable, that is, they have a low decomposition rate in the temperature range of ordinary use. However, when these oils are used in a thermal storage application, large quantities of the fluid are kept at elevated temperatures for extended periods of time; under these conditions even a small decomposition rate can be important.

Three sources of decomposition data were found: Dow Chemical, Monsanto, and Atomic Energy of Canada Limited (AECL). This decomposition data is presented in Figure VI-27. Also shown on Figure VI-27 is low cost oil weight loss data generated by MDAC. This data was used to predict long term oil performance. This oil performance prediction indicates that an oil side stream vacuum distillation unit could be necessary to continuously reprocess the oil. We are currently in the process of evaluating the potential benefit of such an oil side stream unit.

E. SCHEDULE FOR PROGRAM COMPLETION

The master schedule for the completion of the program is shown in Figure VI-28. The program is on schedule at this time and no significant delays or interruptions are foreseen. The Research Experiment test program will be in progress until mid-January and the Pilot Plant preliminary design will be conducted between that time and early April.

Figure VI-27 Total Oil Decomposition Rate as a Function of Temperature

Figure VI-28. Thermal Storage Subsystem Master Schedule.

VII. BALANCE OF PLANT INCLUDING ELECTRIC POWER GENERATION SUBSYSTEM

A. INTRODUCTION

The Bechtel portions of the 10 MWe Solar Pilot Plant include the Electric Power Generation Subsystem (EPGS), the receiver tower, the buildings, all non-solar unique systems, overall plant layout, and foundations for the Collector Subsystem. A conceptual design was prepared for all of these plant elements and was described in the first quarterly progress report in January, 1976. Subsequent efforts have included cost estimating, environmental analyses, EPGS optimization, support of Martin Marietta Corporation (MMC) computer simulation model preparation, and consulting to MMC on other tasks as requested.

This report summarizes the conceptual design and other task efforts covered in three prior reports for the periods from project inception on 7/1/75 through 6/30/76. EPGS optimization and interface engineering tasks performed in the period from 7/1/76 through 9/30/76 are covered in more detail.

B. CONCEPTUAL DESIGN OF EPGS

The EPGS conceptual design was based on the pilot plant requirements delineated by the Energy Research and Development Administration (ERDA). The major EPGS requirements are summarized as follows:

- Convert the receiver thermal output into 10 MWe net of electrical power at the design point, or
- Convert the Thermal Storage Subsystem (TSS) output into 7 MWe net of electrical power at the design point, or
- Convert simultaneous thermal outputs from the receiver and the TSS into at least 7 MWe net of electrical power at the design point, and
- Condition the working fluid (i.e., pressurize, heat, deaerate, treat chemically, etc., the condensate) and return it to the other subsystems, and
- Respond to transient conditions rapidly enough to avoid adverse effects on the operation of all subsystems.

The design point, as specified by ERDA, is at winter solstice at 2:00 pm with an ambient dry bulb temperature of 302⁰K (85⁰F).

The EPGS conceptual design effort was divided into three basic phases: the turbine selection, cycle studies, and heat rejection system selection; the development of the mechanical system descriptions, system diagram, line sizing, electrical system and instrumentation and controls; and preparation of layout drawings for the EPGS structure and the control and administration building. Start-up considerations were given special attention due to the necessity for a daily plant startup cycle.

Applicability of the conceptual design for use in a modularized commercial size plant has been stressed throughout the design effort.

A Reliability and Failure Modes and Effects Analysis (FMEA) of the EPGS was conducted to ensure a reliable design with adequate safeguards and redundancy.

1. Turbine Selection, Cycle Studies, and Heat Rejection System Selection

- a. Turbine Selection - A survey of the steam turbines available for use in the pilot plant led to the selection of a GE admission type turbine. In making this selection, first the desirable characteristics of the pilot plant turbine were compiled, and then the various turbine manufacturers were contacted to determine the characteristics of their commercially produced units.

In addition to the already stated criteria of meeting the ERDA 10 MWe and 7 MWe design requirements, the criteria which were used for the selection of the turbine type are the following:

- The turbine must be a standard design type normally offered by the manufacturer
- The turbine must be efficient using both direct receiver steam and steam produced by the TSS
- The turbine type should be adaptable by simple size scaling to a 100 MWe size range commercial facility
- The turbine must be tolerant of the daily cyclic operations characteristic of a solar power plant

The GE admission turbine was selected as providing the best match for the criteria. Also GE was the most responsive of the three manufacturers in providing design and performance data.

Other manufacturers contacted included Turbodyne, DeLaval, Westinghouse, and Allis Chalmers. Turbodyne recommended a dual inlet turbine in which both receiver and TSS steam were admitted through separate throttle valves to the turbine first stage. This arrangement may encounter thermal cycling problems in the first stage as the steam source is alternated between the receiver and the TSS. DeLaval manufacturers admission turbines similar to the GE design, but does not produce these units in the 100 MWe commercial size range. Westinghouse and Allis Chalmers do not manufacture turbines in the 10 MWe pilot plant size range.

The selected turbine has partial arc control valves at both the main and admission steam inlets, operated by an electro-hydraulic control system (EHC). A 12.5 MWe standard turbine-generator size is required to supply the specified 10 MWe net plus approximately 10% auxiliary loads.

Operations from the receiver alone, from storage alone, and combined operation do not require throttling and the consequent losses are avoided. Maximum available size for this turbine type is limited by the admission valve gear size to 125 MWe (when operating from the receiver).

The turbine inlet steam conditions were fixed for the conceptual design at 8720 kPa (1,250 psig), 783°K (950°F) for the main steam from the receiver, and 4240 kPa (600 psig), 672°K (750°F) for the admission steam from storage.

- b. Cycle Studies - The main steam inlet conditions were based on the recommendation of GE, made initially to Martin Marietta prior to this project, and subsequently to Bechtel. The temperature limit of 783°K (950°F) was to allow the use of a more ductile steel, as is recommended for turbines used in cyclic operation. The pressure limit of 8720 kPa (1,250 psig) is to reduce the thickness of the turbine casing, thereby reducing thermal stresses caused by cycling.

These steam conditions were reevaluated late in the conceptual design phase and main steam pressure was increased to 8790 kPa (1275 psig) to accommodate a TSS performance requirement.

The turbine exhaust conditions were determined by the heat rejection system type and size and the prevailing ambient conditions. Air cooling was chosen as the conceptual baseline approach in view of the uncertainties of water availability at sites in the southwestern states. This resulted in high turbine backpressures relative to more conventional water cooled designs. Ambient conditions were assumed for a hypothetical site based on a review of temperatures in the Southwest. 315°K (108°F) 1% dry bulb temperature was assumed for the study.

Performance calculations based on GE heat balances were made for the selected turbine system. Figures VII-1 and VII-2 illustrate the base heat balance diagram and Mollier Diagram resulting from the Conceptual Design Phase. Table VII-1 provided the heat balance values corresponding to Figure VII-1

- c. Heat Rejection System - Several manufacturers (GEA, MRM, Hudson and Marley) were contacted for performance information and physical data for a dry heat rejection system for the pilot plant. No restriction as to type was imposed in the request for information to the manufacturers.

GEA (with a direct air cooled condenser type system) was the only respondent during the conceptual design phase. They proposed a system with the fans and heat exchanger surface located above the turbine-generator area to provide head for the condensate pumps, to allow an air space under the fans to ensure an adequate air supply, and to minimize the exhaust ducting size and pressure drop.

In order to select the heat rejection system size, its cost differential was traded against plant performance differential. As the heat rejection system size is increased, lower turbine backpressures are obtained resulting in greater plant output. Performance curves, costs, and fan power requirements were obtained from GEA for three dry heat rejection systems, resulting in 20 kPa (6 in-Hg), 30 kPa (9 in-Hg), and 41 kPa (12 in-Hg) turbine back pressures. These condenser designs were based on the requirements specified by Bechtel for full turbine flow at 108°F dry bulb temperature. Based on a year-round comparison of the additional kilowatts resulting from the larger heat rejection systems against their additional cost, the 20 kPa (6 in-Hg) backpressure dry condenser was selected for the conceptual design baseline.

