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CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM, PHASE 1

Progress Report for Period Ending December 31, 1975

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April 1976

Work Performed Under Contract E(04-3)-1110

Martin Marietta Corporation
Denver, Colorado

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Division of Solar Energy

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Summary
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I. INTRODUCTION

A. PROGRAM SCOPE AND ORGANIZATIONAL TEAM

The Central Receiver Solar Thermal Power System, Phase 1, program is authorized by Energy Research and Development Administration (ERDA) contract E(04-3)-1110, and has a period of performance from June 15, 1975 to June 14, 1977. Mr. George M. Kaplan, ERDA, Solar Energy Division, is Program Director. The program objective is the preliminary design of a 10 MWe pilot solar power plant, shown conceptually in Figure I-1, supported by major subsystem experiments.

Primary importance has been placed on demonstrating features of envisioned large scale commercial solar plants within the pilot plant. Subdivision of the design and the program work breakdown task elements has been by major subsystem.

A team of four organizations led by Martin Marietta Aerospace, Denver, Colorado, have combined forces to perform the program. Martin Marietta is the integrator for the overall effort, collector subsystem designer, and will build the collector experiment. Bechtel Corporation, San Francisco, California, is responsible for the electric power generation subsystem and the architect-engineer tasks of the pilot plant. Foster Wheeler Energy Corporation, Livingston, New Jersey, will perform pilot plant receiver design and build a 5 MWth receiver for the receiver subsystem research experiment to be tested at Sandia. Georgia Institute of Technology has responsibility for the pilot plant thermal storage design and the thermal storage subsystem experiment to be performed in association with the Georgia Power Company.

B. PROGRAM PLAN

This project covers a 24-month effort to accomplish the preliminary design and program definition of a 10-MWe central receiver pilot plant and to perform a series of experiments on three key subsystems, collector, receiver, and thermal storage. The objectives of these efforts are to:

- Establish the technical feasibility of a solar thermal power plant of a central receiver type with significant commercial potential;
- Obtain sufficient development, production, and operating data to indicate the economics of operation for commercial plants of similar design.

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

The three subsystem experiment activities are to provide a data base for preliminary design of the central receiver pilot plant. The pilot plant is itself to be directly related to a potentially attractive commercial central receiver power plant concept. Primary program outputs will be the Pilot Plant Preliminary Design Report, including system and subsystem specifications, subsystem research experiment documentation and hardware, and computer simulation documentation.

There are basically four significant stages of work. These are: 1) pilot plant conceptual design stage; 2) pilot plant system and electrical power generation subsystem optimization stage; 3) subsystem research experiment stage; and 4) pilot plant preliminary design stage.

The principal objectives of the first stage are to develop sufficient conceptual designs of the pilot plant system and subsystems to justify the design approach, provide a framework within which to identify research experiment requirements for components of the collector, receiver, and thermal storage subsystems. Details will be provided to the extent needed to verify the technical feasibility of the approach, broadly define functional characteristics of the systems, and provide sufficient physical and performance data upon which to base system criteria for succeeding design activities.

The principal purpose of the system and electrical power generation subsystem analysis optimization stage is to perform trade-offs, evaluations, and refinements of the baseline configuration and, in parallel with on-going subsystem research experiments, to achieve a more optimized plant concept, as the basis for proceeding effectively with the preliminary design effort.

The preliminary design stage is aimed at bringing together the results of conceptual design work, subsystem research efforts, EPG subsystem optimization study, computer simulation study, and other pertinent program studies into an integrated preliminary pilot plant design.

Figure I-2 presents the master program schedule and is broken down to show each of the major tasks. The primary efforts through the first six months of the program have dealt with the development of the pilot plant preliminary baseline design, the conceptual design of the three research experiments and initiation of the research experiment detailed design. These activities are discussed in detail in subsequent sections of this report.

Figure I-2 Master Program Schedule

As previously discussed, the effort on this program is being conducted by a team of contractors consisting of Martin Marietta Corporation, Bechtel Corporation, Foster Wheeler Energy Corporation,, and Georgia Institute of Technology. Figure I-3 shows the major work efforts broken down by the Work Breakdown Structure and which organization has the primary responsibility for each major task. Martin Marietta, as the prime contractor, has the overall responsibility for the total program direction.

C. PILOT PLANT OVERVIEW

The simplified schematic of Figure I-4 provides an overview of the major subsystems, including the master control, that make up the pilot plant.

The collector subsystem consists of an array of controlled heliostats to continuously reflect the solar rays to the receiver subsystem. The array of heliostats are capable of being mounted on either a flat or terraced field facing south. Each heliostat is capable of fully independent and automatic solar tracking operations. Heliostats respond to executive control by a master control system which overrides the normal fine-pointing system.

The receiver provides a means of transferring the incident radiant flux energy from the collector subsystem into a steam working fluid for: 1) generating electric power with a conventional prime mover located within the electrical power generation subsystem (EPGS); 2) conversion to stored thermal energy by the thermal storage subsystem (TSS); and 3) generating electric power with a conventional turbine/generator while also charging the TSS. The receiver subsystem consists of a heat exchanger cavity that intercepts the reflected radiant flux from the collector subsystem, and support equipment which regulates the fluid flow, temperature, and pressure to provide adequate thermal control and maximize receiver efficiency.

The thermal storage subsystem provides a thermal storage medium and necessary equipment to deliver and accept the working fluid to and from the receiver subsystem and EPGS. The thermal energy stored within the TSS is used for generating both saturated and superheated steam. The saturated steam shall be used by the EPGS to maintain proper sealing while superheated steam is required for thermal management of components, to maintain required steam input to the EPGS turbine during transient adverse weather conditions and evening operation periods, and to provide

Figure I-3 Primary Roles of Phase 1 Team

Figure I-4 Schematic of 10-MWe Pilot Plant

thermal control to maximize plant efficiency.

The electrical power generation subsystem provides the means for converting the thermal output of the working fluid delivered from the receiver at the base of the tower and/or the TSS into electrical power output from the generator. The output from the electrical power generator is regulated and transformed for integration into existing electrical power system networks. The subsystem also provides for the conditioning of the working fluid required to maintain proper plant efficiency and for delivery of the working fluid as suitable feedwater returns to the receiver and/or the TSS.

Ground rules were established for the operating conditions of the pilot plant. Based on these ground rules, subsystem performance requirements and operational limits were developed to utilize the maximum performance capability of the various subsystems and still provide an economical pilot plant.

Having fixed the performance requirements of the receiver, electrical power generation and thermal storage subsystems, the collector subsystem size is optimized and minimized at the design point to stay within these ground rules. To meet these requirements, the collector subsystem must contain 1718 heliostats that have 37.2 m^2 (400 ft^2) of reflective surface to satisfy the design point of 10 MWe net at 2:00 p.m. on Winter Solstice. The pilot plant performance point chart of Figure I-5 identifies the power in megawatts required at each significant point in the system where performance can be accounted for. In addition, the inset table identifies system efficiency for each respective area of the system from the point at which the solar insolation is received at the heliostat until it is delivered as electrical power.

The "potential power to system" is that power available if all heliostats were pointed such that their surfaces were perpendicular to the sun's rays. The pilot plant performance calculations start with the turbine heat rate of $13.09 \times 10^6 \text{ J/kW-hr}$ ($12,412 \text{ BTU/kW-hr}$) required to produce the specified 10 MWe, while providing the required auxiliary electrical loads necessary for power plant operation. From this point, all losses are accounted for in the collector subsystem, receiver subsystem, and the interfaces between those subsystems and the electrical power generation subsystem. After determining the required "potential power to system", the number of heliostats required is determined based on the solar insolation at the design point.

Figure I-5 Pilot Plant Performance-Design Point

For purposes of the pilot plant design, the solar insolation is 0.8255 kW/m^2 . This 2:00 p.m. insolation value was developed by averaging several "clear days" around winter solstice for Inyokern in 1963.

D. 100 MWe POWER PLANT REFERENCE DESIGN

Fundamental design features of the energy collection, receiver, and electric power generation subsystems for 100 MWe size solar power plants were established during the Solar Power System and Component Research Program (NSF Grant AER 75-07570).

Key among these were modularization of the collector into fields of 1600 - 2000 heliostats located north of the receiver tower, as illustrated in the plant layout section of Figure I-6.

Minimization of land usage resulted from terracing the receiver field as shown in the geometric model. Focusing heliostats enabled use of a minimum size receiver aperture. Basic to the attainment of high thermal performance is the use of a cavity receiver with its aperture facing the north and tilted to present a balanced view of the collector. Air cooling of the electric power generation system was considered essential to the broad application of large solar power systems in areas already short of water.

For the current program, the Central Receiver Solar Thermal Power System baseline reference plant is to be capable of "intermediate load" operation which includes operation after dark for periods up to six hours following solar operation. Shown in Figure I-7 is the layout for such a 100 MWe intermediate load solar plant. The number of collector modules has grown from 8 to 14 to accommodate the stored energy requirements and a sensible heat thermal energy storage subsystem has been integrated into the plant. Dimensions shown reflect use of flat terrain for the plant. The collector concept is adaptable to terrains ranging between flat and general southerly slopes up to 13° .

The 100 MWe plant conceptual configuration consists of 14 solar collector modules, the main plant site and the thermal storage site. Each site is 619.3 m (2032 ft) north to south and 604.7 m (1984 ft) east to west for a total area of $3.745 \times 10^5 \text{ m}^2$ ($4.031 \times 10^6 \text{ ft}^2 = 92.5 \text{ acres}$). The collector modules each contain 1665 heliostats in accordance with the current plant sizing calculations. The number could grow to 1718 per site within the current dimensions and be compatible with the pilot plant collector layout.