Figure VII-3 illustrates the design features of the selected GEA system.

FIGURE VII-1 BASE HEAT BALANCE DIAGRAM
(SEE TABLE VII-1 FOR VALUES OF A THRU Q)

FIGURE VII-2 EPGS MOLLIER DIAGRAM, 12.5 MWe GROSS

TABLE VII-1 BASE HEAT BALANCE VALUES

CYCLE POINT	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
RECEIVER STEAM ALONE 12.5 MW _e GROSS																	
● PRESSURE, kPa	8722				2261	889	279	109						20	104	265	3448
(psia)	1265				329.9	129.0	40.44	15.82						2.947	15.03	38.41	500
● TEMPERATURE, K	783	489												334	339	402	
(°F)	950	420.8												140.8	151.0	264.8	
● ENTHALPY, J/g	3415	926								3182		2798	2666	253	277	544	
(BTU/lb)	1468	398.0								1368		1203	1148	108.7	119.0	233.7	
● FLOW, kg/h	59003	59003	0	0					43776	4972	3354	2731	3142	47948	3142	59003	
(lb/h)	130077	130077	0	0					96508	10962	7394	6020	6926	105701	6926	130077	
RECEIVER STEAM ALONE 11.0 MW _e GROSS																	
● PRESSURE, kPa	8722				1876	783	246	97						20	92	234	3448
(psia)	1265				272	113.6	35.66	14.06						2.947	13.36	33.87	500
● TEMPERATURE, K	783	482												334	339	397	
(°F)	950	408.5												140.8	151.0	254.7	
● ENTHALPY, J/g	3415	895								3166		2788	2657	253	275	526	
(BTU/lb)	1468	384.6								1361		1199	1142	108.7	118.1	226.1	
● FLOW, kg/h	51963	51963	0	0					39069	4191	2835	2351	2551	42568	2551	51963	
(lb/h)	114556	114556	0	0					86130	9239	3250	5182	5626	93885	5626	114556	
ADMISSION STEAM ALONE 9.4 MW _e GROSS																	
● PRESSURE, kPa			4238		2199	840	264	105							100	250	3448
(psia)			614.7		318.9	121.9	38.24	15.24							2.947	14.48	36.33
● TEMPERATURE, K			672	488											334	339	401
(°F)			750	418.3											140.8	151.0	261.5
● ENTHALPY, J/g			3209	919						3120		2753	2627	253	277	536	
(BTU/lb)			1380	395.2						1342		1184	1130	108.7	119.0	230.3	
● FLOW, kg/h	0	0	57313	57313					42657	5026	3514	2596	2987	46175	2987	57313	
(lb/h)	0	0	126352	126352					94042	11080	7748	5724	6584	101796	6584	126352	

(FOR LOCATION OF CYCLE POINTS A THRU Q, SEE FIGURE VII-1 ON PAGE VII-5)

COURTESY OF GEA

FIGURE VII-3 DRY COOLING SYSTEM DESIGN FEATURES

2. Commercial Plant Applicability

Since the 10 MWe pilot plant is intended to simulate future commercial plant operation, the commercial plant turbine possibilities were examined before choosing a pilot plant turbine type.

Three commercial utility plant turbine manufacturers (General Electric, Westinghouse and Allis-Chalmers) were contacted for their recommendation of a turbine type for this application in the 100-300 MWe range. They all recommended that the first commercial plant be kept near the bottom end of this range, since greater scale-up from a 10 MWe pilot plant would be unrealistic. General Electric proposed both the mixed pressure and admission turbines, but recommended the admission turbine as the most flexible. Westinghouse recommended an induction type turbine. Allis-Chalmers declined to recommend a specific turbine type since they do not manufacture turbines smaller than 200 MWe.

3. Startup and Modes of Operation

- a. Startup - Daily thermal cycling of the turbine for starts and operational mode changes will impose life reducing thermal stresses on the turbine. However, startup procedures can be formulated to minimize the diurnal thermal stresses.

Turbines can be subjected to limited thermal cycling without adversely affecting the turbine life. As thermal cycling becomes more severe, turbine life is shortened. This relationship is expressed for each turbine by a cyclic life expenditure curve relating the rate of temperature change and the amount of life expenditure for that change. An example of this type of curve is shown in Figure VII-4 for a large commercial turbine.

In the case shown, up to a 56°K (100°F) step temperature change could be imposed on the turbine repeatedly and still retain a 20 year life expectancy (6000 starts or 0.17% per cycle). An overnight temperature drop of 167°K (300°F) would require one hour to bring the turbine up to temperature. This does not include time to bring the turbine up to speed and synchronization or consideration of other transients and their life reducing effects. A realistic hot startup time for the turbine illustrated in Figure VII-4 considering these other factors would probably be between 90-120 minutes. This time can be reduced for the commercial unit by incorporating overnight heat retention measures.

Figure VII-5 shows the corresponding turbine cyclic life expenditure curve for the GE admission turbine selected for the pilot plant. Comparing this with Figure VII-4, it can be seen that this smaller turbine is more amenable to thermal cycling. Up to an 83°K (150°F) step temperature change can be imposed repeatedly without any adverse effects. An overnight temperature drop of 167°K (300°F) requires only 25 minutes to bring the turbine up to temperature (i.e., a ramp rate of 389°K/H (700°F/h) for 0.017 percent life expenditure per cycle). A realistic hot startup time (considering time to bring the turbine up to speed, synchronize and handle other transient effects) would be approximately 50 minutes. It is interesting to note that the pilot plant turbine can be subjected to repeated temperature ramp rates of up to 222°K/h (400°F/h), regardless of the total temperature change, without affecting the life of the turbine. The typical commercial turbine discussed previously could only tolerate up to 111°K/h (200°F/h) repeated ramp rates.

b. Modes of Operation - As determined in the conceptual design stage, the modes of operation of the EPGS are:

- Overnight shutdown
- Startup sequence
- Operating from the receiver alone
- Operating from the receiver while charging storage
- Operating from the receiver and storage
- Operating from storage alone
- Shutdown sequence
- Receiver charging storage alone

4. EPGS Systems

For convenience, the EPGS has been broken into five major systems as follows:

- Main Steam, including related auxiliary systems such as extraction steam, turbine steam seals, turbine-generator lube oil, turbine control oil, and condenser air removal

FIGURE VII-4 TYPICAL COMMERCIAL TURBINE CYCLE LIFE EXPENDITURE

FIGURE VII-5 RECOMMENDED TEMPERATURE RAMP VS. TEMPERATURE CHANGE FOR GE 12.5 MWe ADMISSION TURBINE

- Feedwater, including related systems such as feed-water heater vents and drains, full flow demineralizers, and feedwater pumps
- EPGS Auxiliary Systems, such as the makeup demineralizers, closed cooling water, cranes and hoists, fire protection, compressed air, and emergency diesel generator
- EPGS Controls Systems, including turbine-generator instrumentation and controls, feedwater controls, and other internal control systems
- EPGS Electrical Systems, including motor control centers, generator breakers, grounding and other electrical aspects of the EPGS

System descriptions have been developed for these systems. These systems will be further defined during the preliminary design stage.

The EPGS conceptual design plant layout is shown in Figure VII-6. The air cooled condenser units cover the turbine-generator area and the turbine exhaust ducting extends from beneath the turbine up to the steam distribution headers running horizontally above the air cooled condenser units. The deaerator is located on the air cooled condenser deck to provide sufficient suction head to the booster pumps.

The turbine-generator is mounted on an elevated pedestal with space underneath for the main exhaust duct, the main steam and admission inlet lines, extraction piping, oil reservoirs, pumps and other miscellaneous turbine related equipment.