AIR-COOLED PILOT PLANT

GEOMETRIC MODEL OF MIRROR FIELD

**1/32 SCALE MODEL STEAM
GENERATOR CAVITY**

**PLANT LAYOUT AND EXAMPLE SUBSYSTEM
CONFIGURATIONS—100-MWe SOLAR ENERGY
CONVERSION POWER PLANT**

Figure I-6

*Plant Layout and Example Subsystem Configurations--100-MWe Solar
Energy Conversion Power Plant*

Collector Subsystem
Modules (1665 Helio-
stats/Module)

Figure I-7
Field Layout - 100 MWe Solar Plant - with Temperature
Storage Cycle, 4137 kPa, 672 K (600 psi, 750°F)

Thermal storage tankage is scaled up from the pilot plant as follows:

Stage 1 Hitec	12 Tanks 6.1 m (20 ft) dia. x 9.1 m (30 ft) height.
Stage 2 Thermia (Oil)	21 Tanks 18.3 m (60 ft) dia. x 13.7 m (45 ft) height.
Stage 3 Thermia (Oil)	3 Tanks 18.3 m (60 ft) dia. x 12.8 m (42 ft) height.

Site area required for the tankage is considerably below that of the module allocated, approximately $31,000 \text{ m}^2$ (7.6 acres).

Sizing of conceptual 100 MWe solar power plant is compatible with the guidelines of the CRSTPS 10 MW preliminary design program. The environmental conditions controlling the 100 MWe conceptual configuration definition are based on the winter solstice and Inyokern (1963) data. Two major segments of the solar collector subsystem were sized separately and combined into the configuration. For daytime operation of the power plant at full load, the 2 p.m. insolation of 0.825 kW/m^2 was used to size one collector segment. To enable the six hour operation at 70% load from storage, the average insolation for the six hour charging period (9 a.m. to 3 p.m.) of 0.852 kW/m^2 was used to size the second collector segment.

The heat balance used to establish the collector sizing for the daytime 100 MWe operation is based on a gross fuel rate for a GE turbine generator of $9.58 \times 10^6 \text{ J/kW-hr}$ (9077 BTU/kW-hr) modified for the air cooling back pressure of 15.24 cm (6 in.) Hg to $10.05 \times 10^6 \text{ J/kW-hr}$ (9530 BTU/kW-hr).

Plant auxiliary equipment increased the generated power requirement to 110.3 MW, resulting in a plant net heat rate of $11.09 \times 10^6 \text{ J/kW-hr}$ (10,518 BTU/kW-hr). Combined thermal efficiency for the heat source (solar collector and receiver) is 0.773. The required 398.5 MW of thermal energy input to the system is provided by 13,000 heliostats with 0.91 reflectivity surfaces.

Sizing of the collector field segment required to provide energy to the storage system sufficient to enable after dark operation of 6 hours at 70% load is based on a net heat rate of $12.92 \times 10^6 \text{ J/kW-hr}$ (12,248 BTU/kW-hr) when operating from storage supplied steam at 4137 kPa, 672°K (600 psi, 750°F).

For the December solstice operation on which the sizing is based, the daylight operating period is six hours. This results in a

1 to 1 charging time/discharging time ratio. An element of the heat balance unique to the storage is the storage tank insulation loss factor of 0.982. Combined thermal efficiency of the storage segment heat balance is 0.769. The required 326 MW of thermal energy input to the system is provided by 10,308 heliostats with 0.91 reflectivity surfaces each. Total heliostats for the 100 MWe plant is 23,308.

II. 10 MWe PILOT PLANT

A. DESIGN AND OPERATION

1. Pilot Plant Design Requirements

The overall configuration of the pilot plant was discussed in Section I as shown in Figure I-4. Table II-1 lists some of the more significant pilot plant performance requirements

Table II-1 System Performance Requirements

Capable of producing 10 MWe net electrical power when operating from receiver subsystem at design point (incident solar flux 0.8255 kW/m^2 .)

Capable of producing 7 MWe net electrical power when operating from the thermal storage subsystem.

Capable of storing sufficient thermal energy for operation at 7 MWe for 6 hr.

Capable of producing 7 MWe net electrical power when operating from both the receiver subsystem and the thermal storage subsystem.

Total plant lifetime requirement: 30 yrs.

Scheduled plant maintenance: 25 days/yr maximum.

Annual net output: 34,000 MWe-hr

Available for generation of electrical power 90% of the planned time, exclusive of weather conditions and planned plant maintenance.

Pressure and temperature drops between the receiver and the turbine are not to exceed 345 kPa (50 psi) and 5.6 K (10°F), respectively.