Auxiliary Power Requirements

Auxiliary power requirements were calculated for the EPGS for all designated modes of operation at the design point ambient condition. Table VII-2 presents the results for two operational modes; from receiver only, and from storage only. The principal difference in auxiliary power requirements is the elimination of the high pressure feedwater pumps when operating from storage only. Changes in power required for pumps were calculated from flow rate and pressure head values corresponding to the operating modes. Referring to Table VII-2, the cooling system fans stand out as the auxiliary power requirement where the most benefit could be obtained from power reductions. Condenser fan power for receiver only and storage only

VII-12

APPROVED FOR CONCEPTUAL DESIGN									
NO.	DATE	REVISIONS	BY	CHKD	DATE	REVISIONS	BY	CHKD	DATE
1	11-6								
DRAWN BY S. B. MEYERMAN									
CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM - PHASE NO. 1									
ELECTRIC POWER GENERATION SUBSYSTEM PLANT LAYOUT									
11480		11480-004-M1							

TABLE VII-2
EPGS AUXILIARY POWER REQUIREMENTS, kW_e

EPGS	STEAM SOURCE	
	RECEIVER	STORAGE
TURBINE LUBE AND CONTROL OIL SYSTEM	12.0	12.0
COOLING SYSTEM FANS	546.0	546.0
VACUUM PUMP	6.0	6.0
CONDENSATE PUMP	7.5	6.5
BOOSTER PUMP	67.0	60.0
FEEDWATER PUMP, LP	26.0	24.0
FEEDWATER PUMP, HP	168.0	----
CHEMICAL INJECTION	0.5	0.5
AIR COMPRESSOR (22 kW @ .5 SF)	11.0	11.0
FIRE JOCKEY PUMP	2.5	2.5
POTABLE WATER PUMP	1.5	1.5
MAKEUP WATER (DEEPWELL) PUMP	5.0	5.0
SANITARY WATER PUMP	0.8	0.8
COOLING WATER PUMP	11.0	11.0
HVAC	20.0	20.0
LIGHTING	4.0	4.0
CONTROL POWER	10.0	10.0
AIRPLANE BEACON	4.0	4.0
MISC.	10.0	10.0
TOTAL	912.8	734.8

operation are 60% and 74%, respectively, of the total auxiliary power requirement. This concentration of power in one system provides an opportunity for optimization improvements during preliminary and final design.

6. Failure Modes and Effects Analysis

Subsequent to conceptual design completion a Failure Modes and Effects Analysis (FMEA) was performed on the EPGS. Results incorporate the improved reliability caused by use of redundancy in pumps and other components wherever feasible. The scale-up of the Pilot Plant to commercial size will include use of redundant components so that the commercial plant will also provide high reliability. An update of this analysis will be performed when preliminary designs are available.

C. CONCEPTUAL PLOT PLAN DEVELOPMENT

The purpose of the conceptual plot plan is to locate the major subsystems and identify the minor subsystems. Defining the relative positions of the major subsystems enables the finalization of their conceptual design. The identification of minor subsystems--utilities, maintenance areas and roads--showed the additional facilities required for a functional plant. The conceptual plot plan is shown in Figure VII-7.

1. The Plot Plan

Since no site had been selected for the plant, it was assumed to be level, requiring no major grading.

The collector field versus the receiver tower positioning was governed by the north-south orientation as dictated by the system concept. The rest of the plant facilities were grouped around the receiver tower in accordance with their functional relationships, with due consideration for minimizing pipe run lengths.

2. General Plant Arrangement

Figure VII-8 shows details of the principal facilities south of the Collector Subsystem. Referring to this figure, the 21.6 m (71.0 ft) diameter (at the base) receiver tower is located in the center of this diagram. The EPGS plant is located symmetrically about the east-west centerline of the receiver tower, and to the west of it. It is an open steel structure erected on a reinforced concrete slab on grade, 25.0 m (85.0 ft) along the east-west length and 19.8 m (65.0 ft) along the north-south width.

FIGURE VII-7 PLOT PLAN

- ### LEGEND
①	COLLECTOR FIELD
②	RECEIVER TOWER
③	TERMINAL, STOR. SUBSYSTEM, HEAT EXCH'G PAD NO.1
④	TERMINAL, C.O.R. SUBSYSTEM, HEAT EXCH'G PAD NO.2
⑤	TERMINAL, STOR. SUBSYSTEM, HEAT
⑥	TERMINAL, STOR. SUBSYSTEM, HEAT
⑦	TERMINAL, STOR. SUBSYSTEM, HEAT
⑧	TERMINAL, STOR. SUBSYSTEM, HEAT
⑨	POWER GENERATION SUBSYSTEM, TURBINE GENERATOR
⑩	POWER GENERATION SUBSYSTEM, SUBSTATION
⑪	POWER GENERATION SUBSYSTEM, TRANSMISSION LINE TO UTILITY NETWORK
⑫	POWER GENERATION SUBSYSTEM, CONDENSATE STORAGE TANK.
⑬	CONTROL & ADMINISTRATION BUILDING
⑭	MAINTENANCE, SHOPS & FIRE TRUCK GARAGE.
⑮	FIRE WATER STORAGE TANK.
⑯	FIRE PUMP HOUSE
⑰	WATER SUPPLY LINE
⑱	WELL FIELD
⑲	SANITARY SEWER LINE
⑳	SEWAGE TREATMENT PLANT.
㉑	PARKING AREA
㉒	GATE HOUSE

[illegible]

[illegible]

The tank farm of the TSS consists of three Hitec storage tanks, one Hitec surge tank, three hydrocarbon oil storage tanks and one hydrocarbon oil surge tank. Two nitrogen blanket control systems are provided, one for each of the two sets of tanks. This tank farm is located south of the main plant road and the area is served by a road system that forms a complete access road loop. A heavy duty, grating covered, concrete trench system provides for drainage of any hydrocarbon spillage in case of a tank or piping rupture. This trench system discharges the spillage to an open ditch which leads to a retention pond sized at 3,785 m³ (1,000,000 gallons) to hold the full volume of one hydrocarbon storage tank.

D. COLLECTOR FOUNDATION CONCEPTUAL DESIGN

The main objective of this effort was to develop a cost-effective foundation design for the heliostats and their associated sensors in the Collector Subsystem. In this non-site specific conceptual design stage, environmental criteria were hypothesized. Design specifications were also developed by wind tunnel testing performed by Martin Marietta. Based on these criteria and specifications, Bechtel examined nine alternatives for the heliostat foundation, and relative costs were determined for these alternatives. This resulted in selection of the conceptual design foundation.

1. Design Criteria and Data

The environmental design criteria covered wind conditions, soil type, and site topography.

Design load criteria including axial load, shear load, and bending moment were supplied by MMC. An allowable vertical wobble of 0.291 mrad (± 1 arc minute) was allocated for foundation movements.

- a. Wind Conditions - The foundation was designed to support the heliostat for operation at steady wind speeds of up to 13.4 m/s (30 mph) and at gusts up to 17.9 m/s (40 mph). When the wind speed reaches 22.4 m/s (50 mph), the heliostat will be stowed for protection; and in the stowed position it will survive at wind speed up to 53.6 (120 mph). All wind speeds are measured at 9.1 m (30.0 ft) above ground. The foundation for the sensor was designed to support it at the survival condition of a 53.6 m/s (120 mph) wind.
- b. Soil Type - While it is understood that soil properties generally vary within a large selected area, according to geologic formation and location, the soils considered in this study include

such types as medium to coarse dry sand, clays, or silty clays with sand, which are considered to be representative of the Southwest U.S. For the purpose of this conceptual design stage, it was assumed that such soils extend to a depth of at least 9.1 m (30.0 ft) below the surface, and the soil bearing capacity was taken to be 191,500 Pa (4,000 lb/ft²), an average value for these soil types.

- c. Site Topography - A flat site was adopted as a baseline for the conceptual design of the collector field. Sloping or hilly sites may result in changes to the conceptual design.

2. Conceptual Design

The conceptual foundation design selected for both heliostat and sensor was the mat type illustrated in Figure VII-9. Eight other configurations were considered and determined less desirable from either a performance or cost point of view.

E. RECEIVER TOWER CONCEPTUAL DESIGN

1. Receiver Tower

Soil bearing strength and wind loading criteria used to design the receiver tower were the same as those reported in Section VII-D-2, in conjunction with the collector foundation design. A ground level seismic acceleration of $\pm 0.3g$ in the vertical and horizontal directions was used. A receiver weight of 270,000 Kg (300 tons), and a receiver to tower top interface diameter of 50 ft. were provided by MMC with a 131.7 m (432 ft) tower top height. A seismic design approach using Nuclear Regulatory Commission design practice with the less conservative Operation Basis Earthquake (OBE) were used for tower analysis. Three alternate designs were examined: guyed steel tower, free standing steel tower, and reinforced concrete tower. The concrete tower was selected based on cost and tower top sway characteristics. The sway limit of ± 0.53 m (1.75 ft) at the receiver aperture center was met with the conceptual design concrete tower

A service elevator well was located behind the receiver. The elevator reaches the lower level of the receiver; it also provides access to the intermediate working platforms. The conceptual design tower is illustrated in Figure VII-10.

A rectangular access hatch was provided to allow bulky equipment to pass through the supporting platform. Portable hoisting equipment

FIGURE VII-9 MAT FOUNDATION FOR HELIOSTAT AND SENSOR

will be used as required during construction. Figure VII-10 also shows the stairway and the positions of the riser and downcomer pipes. The stairway provides alternate access to the receiver from ground elevation.

2. Riser and Downcomer Piping

The feedwater is carried up the receiver tower in a riser pipe and the superheated steam is carried from the receiver to the EPGS and the TSS in a downcomer piping system. In designing this piping system the following items were addressed:

- Single versus multiple risers and downcomers
- Pipe sizes and required installation
- Method of designing for thermal expansion

A single pipe arrangement has lower capital cost, a lower installation cost, and requires less pump power. The use of single piping runs simplifies the design of flexible configurations and permits structural optimization for support and pipe restraints. In addition, associated valving and controls with common distribution headers is simpler and less costly.

The optimum sizing of pipe requires an analysis of the economic trade-off between the higher capital cost for a large pipe, and the higher pressure drop for a smaller pipe. Thus, it is necessary to know the cost of pump power as determined from both plant cost and plant annual output. With no conceptual design data available in these areas, reasonable pressure and temperature losses in the pipes were assumed. The assumed values are:

Pressure Drop in Riser	=	1,379 kPa (200 psi)
Temperature Loss in Riser	-	2.3 °K (5°F)
Pressure Drop in Downcomer	=	234 kPa (50 psi)
Temperature Loss in Downcomer	=	5.6 °K (10°F)

Pressure head loss is a combination of friction and elevation head losses. The friction head loss for a given pipe size is a function of the total effective length, including the accumulated friction losses due to the number and type of valves, the number of pipe elbows and expansion loop bends, and friction loss in the actual

pipe length. The pipe size required was calculated on the basis of these considerations. The riser was determined to be a carbon steel nominal four inch diameter schedule 80 pipe with 2.5 cm (1 in) insulation, and the downcomer a one and one-quarter percent chrome-one half percent moly nominal eight inch diameter schedule 100 pipe with a 15 cm (6 in) insulation. The total temperature loss of the downcomer is 5.7°K (10°F) of which 1.7°K (3°F) is associated with the 345 kPa (50 psi) pressure loss. The insulation was sized to limit temperature reductions to the remaining 4.0°K (7°F). Calcium silicate was chosen as the insulation material since it is generally used for power plant piping and appears to be more economical than alternative materials such as mineral wool or fiberglass.

Thermal expansion of feedwater riser and steam downcomer piping was accommodated in piping loops. This technique is the standard "state-of-the-art" for power plant design. Bellows-type expansion joints were considered for the conceptual design but were not accepted because they have not been proven reliable in power plant installations at these pressure and temperature levels.

F. PRELIMINARY ENVIRONMENTAL IMPACT IDENTIFICATION AND SITING CRITERIA

1. Introduction

The conceptual design of the pilot plant was reviewed to identify the major environmental impacts and to develop siting criteria aimed at mitigating these impacts. The results of this task effort are summarized in Tables VII-3 and VII-4. They are briefly discussed in this section in terms of impacts to the physical and chemical, biological, and the social environments.

A site which has an avoidance characteristic in Table VII-4 should be removed from consideration because regulatory agencies or the public would not be expected to give approval for construction and operation of the plant. A site which has one or several characteristics in the downgrading category is environmentally less desirable than one without these characteristics, but need not be eliminated from consideration.

More specific environmental analysis is planned during the preliminary design phase using actual site data to make recommendations for design changes if necessary to mitigate downgrading characteristics.

2. Impacts to the Physical and Chemical Environment

- a. Topography - The conceptual design pilot plant occupies a total area of about 393,000 m² (97 acres). Construction of

	PROJECT COMPONENTS		COLLECTOR SUBSYSTEM	RECEIVER SUBSYSTEM	THERMAL STORAGE SUBSYSTEM	ELECTRICAL POWER GENERATION SUBSYSTEM
	ENVIRONMENTAL CHARACTERISTICS					
PHYSICAL AND CHEMICAL	TOPOGRAPHY		• Alterations due to installation of heliostate and roads	• Minor alterations due to receiver tower installation	• Minor alterations due to installation of equipment and spill retention pond	• Minor alterations due to equipment installation
	SOILS		<ul style="list-style-type: none"> • Potential erosion from rainwater • Disturbance of soils during construction • Disruption of existing heating and cooling cycles • Quality degradation if solvents are used for mirror washing 	• Removal of soil in foundation area	<ul style="list-style-type: none"> • Removal of soil in tank and pond area • Potential damage from major spills 	• Removal of soil in foundation area
	SURFACE WATER HYDROLOGY AND QUALITY		<ul style="list-style-type: none"> • Altered surface drainage patterns • Turbidity as a result of erosion • Degradation if solvents are used for mirror washing 	• No major impacts anticipated	• Potential damage from major spills during operation or thermal storage fluid delivery	• No major impacts anticipated
	GROUND WATER HYDROLOGY AND QUALITY		<ul style="list-style-type: none"> • Slight potential for interference with ground water recharge • Quality will be degraded if solvents are used for mirror washing if they enter the ground water system • Potential ground water drawdown for water supply 	• Potential ground water drawdown for make-up water supply	<ul style="list-style-type: none"> • Potential damage from major spills • Potential ground water drawdown for make-up water supply 	• Potential ground water drawdown for water supply
	METEOROLOGY AND AIR QUALITY		<ul style="list-style-type: none"> • Alterations to local microclimate, including temperature and air circulation patterns • Air quality temporarily affected by dust generated during construction 	• Air quality temporarily affected by dust generated during construction	<ul style="list-style-type: none"> • No major impacts anticipated during normal operation • Potential temporary air quality degradation if major spills occur • Air quality temporarily affected by dust generated during construction 	<ul style="list-style-type: none"> • Minor emissions from operation of standby generator • Cooling tower waste heat of about 25 MWt • Air quality temporarily affected by dust generated during construction
	NOISE		• Minor contribution to local noise levels due to vehicles	• Minor contribution to local noise levels due to fluid flows	• Minor contribution to local noise levels due to equipment	• Minor contribution to local noise level due to equipment
BIOLOGICAL	TERRESTRIAL ECOLOGY		<ul style="list-style-type: none"> • Removal of up to 330,000 square meters of habitat • Retardation or prevention of normal revegetation 	<ul style="list-style-type: none"> • Preemption of terrestrial habitat • Interference with flocks of migrating birds - collision with tower or mortality from heat is expected 	<ul style="list-style-type: none"> • Preemption of terrestrial habitat • Potential damage from major spills 	• Preemption of terrestrial habitat
	AQUATIC ECOLOGY		• Potential impact if solvents are used for mirror washing	• No major impacts anticipated	• Potential damage from major spills	• No major impact anticipated
SOCIAL	LAND USE		• Preemption of 330,000 square meters of land	• Preemption of a small section of land	• Preemption of a small section of land	• Preemption of a small section of land
	AESTHETICS		• In flat area, highly visible	• In flat area, highly visible	• In flat area, highly visible	• In flat area, highly visible
	SCIENTIFIC/CULTURAL		• Potential impact on scientific/cultural resources	• Potential impact on scientific/cultural resources	• Potential impact on natural scenic and cultural resources	• Potential impact on scientific/cultural resources