Life Cycle Requirements:

Diurnal Type	13,000
Cold Start	1,300.

Time to Full Power:

Diurnal Start from Hot Standby	- 20 Minutes
Cold Start	- 6 Hours.

Rate of Change of Incident Flux - Cloud Passage at 20 Meter/Sec.

The requirement to produce 10 MWe net power when operating at the design point of a clear day at 2:00 p.m. on winter solstice using the 1963 Inyokern data requires special consideration since winter solstice at Inyokern for 1963 was not a clear day. In order to meet the intent of the requirement, a week's worth of insolation data around winter solstice was evaluated. Figure II-1 shows plots of these days. From these data, an average insolation value of 0.8255 kW/m^2 was established as shown in Table II-2. In addition to storing sufficient energy to produce 7 MWe electrical power for 6 hours, as shown in Table II-1, the thermal storage subsystem will be capable of producing a lower power level for a longer period of time.

Additional ground rules for pilot plant performance were established for the operating conditions. Based on these ground rules, subsystem performance requirements and operational limits were developed to utilize the maximum performance capability of the various subsystems and still provide an economical pilot plant.

The main steam conditions of 783 K (950°F) and 8791 kPa (1275 psig) permit the use of a standard size turbine for the cyclic operation that does not require extensive development. A turbine was selected which best meets both the cyclic and power requirements. The selected turbine is capable of providing in excess of the required 10 MWe bus bar when additional steam flow rates are available.

At the design point, the pilot plant is required to produce 10 MWe at 2:00 p.m. on winter solstice. Sizing the heliostat field for this design point provides additional capability from the collector subsystem around noon due to both an increase in solar insolation and improved cosines of the heliostat field for reflecting the insolation. In order to utilize the full output of the collector subsystem throughout the day, the receiver subsystem is sized to accept maximum solar insolation of 1.02 kW/m^2 throughout the day. Under these conditions, all available solar power from the collector subsystem will be provided for the generation of electrical power or for storage for later use. The selected turbine for the electrical power generation subsystem is capable of generating more than the 10 MWe required at the 2:00 p.m. design point.

The thermal storage subsystem is sized to accept the flowrates generated by the receiver on a typical day with a maximum insolation of 0.9 kW/m^2 . Thus, the thermal storage subsystem is capable of delivering flowrates in excess of those required to meet the specified requirement of 7 MWe net when the electrical

Figure II-1 Direct Insolation for Inyokern, California - Week Following Winter Solstice, 1963

Table II-2 Inyokern, California, 1963, 2 p.m.,
Direct Insolation Data for Clear
Days Near Winter Solstice

Date, 1963	2 p.m. Direct Insolation, kW/m^2	Weather Status
December 15	0.81	Clear Day
December 17	0.82	Clear Day
December 19	0.81	Clear Day
December 20	0.82	Clear Afternoon
December 22*	0.855	Clear Afternoon
December 23	0.840	Clear
December 24	0.85	Clear
December 25	0.845	Clear Afternoon
December 28	0.78	Clear
Mean	0.825	
Standard Deviation	0.024	

*Winter Solstice Date for 1963.

power generation subsystem is operating from storage alone. Since the selected turbine is capable of providing more than 7 MWe on storage alone, the additional flowrate capability provided by the thermal storage subsystem can be utilized.

Having fixed the performance requirements of the receiver, electrical power generation and thermal storage subsystems, the collector subsystem size is optimized and minimized at the design point to stay within these ground rules. The steam conditions to the turbine will be maintained to provide the specified net electrical power output. As subsystems are optimized and efficiencies are increased, the heliostat field will be reduced in size while meeting these ground rules.

2. Pilot Plant Operation

The system schematic for the pilot plant shown in Figure II-2 is a further detailed breakdown of the Simplified Schematic of the 10 MWe pilot plant shown earlier. 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.

Detail design descriptions of the subsystems are provided in later sections of this report.

a. Collector Subsystem Operation - The collector subsystem, not shown in Figure II-2, consists of an array of controlled heliostats located in a north field to continuously reflect the solar rays to the receiver subsystem. The heliostats are capable of being mounted on either a flat or terraced field and are situated so they are facing south. Each heliostat is capable of fully independent and automatic solar tracking operations. The heliostats are capable of responding at all times to

Figure II-2 Pilot Plant System Schematic

executive control by a master control subsystem which will either supplement or directly override the normal fine-pointing system.

b. Receiver Subsystem Operation - The boiler and superheater 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 high pressure boiler feedwater pump. Feedwater is provided to the suction of this pump from either the thermal storage subsystem, the turbine/condenser, or both. 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 superheater to provide the required steam conditions. 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 attenuator for temperature control of the superheated steam. The receiver provides the necessary controls for regulation of feedwater flow, attenuator 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.

c. 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.