TABLE VII-4 SUMMARY OF ENVIRONMENTAL IMPACT SITING CRITERIA

	SITING CRITERIA ENVIRONMENTAL CHARACTERISTICS	DOWNGRADING FACTORS	AVOIDANCE FACTORS
PHYSICAL AND CHEMICAL	TOPOGRAPHY	<ul style="list-style-type: none"> • Extensive topographical modification 	<ul style="list-style-type: none"> • No applicable criteria
	SOILS	<ul style="list-style-type: none"> • Highly productive soils from a commercial fertility or an ecological perspective 	<ul style="list-style-type: none"> • Highly erodable or unstable soils
	SURFACE WATER HYDROLOGY AND QUALITY	<ul style="list-style-type: none"> • Changes in surface drainage patterns, resulting in increases or decreases of flow to the detriment of human or ecological use • Temporary deterioration of quality to the detriment of human or ecological use 	<ul style="list-style-type: none"> • Long-term deterioration of quality to the detriment of human or ecological use
	GROUND WATER HYDROLOGY AND QUALITY	<ul style="list-style-type: none"> • Removal of ground water resources from other existing beneficial uses 	<ul style="list-style-type: none"> • Reduction of ground water quality below limits acceptable for other beneficial uses, particularly human consumption
	METEOROLOGY AND AIR QUALITY	<ul style="list-style-type: none"> • No applicable criteria 	<ul style="list-style-type: none"> • No applicable criteria
	NOISE	<ul style="list-style-type: none"> • No applicable criteria 	<ul style="list-style-type: none"> • No applicable criteria
BIOLOGICAL	TERRESTRIAL ECOLOGY	<ul style="list-style-type: none"> • Recreationally or commercially important, on a regional basis, wildlife species • Adjacent important wildlife populations whose success would be affected • Vegetation species critical to the food base of important wildlife 	<ul style="list-style-type: none"> • Wildlife, vegetation or habitats which are threatened or endangered
	AQUATIC ECOLOGY	<ul style="list-style-type: none"> • Recreationally or commercially important, on a regional basis, aquatic species and habitats • Adjacent important aquatic populations and habitats whose success would be affected 	<ul style="list-style-type: none"> • Aquatic species or habitats which are threatened or endangered
SOCIAL	LAND USE	<ul style="list-style-type: none"> • Incompatible with present or future land use • Dislocation of any residences and business 	<ul style="list-style-type: none"> • Existing heavily utilized recreation activities or commercially important agriculture • Hazards to the pilot plant, e.g., airports • Dislocation to the detriment of community structure
	AESTHETICS	<ul style="list-style-type: none"> • Highly visible or reflections visible from road or residences 	<ul style="list-style-type: none"> • Highly prized scenic vistas
	SCIENTIFIC/CULTURAL	<ul style="list-style-type: none"> • Adjacent scientific, cultural, or archaeological features which would be adversely affected 	<ul style="list-style-type: none"> • Known scientific, cultural or archaeological features on site

the plant will require clearing and grading to create the design profile and drainage for the facilities. This will modify the area's topography. By selecting a site that is relatively flat, large-scale topographic changes can be avoided. The topographic features of the site should also be assessed in terms of aesthetics. Undesirable visibility of the plant from nearby residences or roads should be minimized.

- b. Soils - Soils in the site area will be disturbed during construction and will be partially covered or shaded by the constructed facilities. This will cause a change in the land surface characteristics and result in the disruption of the natural heating and cooling cycles. The natural interaction between the soils and the atmospheric environment will be changed and their function as a biologic substrate will be curtailed. Sites having highly productive soils from a commercial fertility or an ecological perspective should be downgraded. Also, highly erodible and unstable soils are unsuitable both from an environmental and from an engineering perspective and should be avoided.
- c. Microclimatology - The shading of a significant land area under solar collectors and the addition of waste heat from the condenser with a plume under some conditions could cause very localized climatic differences. The probable remote location of the plant and the limited geographic area involved will minimize the effects of such changes on human or animal habitats. This potential environment effect will nevertheless be considered in more detail relative to the specific site selected.
- d. Other - Surface water hydrology and quality, groundwater hydrology and quality, meteorology and air quality and noise were also evaluated for impact to the physical and chemical environment.

3. Impacts to the Biological Environment

- a. Terrestrial Ecology - The type of biological communities and their sensitivity to disturbance varies with location. The Southwest U.S. is characterized by desert or arid grassland, woodlands, or chaparral. Although biological communities in this area are relatively stable and hardy ecologically, once this environment and balance is disturbed, recovery is slow.

Of concern in a Southwest U.S. site (where major flyways traverse) is the potential for birds to collide with the receiver tower or to die from exposure to concentrated solar beams. The hazard from concentrated beams would be only in the immediate vicinity of the receiver aperture. Operating experience since 1970 with the French Solar Furnace at Odeillo has indicated that birds will avoid the solar beams but that insects are attracted and destroyed at the receiver aperture.

- b. Aquatic Ecology - Aquatic biology will also be considered in the conceptual evaluation and will be more specifically addressed after a site is selected.

4. Impacts to the Social Environment

- a. Land Use - The plant requires a large parcel of land and existing land use must be preempted. To minimize this impact, sites where the plant would be incompatible with present or future land use should be downgraded.
- b. Aesthetics - The pilot plant conceptual design assumes a flat site. For this reason, the conspicuousness of the receiver tower, the extensive reflective nature of the collector field and the general industrial appearance of the plant may impact on the aesthetics of scenery and broad uninterrupted panoramas. Therefore, sites where the plant would be prominent in highly prized scenic vistas should be avoided. Sites where the facility would be highly visible from nearby roads or residences should be downgraded.
- c. Scientific/Cultural - Sites containing known archaeological, cultural or scientific features should be avoided. A candidate site should be downgraded if it is situated near or adjacent to such a site and use of it would adversely affect its historical/archaeological features.

G. EPGS OPTIMIZATION

1. Introduction

The major effort subsequent to conceptual design was the completion of the EPGS optimization study. The prime objective of the study was to reduce total solar pilot plant costs by trading costs associated with EPGS performance improvements against reduced costs in other plant subsystems. For the optimization, plant performance was evaluated using the 10 MWe (receiver operation) and

7 MWe (storage operation) design requirements. A detailed report covering the optimization was delivered to MMC in September, 1976.

During the optimization, care was taken to avoid introducing equipment to the pilot plant which would be substantially different from equipment which would be used in a 100 MWe (or larger) commercial plant. In some studies, such as the condenser location studies described in Section VII-G-3b below, results are applicable to both the pilot plant and the commercial plant.

2. Optimization Areas

The optimization study covered several aspects of the turbine cycle including selection of the main turbine inlet and exhaust conditions, selection of the dry condenser design point, determination of the optimum number of feedwater heating stages, and optimization of feedwater heater surface areas. Also included in the study were recommendations covering the feedwater pump arrangement, size of major pipe runs, and the source of auxiliary steam for overnight shutdowns.

The conceptual design turbine cycle previously described in Section VII-B-1 was used as a starting point for the optimization analysis. Table VII-5 compares a conceptual turbine cycle performance with the optimized design. To achieve an accurate comparison, the conceptual design performance shown in Table VII-5 has been updated to reflect the revised EPGS auxiliary power requirements presented in the April 1976 quarterly report.

Heat balance diagrams showing optimized turbine performance at the 10 MWe or 7 MWe ERDA design performance levels are shown in Figures VII-11 and VII-12. The performance data used to construct these diagrams, as well as turbine performance data used in the optimization analyses, were calculated using a specially modified Bechtel computer code.

The power cycle shown in Figures VII-11 and VII-12 is more complex than normally recommended for a 10 MWe power plant. The explanation for this complexity lies in considering the cost trade leverage available when optimizing the EPGS. For the pilot plant, EPGS costs per kilowatt are less than one-third the costs per kilowatt estimated for the collector field and receiver.

When the costs of EPGS efficiency gains are weighed against the corresponding reduction in collector field and TSS costs, substantially more investment in the EPGS can be justified than for

TABLE VII-5

PERFORMANCE COMPARISON - CONCEPTUAL DESIGN VS. OPTIMIZED DESIGN

	10 MWe ERDA DESIGN POINT		7 MWe ERDA DESIGN POINT	
	<u>CONCEPTUAL DESIGN *</u>	<u>OPTIMIZED DESIGN</u>	<u>CONCEPTUAL DESIGN *</u>	<u>OPTIMIZED DESIGN</u>
NET ELECTRICAL GENERATION, kWe	10,000	10,000	7,000	7,000
THERMAL INPUT TO TURBINE CYCLE, kWt	36,086	34,595	30,566	28,520
GROSS HEAT RATE, kJ/kW-hr (Btu-kW-hr)	11,412 (10,864)	11,341 (10,749)	13,531 (12,825)	13,240 (12,550)
STEAM FLOW TO TURBINE, kg/hr (lb/hr)	52,087 (114,832)	50,341 (110,982)	47,126 (103,896)	42,920 (94,630)
GROSS ELECTRICAL GENERATION, kWe	11,334	10,982	8,132	7,750
INLET PRESSURE, kPa (psig)	8,722 (1,250)	8,894 (1,275)	4,238 (600)	4,238 (600)
INLET TEMPERATURE, °C, (°F)	510 (950)	510 (950)	399 (750)	420 (800)
EXHAUST PRESSURE, kPa (inches HgA)	12.5 (3.7)	14.0 (4.1)	12.0 (3.5)	12.0 (3.5)
FAN SPEED	FULL	HALF	FULL	HALF

* Updated to include revised auxiliary power requirements

LEGEND

SI UNITS:

kJ/kg - ENTHALPY, KILOJOULES/KILOGRAM
kg/hr - FLOW, KILOGRAMS/HOUR
kPa - ABSOLUTE PRESSURE, KILOPASCALS
C - TEMPERATURE, °C

ENGLISH UNITS:

(H,h) - ENTHALPY, BTU/LBM
(#) - FLOW, LBM/HR
(P) - PRESSURE, PSIA
(F) - TEMPERATURE, °F

FIGURE VII-11 ERDA 10 MWe DESIGN POINT HEAT BALANCE FOR RECEIVER STEAM OPERATION

LEGEND

SI UNITS:

kJ/kg-ENTHALPY, KILOJOULES/KILOGRAM
kg/hr-FLOW, KILOGRAMS/HOUR
kPa -ABSOLUTE PRESSURE, KILOPASCALS
C -TEMPERATURE, °C

ENGLISH UNITS:

(Btu/lbm)-ENTHALPY, BTU/POUND
(#) -FLOW, LBM/HR
(PSIA) -PRESSURE, POUNDS PER SQUARE INCH
(°F) -TEMPERATURE, DEGREES FAHRENHEIT

conventional power plants. In addition, the effect of the cost trade is to shift the expenditures from areas under technological development (the collector field and TSS) to an established technology (the turbine plant). For these reasons, an efficient, cost effective EPGS is a most effective means of lowering the pilot plant costs.

3. Summary of Recommendations

The recommendations presented in the optimization report and a brief summary of the studies and reasoning supporting each recommendation are included here. Studies of the turbine cycle proceeded in an order which generally follows the order of the presentation in this section. First, turbine inlet steam conditions for both receiver steam and admission (storage) steam were analyzed and fixed. The condenser and turbine last stage blading were then studied to fix turbine exhaust conditions. Following this, options involving the feedwater heaters and pumps were considered to complete the turbine cycle portions of the study. Studies relatively independent of the turbine cycle were conducted in parallel with the turbine studies. Examples of this type of study are condenser location, and pipe and valve sizing.

- a. Turbine Related Studies - Recommendations related to the turbine included use of 8894 kPa, 783^oK (1275 psig, 950^oF) throttle conditions for receiver (solar) steam operation, use of 4240 kPa, 700^oK (600 psig, 800^oF) throttle conditions for admission (storage) steam operation, and use of nominal 10" last stage blading in the turbine design.

The steam conditions recommended for receiver steam operations were based on a combination of factors including a turbine manufacturer's comments, and consideration of effects on the TSS. General Electric Company engineers stated that, while feasible, they had not designed and built a turbine of the size and type intended for the cyclic operation of the pilot plant with throttle pressures in excess of 8720 kPa (1250 psig). They further advised that, although more energy would be available in the steam at higher pressures, increased steam-leak-offs and shorter, less efficient turbine blading would partially offset the potential performance gains. Also, raising throttle pressure significantly would raise hot oil storage temperatures in the TSS. Since oil decomposition rates at elevated temperatures are an area under investigation, further increases in throttle pressures were considered imprudent for the preliminary design. For these reasons, a pressure of 8894 kPa (1275 psig)

was selected as providing a good match of inlet conditions for TSS and EPGS operations. The 783°K (950°F) throttle temperature is based on earlier recommendations for maximum throttle temperature applicable to units used for peaking (cyclic) service.

The recommendation of 4238 kPa, 700°K (600 psig, 800°F) admission is based primarily on the improved cycle efficiency and resulting decrease in required TSS storage capacity obtained when the admission temperature is increased from 672°K (750°F) to 700°K (800°F). In addition, an almost exact match with the steam temperatures that exist at the admission point during receiver steam operation is achieved. Admission pressures of 3550 kPa (500 psig) and 3860 kPa (400 psig) were also studied, but were eliminated from further consideration because of the resulting poorer heat rates and corresponding increases in thermal storage requirements.

As part of the condenser studies reported in the following paragraphs, an investigation of the turbine last stage (lowest pressure) blade length was conducted. The standard GE nominal blade lengths of 8 inches and 10 inches were considered using performance data supplied by GE. The results indicate that the 10 in. last stage blade (LSB) gives slightly better performance at the ERDA 10 MWe and 7 MWe performance requirements. For this reason, and because the longer blade allows for efficient adaptation to wet cooling schemes, the 10 in. LSB was selected in place of the 8 in. LSB included in the conceptual design report.

- b. Condenser Studies - Like the conceptual design, a dry, air cooled condenser was assumed to achieve maximum siting flexibility for the pilot plant. The condenser design point, location and configuration were analyzed and recommendations formulated.

With regard to the design point, the recommendation was made that the condenser be designed to produce a turbine exhaust pressure of 16.9 kPa (5 inches HgA) at the design maximum ambient temperature of 315°K (108°F). If the design ambient temperature for the actual site exceeds this temperature, a recommendation that the design point be reexamined was made since increases in condenser surface area could quickly upset the calculated cost trades.

A study of location of the dry condenser resulted in a recommendation to relocate the condenser from the roof of the EPGS structure (as assumed for the conceptual design) to the ground

adjacent to the EPGs. This relocation reduced overall capital costs, and permitted using flat rather than sloping "chevron" design heat exchange surfaces. The flat surfaces permit more efficient fan cooling of the surfaces and facilitate dry condenser maintenance.

- c. Plant Control Studies - Recommendations in the plant control area include use of Electro-Hydraulic Control (EHC) for the turbine, use of the standard partial arc steam valve positioning, and the development of an optimum fan power control strategy.

The selection of the EHC option for the turbine is based on analysis of the options provided by the turbine manufacturers. Turbine manufacturers (including GE) normally offer at least two alternate control schemes for their steam turbines. The simpler, usually called Mechanical Hydraulic Control (MHC) uses turbine lube oil as the hydraulic working fluid for the control valves, and, as a rule, employs mechanical linkages and governors to control turbine speed and load. The more sophisticated Electro-Hydraulic Control (EHC) relies on a separate high pressure hydraulic system to control steam inlet valve position, and includes use of electronics to control turbine speed and load.

The EHC system is recommended for the pilot plant for two principal reasons. First, the electronic control approach can be more readily adapted to any interface requirements remote from the EPGs such as those associated with the receiver and the TSS. Secondly, since EHC control systems are currently being ordered with most larger turbines, inclusion of EHC for the pilot plant would permit a better simulation of a commercial facility.