When the subsystem is being charged (energy extracted from the steam), the steam coming from the receiver is diverted at the base of the tower to the thermal storage subsystem and into the

first desuperheater. At this point, the steam is desuperheated, as required, with feedwater to assure protection of the Hitec from overtemperature in the first stage heat exchanger. In this first stage, most of the superheat is removed from the steam and is stored as sensible heat in the Hitec. This is accomplished by the Hitec flowing from the cold tanks counter-current to the steam and going to the hot tanks. During this charge mode, the flow of Hitec is controlled by the steam exit temperature from the heat exchanger. The exit from this first stage is sent to a spray desuperheater to assure that the temperature of the superheated steam entering the condenser will not cause the oil to overheat. Feedwater is provided to this desuperheater also, and the feedwater is controlled by the exit temperature from the desuperheater.

In the second stage, the steam enters a condensing heat exchanger where the latent heat is removed and stored as sensible heat in a hydrocarbon oil. The steam pressure in the condensing heat exchanger controls the rate of flow of oil which, in turn, controls the rate of condensation. The water exiting the condenser is slightly subcooled and is further subcooled in the last heat exchanger. The exit water from the subcooler heat exchanger is provided as feedwater to the suction of the high pressure boiler feedwater pump for return to the receiver steam drum via the tower.

When the subsystem is being discharged (producing superheated steam for the generation of electrical power) feedwater is provided from the low pressure boiler feedwater pump and flows in the opposite direction to the charge mode. The feedwater enters the heat exchanger and is preheated prior to entering the next heat exchanger which now becomes an evaporator heat exchanger. Oil is now flowing from the hot tanks through the evaporator, then to the preheater and then to the cold tank. The flow of oil (which can be compared to the firing rate of a boiler) is controlled by the pressure of the superheated steam leaving the thermal storage subsystem. The flow of feedwater into the thermal storage subsystem is controlled by a three element feedwater regulator similar to those used in conventional boilers. The steam is separated and enters the next heat exchanger which now serves the function of a superheater. The flow of Hitec from the hot tank to the cold tank is controlled by the exit temperature of the superheated steam. This superheated steam is sent to the admission steam point on the turbine.

d. 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 high pressure, high temperature main steam from the receiver, or lower pressure and temperature admission steam from the thermal storage. The subsystem contains four stages of feedwater heating using uncontrolled extraction from the turbine.

As shown in Figure II-2, superheated steam is provided down the tower from the receiver to the turbine main steam throttle valve for conversion to electrical power. A main steam bypass valve is provided to divert the superheated steam to the condenser in the event of overpressure of the main steam. Desuperheaters are provided at the condenser to accommodate the superheated steam. After leaving the turbine and the condenser, the condensate is pumped through the feedwater heaters, increased in pressure, and returned to the receiver or to the thermal storage subsystem or both, depending on the mode of operation. For the mode of operation where the electrical power generation subsystem is generating from the receiver alone and the thermal storage subsystem is neither charging or discharging, all feedwater is returned to the receiver from the high pressure boiler feedwater pump.

When the lower pressure and temperature superheated steam is being provided by the thermal storage subsystem, it is accepted by the turbine admission steam throttle valve. An admission steam bypass valve is provided to divert this steam to the condenser in the event of overpressure from the thermal storage subsystem. After passing through the turbine and the condenser, the condensate is pumped through the feedwater heaters, increased in pressure and all provided back to the thermal storage subsystem by low pressure boiler feedwater pump, if the receiver is not operating. If both the receiver subsystem and the thermal storage subsystem are generating steam, the feedwater is proportionally sent to both subsystems. For the mode of operation where the receiver subsystem is supplying steam both to the electrical power generation subsystem and to the thermal storage subsystem, then all feedwater is provided back to the receiver by the high pressure boiler feedwater pump.

The predominant turbine control modes are shown in Figure II-3 for various pilot plant operating modes. For the turbine control modes, HP refers to high pressure main steam from the receiver and LP refers to low pressure admission steam from the thermal storage. On load control, the turbine flowrates are determined by the electrical load requirements. In the turbine following mode, the turbine is following the output of the

receiver by the electrical load requirements. In the turbine following mode, the turbine is following the output of the receiver by maintaining backpressure and adjusting the steam flowrates accordingly. An X indicates the possible operating mode, where a circled X indicates the preferred mode of operation to produce the maximum amount of electrical power from the available solar insolation.

	Operating from Receiver Alone	Operating from Thermal Storage Alone	Operating from Receiver and Thermal Storage	Operating from Receiver and Charge TSS
Load Control HP	X	--	--	X
Turbine ¹ Following HP	⊗ ²	--	X	⊗ ²
Load Control LP	--	X	X	--

¹Turbine follows the receiver output on back-pressure control

²Preferred mode of operation for maximum use of solar energy, electrical output subject to transients from insolation.