The recommendation to use partial arc steam valve positioning is based on thermal stress considerations and the size of the pilot plant turbine. For efficiency, turbine control valves can be arranged to open sequentially, admitting steam to increasingly larger segments of the turbine first stage nozzles. This sequencing keeps the first stage blading near its optimum operating point, and reduces the throttling losses at turbine partial loads. This sequencing of control valves is referred to as partial arc positioning and is contrasted to full arc positioning where all turbine control valves move in parallel, admitting equal amounts of steam around the periphery of the turbine first stage.

For the type of 12.5 MWe turbine to be supplied for the pilot plant, GE normally provides partial arc positioning, and no change to full arc positioning is recommended. GE advised that for larger turbines, such as would be used for a commercial solar plant, the daily cycling of a unit with partial arc positioning would induce sufficient additional thermal stresses in the turbine casing to cause concern. Under these conditions, full arc starting of the unit with transition to partial arc operation after the warm-up period would be recommended.

A recommendation that an optimum fan control strategy be developed is based on the fact that fan power requirements for the dry cooled condenser can have a significant effect on plant output. This is because the maximum fan power accounts for approximately one-half of all auxiliary power requirements.

Two different methods for fan control were available from the condenser suppliers surveyed. One approach employed two speed motors for the fans; the other used variable pitch fan blades to vary fan capacity. With either approach, a strategy for obtaining maximum plant output for any given plant load and ambient dry bulb temperature could be developed.

With two speed motors, the speeds can be switched sequentially to achieve the desired control range. With variable pitch fans, even more precise control would be possible, although the added expense of the variable pitch equipment would have to be evaluated against the possible increases on plant electrical generation.

- d. Feedwater Heater Optimization - Feedwater heater train performance was studied to find the preferred number of stages, and optimum surface areas. Performance calculations were performed using the modified turbine cycle heat balance program. Performance gains and losses were traded against costs in the EPGS, the TSS, and the collector field. Feedwater heater cost differentials were based on data supplied by two feedwater heater manufacturers.

Specific recommendations included the following:

- Use of four stages of feedwater heating
- Sizing of heater surface areas to produce the terminal temperature differences and drain cooler approach temperatures indicated in Figures VII-11 and VII-12

- The continued use of extraction steam to heat feedwater during TSS operations

Computer performance calculations and cost data were used to compare three, four and five stages of feedwater heating. Results indicated that the conceptual design selection of four stages was appropriate, and should not be changed.

Heater surface areas were then examined in a separate study using similar techniques to determine optimum terminal temperature differences and drain cooler approach temperatures.

Feedwater heater terminal temperature difference is a measure of heater overall performance, and is defined as the difference in temperature between steam condensing in the heater shell and feedwater leaving the heater. For a heater with a condensing zone but no desuperheating zone, such as the lowest pressure heater, the theoretical maximum performance would be achieved with a terminal difference of 0°K (0°F). This performance is not economic in practice, however, because the heat transfer surface required tends toward infinity. For heaters with desuperheating zones, such as the higher pressure heaters, negative terminal differences are possible.

Drain cooler approach temperature is defined as the difference in temperature between drains (condensate) leaving the feedwater heater shell and feedwater entering the heater. It is a measure of drain cooler performance. Approach temperature is usually limited to a minimum of 5.6°K (10°F) in practice. Lowering of approach temperatures below 5.6°K (10°F) could be achieved by use of a drain cooler external to the heater; however, results indicated that there would be no economic advantage to do so.

In addition to the above investigations covering feedwater heater design operating parameters, a study of a suggestion given by an external reviewer to eliminate feedwater heaters from the flow path during TSS power generation operations was conducted. Proposed benefits for the plan were reduced steam and feedwater flows, reduced TSS oil storage inventory, and smaller TSS heat exchangers because of larger available system temperature differences. Bechtel analysis of the proposal led to the conclusion that other effects, as delineated below, prevent the realization of the improved performance.

To analyze the proposed plan a series of turbine heat balances was computed using four, two and no feedwater heaters to preheat

the feedwater to the TSS. The conceptual design 8 in. last stage blade turbine with a nominal 20.3 kPa (6 in. HgA) condenser was used for the comparison. All balances were computed at a gross turbine-generator output of 8132 kWe with varying condenser pressures to approximate operation at 7000 kWe net with 85°F ambient dry bulb. Minor effects such as changes in pump efficiencies and line pressure drops were neglected.

A degradation of performance was found for the mode of steam use from the TSS. This degradation can be explained by considering the TSS as an external heat source supplying energy to the turbine cycle. By treating the TSS as an external heat source the usual textbook analyses of regenerative feedwater heating using extraction steam can be applied. As the feedwater heaters are removed from service, the turbine cycle gross heat rate increases from 13,500 kJ/kW-hr (12,800 Btu/kW-hr) to 15,700 kJ/kW-hr (14,900 Btu/kW-hr). This trend is in agreement with the usual "textbook" analysis of the efficiency effects of regenerative feedwater heating.

Since the poorer heat rates would significantly increase the thermal storage requirements in the TSS, the scheme was dropped from further consideration.

- e. Pump Selection Studies - Three pump optimization studies were undertaken to determine the most economic configuration of the EPGS feedwater system:

- The possible elimination of the low pressure feedwater pumps from the EPGS
- The study of two full capacity pumps vs. three half capacity pumps each for the condensate pumps, booster pumps, and high pressure boiler feedwater pumps
- A comparison of centrifugal vs. reciprocating pumps

The first study was directed at the possibility that plant complexity, and thus overall plant costs, could be minimized by reducing the number of EPGS pumps in series. The second study was based on the consideration that larger power plants often utilize three pumps in parallel for each service, with less than full capacity in each pump. These two alternatives were analyzed on the assumption of equal reliability, i.e.,

either system could function with a single pump out of service. Finally, reciprocating pumps were evaluated for possible reductions in capital cost and pumping power requirements.

The study results indicated that eliminating the low pressure feedwater pumps, while compensating by raising the design head on the booster pumps, could achieve a simplification of the EPGS without any significant cost impacts. However, the cost benefit or lack of it depends on the pump manufacturer's available off-the-shelf selections. For this reason, the recommendation to eliminate the low pressure feed pump for the preliminary design also included an additional recommendation that competitive cost bids for configurations with and without the low pressure feedwater pump be obtained during the final design phase. This will give additional flexibility to each pump manufacturer in determining the best available pump arrangement.

The study of two full capacity vs. three half-capacity pump trains showed the two train arrangement to be the most cost-effective. The efficiency of the full-capacity pumps was found to be better at the ERDA 10 MWe and 7 MWe design point operating conditions. In addition, the unit cost for pumps in the general size range was found to be about the same, giving a definite advantage to the full capacity pumps.

On the basis of preliminary design results obtained from one pump manufacturer for reciprocating and centrifugal high pressure boiler feedwater pumps, centrifugal pumps are recommended for the EPGS feedwater system in lieu of reciprocating pumps.

One major reason for selecting centrifugal pumps is that centrifugal pumps would likely be installed in a 100 MWe, scaled-up version of the 10 MWe solar pilot plant. Selection of centrifugal pumps over reciprocating pumps for the pilot plant would thus allow better prediction of EPGS feedwater system operating characteristics for the larger plant.

Additional disadvantages of reciprocating pumps cited by the manufacturer were: approximately 10 percent higher pump and motor costs, a pump horsepower requirement of approximately 600 kW (800 hp) as compared to approximately 300 kW (400 hp) for a centrifugal pump at high pressure feed pump rating conditions, and a size of approximately 2 m (6 ft) width x 3 m (10 ft) height x 9 m (24 ft) length, as compared to 0.6 m (2 ft) x 1 m (3 ft) x 2 m (8 ft) for a centrifugal pump. In addition, a greater maintenance requirement is expected.

The high pressure feed pumps were selected for this analysis because they represent the largest capital cost and require the greatest pump power in the cycle. Although condensate pumps and booster pumps were not considered in the study, the results are considered applicable to the entire cycle.

- f. Pipe Sizing Studies - Parametric studies of riser (feedwater) and downcomer (steam) line sizes versus cost and pressure losses indicated a potential for cost reductions.

Reductions in pipe size from the conceptual design nominal 8 inch diameter downcomer and 4 inch diameter riser can result in cost savings in the pipe. Potential increases in TSS and receiver costs may result from pressure loss increases resulting from pipe size reductions. MMC is evaluating required changes to accommodate these pressure losses in the TSS and the receiver. If significant adverse costs are found, the conceptual design diameters for either or both pipe runs may be retained.

The feasibility of incorporating bellows-type expansion joints in place of the expansion loops to accommodate tower and field run pipe thermal expansion was evaluated. A bellows manufacturer provided a design package, quoting a price which results in a significant saving for the downcomer steam line due to reduction of line length in expansion loops. However, the pressure and temperature levels required by the pilot plant design are higher than any existing steam line systems with installation of similar expansion joints.

A steam plant with 12 inch diameter piping at the Dallas-Fort Worth Airport has employed similar bellows in a 6300 kPa, 673°K (900 psi, 750°F) steam system. The design quoted by the manufacturer was specifically for the 12 inch diameter, 8894 kPa, 783°K (1275 psig, 950°F) pilot plant service. Since the conceptual design's temperatures and pressures are higher than those used in state-of-the-art power plant design, use of bellows has not been recommended for use in the pilot plant. Since piping costs may prove to be the limiting parameter controlling maximum feasible commercial plant size, the significant potential cost reductions for multiple module solar installations does provide an incentive for further consideration of bellows-type joints during commercial plant design.

In the preliminary design effort the above activities will be reviewed and considered further. The potential for reduced

diameter steam pipe lines mentioned above may enhance the feasibility of bellows-type expansion joints. The utilization of expansion joints exclusively in the riser (feedwater) piping at 13,380 kPa, 539^oK (1925 psig, 510^oF) will be considered.

- g. Auxiliary Steam Supply Studies - The conceptual design for the pilot plant includes a special heat exchanger in the TSS to provide steam during the usual daily shutdowns. The steam was to be used for turbine steam seals, steam tracing of TSS molten salt piping, and other miscellaneous uses. Further study indicated that the special heat exchanger should be eliminated for both operational and cost reasons, and be replaced by a small fossil boiler of the same capacity.

Operationally, a separate fossil boiler would permit normal startup and operation during periods when the TSS is shut down for repair or maintenance, thus increasing plant availability. The separate boiler would also eliminate the need for a small emergency boiler to provide TSS tracing steam during long outages.

The recommendation of a fossil boiler is based solely on economic and technical considerations. The presence of a fossil boiler at a solar pilot plant site may create negative psychological impressions for some observers. If this psychological effect is considered to be of overriding importance, use of the TSS heat exchanger or an electric boiler may be justified.

I RECEIVER TOWER STRUCTURAL OPTIMIZATION STUDY

1. Introduction

In the conceptual design a Foster Wheeler designed steel receiver support structure was shown mounted on top of the Bechtel concrete tower. Seismic loads into the receiver, which is hung from the support structure, were difficult to accommodate due to multiplication of driving loads in the steel support structure. Two versions of concrete receiver towers extending to the top of the receiver were considered by Bechtel as a method of eliminating the steel support structure and reducing seismic loads to the receiver.

In the all-concrete concepts the receiver boiler would be hung from steel beams so that it is within the concrete envelope of the tower. Advantages of this concept include the elimination

of the large supporting concrete platform for the steel superstructure at the interface with the concrete platform, improved seismic performance of the tower receiver combination, easier maintenance, better containment of wind or seismically damaged receiver components, and greater personnel safety.

2. Description of Structural Systems

The four cases studied are illustrated in Figure VII-13.

- Case A is a tapered tower similar in exterior configuration to that used in the conceptual design, with an enlarged concrete cap to accommodate the new footprint of the steel structure as provided by Foster Wheeler Energy Corporation (FWEC). The thickness of the concrete in the tower and the foundation was increased to allow for the increased gravity and seismic loads of the receiver at 3.34 MN vs. 2.67 MN (750 kips vs. 600 kips) and the steel framing at 6.23 MN vs. 1.11 MN (1400 kips vs. 250 kips). The top elevation of the cap is 132.7 m (432 ft)
- Case B is a cylindrical tower with a top configuration based on the clearance and opening requirements provided by FWEC. The top elevation of the concrete was set at 155.4 m (510 ft) to allow for the 0.914 m (3 ft) depth of steel framing. It should be noted that the top of the cylinder is open. FWEC would furnish and erect the steel framing at the top to support the boiler, snubbers, and any planes of bracing required to transmit the snubber forces to the concrete cylinder.
- Case C is similar to Case B in shape with tower top plan dimensions similar to that of Case A, utilizing a thinner cap. The cap is thinner due to the shorter length of cantilever.
- Case D is a conical tower with a top configuration similar to that in Case B, but with the base spread to 36.58 m (120 ft) diameter. Seismic analyses have been run for this case using a Bechtel lumped mass approach similar to that used in the conceptual design.

CASE A

CASE B

CASE C

CASE D
24.69M
(81'-5" O.D.)

TOWER TOP RECEIVER SUPPORT CONCEPTS

FIGURE-VII-13

3. Comparison of Structural Systems

The advantages and disadvantages of the above four concepts are compared in Table VII-6. Actual site soil and seismic conditions as well as design criteria such as stray insolation flux protection may have significant impacts on actual cost. A detailed analysis will be performed during the preliminary design to provide a more reliable comparison.

4. Receiver

A 155.4 m (510 ft) receiver top support height and a 2.67 MN (750 kips) receiver weight were specified by FWEC and MMC. The lateral sway of the tower was to be limited to less than ± 0.53 m (1.75 ft) at the center of the receiver aperture. The peak ground acceleration due to earthquakes was assumed to be 0.2g. The governing lateral force was due to earthquakes and not wind. The design soil pressure was assumed to be 192 kPa (4000 psf). The concrete strength was assumed to be 41.4 MPa (6000 psi) at 29 days similar to that used in the conceptual design.

5. Receiver Seismic Loads

Receiver lateral loads obtained from FWEC were used in estimating the total number of snubbers required for a 2.22 MN (500 kips) capacity snubber. The total number of snubbers required was estimated to be approximately 24 for horizontal seismic forces and 8 for vertical seismic forces.

6. Analysis and Results

The analytical modeling techniques used are similar to those used for the conceptual design tower. The tower was broken into 13 lumped masses or nodes. Five of the node points were selected to correspond with the elevations where receiver support attachments in the horizontal or vertical planes would be required, and acceleration time histories and response spectra were calculated using the standard STARDYNE analysis program.

J. CONCEPTUAL PLANT COST

A conceptual cost estimate was performed for the EPGS and all other BOP within the Bechtel scope as well as for the Georgia Tech designed TSS. These costs were incorporated into the overall Pilot Plant cost estimate by MMC and are, therefore, not repeated here. Further cost estimates for the Pilot Plant preliminary design and for a commercial plant concept will be performed prior to contract completion.

TABLE VII-6

TOWER TOP RECEIVER SUPPORT CONCEPTS COMPARATIVE FEATURES

	CASES			
	A	B	C	D
● COST	HIGH	LOWEST	HIGH	HIGHEST
● PEAK ACCELERATION IN RESPONSE SPECTRA FOR BOILER DESIGN	HIGH	LOW	HIGH	LOWEST
● CONTAINMENT OF COMPONENTS IN EARTHQUAKES	LOW	HIGH	LOW	HIGH
● CONVECTIVE COOLING LOSSES	HIGH	LOW	HIGH	LOW
● MAINTENANCE CONVENIENCE DURING OPERATION	LOW	HIGH	LOW	HIGH
● STRUCTURAL DESIGN CONTINUITY	LOW	HIGH	LOW	HIGH
● PERSONNEL HAZARD FROM STRAY INSOLATION DURING OPERATION	HIGH	LOW	HIGH	LOW
● EASE OF RECEIVER ASSEMBLY	HIGH	LOWER	HIGH	LOWER