Figure II-3. Turbine Control Modes

e. Master Controls - The master controls provide 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.

The heliostat executive control system uses a digital computer to command the heliostat to the desired mode of operation. The computer will be used to perform other master control system functions, including collection and recording of plant data, off line performance calculations, and display, recording, and analysis of alarms. The computer may be used for additional

functions such as the control of sequencing from one mode of plant operation to another and event recording for fault analysis. Plant protective relaying and metering and control of synchronization and tie-in to the electrical power network will be accomplished using typical power plant equipment.

If a link to an area control center is required to provide for receipt of automatic load-change commands, the equipment to provide that link will be included in the master control subsystem.

The major components of the master control subsystem include the plant control and display board, protective relay and metering panels, computer, CRTs with keyboard, tape, disk, printer, and card reader/punch. The plant control and display board will include sections for each of the four subsystems--collector, receiver, thermal storage, and electrical power generation. The collector subsystem controls, for example, will include switches for controlling the following functions: addressing heliostats individually, by groups, and as a whole field, acquire and fine track, synthetic track, calibrate, operator request for heliostat position, and stop. The receiver control board will include switches for controlling: aperture door open and close, feedwater inlet shut-off valve, superheater outlet shut-off valve, feedwater inlet flow control valve, and attenuator flow control valve.

Plant protective relays will be provided according to standard power plant design practice for such functions as: generator overcurrent, negative phase sequence, reverse power, under frequency, out of step, loss of field, transformer overcurrent, fault pressure, switchgear undervoltage, instantaneous overcurrent, etc.

It is expected that a minicomputer will be capable of meeting the pilot plant requirements. Computer requirements will include: a real-time operating system with priority interrupt capability, real-time clock, interval timer, hardware multiply/divide, floating point hardware, memory parity, and memory protect features. The computer peripherals will provide the capability for convenient use of the computer for off-line performance computations in addition to its role as a real time controller.

B. PERFORMANCE AND ANALYSIS

The design point for the pilot plant when generating electrical power from the receiver is 2:00 p.m. on winter solstice at Inyokern, California. With a solar plant, the solar 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 solar insolation as well as having direct effects on subsystem operation. The combined effects of the variation in solar insolation with 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. 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 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 the southwest. 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.

1. Performance Factors for the Pilot Plant

The major pilot plant performance factors that affect the electrical output have been identified and analyzed. These sources

of loss in performance include those caused by pilot plant configuration, environment, design, and manufacturing. The losses start with the collection of the solar insolation and end with the generation of electrical power. Since the electrical power generation subsystem hardware used is restricted to those commercially available; to the maximum extent possible, the sources of loss have been separated into those associated with the collector subsystem and the receiver subsystem; and the designed performance of the electrical power generation subsystem has been used.

a. Collector Subsystem Loss Analysis Summary - The losses associated with the collector subsystem are shown in Table II-3. Those loss sources are discussed below.

- 1) Heliostat Utilization - It is assumed, for purposes of sizing the heliostat field that 0.6% of the heliostats will be down for maintenance or calibration at any given time. As this conservative value is revised, the number of heliostats in the field will decrease.
- 2) Tower Shadowing - This is the power loss resulting from the shadow of the receiver tower moving across the heliostat field. The efficiency factor shown in Table II-3 is for the 2:00 p.m. winter solstice design point. Figure II-4 shows the receiver tower shadow position for each hour of the winter solstice day between 0730 and 1130. The afternoon hours from 1230 to 1630 will be the mirror image of the morning hours shown since the field is symmetrical. The shadow path for an equinox day is also shown for the line that traces the top edge of the tower across the field. The tower shadow for summer solstice does not impinge on the north field.

Shadow projections like the one shown in Figure II-4 were used to determine the number of heliostats that will be lost due to tower shadowing. Table II-4 is a summary of these data showing a tabulation of the sun's position for each hour of the day for the 21st day of the month and the equivalent number of heliostats shaded by the receiver tower shadow. The "equivalent number" was determined by projecting the tower shadow into the heliostat field and then determining the heliostats covered by the shadow. If a heliostat was partially covered by the shadow, its loss contribution was determined as follows:

More Than Half Covered - Considered a total loss because the fine track system can no longer function.

Table II-3 Collector Subsystem Loss Analysis Summary

Error or Loss Source	Tracking Error		Efficiency Factor
	arc min	mrad	
Heliostat Utilization	--		0.994
Tower Shading			0.974
Cosine Effect:	--		0.939
Heliostat Shadowing			
Heliostat Blocking			
Off-Axis Tracking			
Reflectivity	--		0.856
Pointing and Tracking:			
Pointing	0.5	0.15	
Sensor Alignment	0.5	0.15	
Hysteresis	1.0	0.29	
Circuit Calibration	0.5	0.15	
Drive Mechanism Backlash	--	--	
Sensor Wind Loading	0.5	0.15	
Structural Deformation			
Wind Velocity = 0 - 2.2 m/s (0-5 mph)	0.5	0.15	
Wind Velocity = 2.2 - 13.4 m/s (5-30 mph)	2.5	0.73	
Foundation and Pedestal Deflection	--	--	
Total Pointing and Tracking:			
Wind Velocity = 0 - 2.2 m/s (0-5 mph)	3.5	1.02	0.988
Wind Velocity = 2.2 - 13.4 m/s (5-30 mph)	5.5	1.60	0.982
Optical Inaccuracies			
Aberration	--	--	
Tangential and Focusing Error	3.7	1.08	0.988
Total Collector Subsystem			
Wind Velocity = 0 - 2.2 m/s (0-5 mph)			0.760
Wind Velocity = 2.2 - 13.4 m/s (5-30 mph)			0.755

Table II-4 Receiver Tower Shading Effects - Based on Inyokern, California, Data

	Equivalent Number of Heliostats Shaded											
	356 Dec 22	21 Jan 21	52 Feb 21	80 Mar 21	111 Apr 21	141 May 21	172 Jun 21	202 Jul 21	233 Aug 21	264 Sep 21	294 Oct 21	325 Nov 21
0530	0	0	0	0	0	0	0	0	0	0	0	0
0630	0	0	0	0	0	0	0	0	0	0	0	0
0730	13	10.8	2	0	0	0	0	0	0	0	2	10.8
0830	48	29.2	16.5	4	0	0	0	0	0	4	16.5	29.2
0930	51.5	40.8	27.0	14.6	0.4	0	0	0	0	14.6	27.0	40.8
1030	36.7	33.8	20.7	10.6	1.3	0	0	0	1.3	10.6	20.7	33.8
1130	31.7	29.0	17.3	7.8	2.0	0	0	0	2.0	7.8	17.3	29.0
1230	31.7	29.0	17.3	7.8	2.0	0	0	0	2.0	7.8	17.3	29.0
1330	36.7	33.8	20.7	10.6	1.3	0	0	0	1.3	10.6	20.7	33.8
1430	51.5	40.8	27.0	14.6	0.4	0	0	0	0	14.6	27.0	40.8
1530	48	29.2	16.5	4	0	0	0	0	0	4	16.5	29.2
1630	13	10.8	2	0	0	0	0	0	0	0	0	0
1730	0	0	0	0	0	0	0	0	0	0	0	0
1830	0	0	0	0	0	0	0	0	0	0	0	0
Avg No. Heliostats Shaded	38.8	29.4	15.9	6.7	0.5	0	0	0	0.5	6.4	15.9	29.4
Operating Hours	9	9.5	10.5	11	12.5	13.5	14	14	12.5	11.5	10.5	9.5
Efficiency Factor	.977	.983	.991	.996	.9997	1.0	1.0	1.0	.9997	.996	.991	.983

Less Than Half Covered - The part that was shaded was summed with all other partials to get a total number.

Exactly Half Covered - Half of these were considered a total loss and the other half were summed with the partials in item 2.

The efficiency factor given at the bottom of the table was obtained by determining the average number of heliostats shaded for the nine operating hours of the day and then using the following equation:

$$\text{Efficiency Factor} = \frac{\text{Total \# Heliostats} - \text{Avg \# Heliostats Shaded}}{\text{Total \# Heliostats}}$$

- 3) Cosine Effect - The effective cosine for the collector subsystem is shown in Table II-5 for the 21st day of each month for each hour of the day. The collector field effective cosines shown are the combined effect of the cosine of the angle of solar incidence on the mirror surface, the shading of the sun from one heliostat by another, and the blocking of the reflected solar insolation leaving a heliostat and going to the receiver. Each effective cosine shown is an average for the total collector field. The values are determined using a computer program and are a function of the location of the plant site, the field layout, the location and height of the receiver aperture, the time of year, and the time of day. The cosine values are determined for the individual heliostats and the shading and blocking are separately determined. Subsequently, the values are combined and averaged. The layout of the heliostats within the field is such that the effects of blocking are negligible by design. Also, the field layout is designed such that there will be no shading with sun elevation above .524 radian (30 degrees).

The shading portion of the effective cosine is shown in Table II-6. Heliostat shading is that fraction of the mirror surface area that is shaded from the sun's rays by another heliostat. The data shown in the table is the average fraction shaded for all of the heliostats in the field for the dates and times of day shown. Shading is a function of heliostat spacing and location within the field, the location of the site, and the time of day and year. As the sun azimuth and elevation angles change, the effects of shading vary due to the field layout with the effects being more predominant in the morning hours.

Table II-5 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-6 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.

- 4) Reflectivity - The mirror reflectivity varies with the type of mirror selected. As mirrors with higher reflectivity are selected, the number of heliostats required for the collector subsystem will be reduced, thereby maintaining the required performance at the design point.
- 5) Pointing and Tracking - The pointing and tracking losses are the associated error built up from design through operation. These combined errors result in the heliostats pointing off the target during operation. The effects of different wind velocities on structural deformation was determined. A computer study was performed for the heliostat structure deformation to determine the effect of a 13.4 m/s (30 mph) wind on each mirror of the heliostat. Deflection angles and direction of rotation were obtained for each mirror. The tracking sensor is aligned with the center mirror of the heliostat; therefore, any deflection of the center mirror will cause the tracking system to be activated to drive the heliostat and bring the mirror back into alignment with the sensor. The tracking error caused by the wind force then is the average of the difference between the deflection of the center mirror and the deflections of all other mirrors. Averaging all of the individual mirror deflections results in an effective tracking error of .625 m radian (2.15 arc-minutes). To allow some margin and to include thermal distortion effects, the heliostat tracking error allocation for wind and thermal distortion was established as .727 m radian (2.5 arc-minutes) for wind speeds up to 13.4 m/s (30 mph). The total power loss due to tracking error was determined from experimental data.
- 6) Optical Inaccuracies - Optical inaccuracies consist of spherical aberration and tangential errors. Aberration is the noncircular sun image caused by the off-axis orientation of the mirror surface. Both of these factors result in an increase in the image size, and therefore, play a part in sizing the receiver aperture. The reflected energy that spills out of the receiver aperture due to tangential error is accounted for here.

The total effect of all the loss sources discussed above is a reduction in power to the receiver of 24.0% for wind velocities up to 2.2 m/s (5 mph), and 24.5% for wind velocities from 2.2 m/s (5 mph) to 13.4 m/s (30 mph).

b. Receiver Subsystem Loss Analysis Summary - The losses associated with the receiver subsystem are shown in Table II-7. The efficiency of the receiver subsystem is reduced by a loss of solar energy from the receiver cavity. In addition, losses associated with transporting the working fluid to the receiver are assigned to the receiver subsystem. These loss sources are discussed as follows.

Table II-7 Receiver Subsystem Loss Analysis Summary

<u>Loss Source</u>	<u>Efficiency Factor</u>
Radiation	0.967
Convection	
No Wind	0.983
10-mph Wind	0.968
15-mph Wind	0.963
Receiver Insulation	0.998
Tower Riser	0.995
Tower Downcomer	0.996
Total Receiver Efficiency	
No Wind	0.940
10-mph Wind	0.926
15-mph Wind	0.921

- 1) Radiation - Radiation from the receiver cavity consists of both solar radiation and infra-red radiation.
 - Solar Radiation - The solar absorptivity of the boiler and superheater are maintained as high as possible to absorb the maximum solar energy incident in the cavity. However, even at best, some of the incident solar energy will be reflected out of the cavity.
 - Infra-Red Radiation - As the cavity internal surface temperatures rise, infra-red radiation occurs. Some of this radiation is lost out the cavity aperture. This loss can be minimized by optimizing the surface emissivity and absorptivity and the cavity dimensions (L/D ratio).

The total radiation loss has been established as 3.3%, as indicated in Table II-7.

- 2) Convection - Heat loss occurs due to atmospheric air circulation in and out of the cavity. Convection can be minimized by maintaining a high emissivity of the ceramic walls to obtain a moderate wall temperature. Calculation and experimental data have shown the convection loss to be about 1.7 percent for a no wind condition and 3.2 percent and 3.7 percent for 4.5 m/s (10 mph) wind and 6.7 m/s (15 mph) wind, respectively.
- 3) Receiver Insulation - The receiver cavity is covered on all sides with 12 inch thick insulation to keep the conduction losses to a practical minimum. This practical minimum loss is expected to be less than 0.5 percent of the total energy at the aperture opening.
- 4) Tower Riser - The tower riser loss is that loss of heat associated with transporting the feedwater from the electrical power generation subsystem to the receiver.
- 5) Tower Downcomer - The tower downcomer loss is that loss associated with transporting the superheated steam from the receiver to the turbine in the electrical power generation subsystem.

2. Design Point Sizing for the Pilot Plant

The pilot plant is designed to deliver 10 MWe bus bar electricity at 2:00 p.m. on winter solstice at Inyokern, California. The design point defines the subsystem performance requirements and size of the collector subsystem but not for the receiver subsystem, 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. The receiver subsystem sizing requirements are discussed in paragraph II.B.3 and the thermal storage is discussed in II.B.4.

The energy balance chart in Figure II-5 depicts the megawatt thermal potential solar insolation required to the system in order to obtain 10 MWe at the bus bar. The various locations within the system are shown on the energy balance chart along

Figure II-5 Pilot Plant Performance - Design Point

with the megawatts of thermal power at that location for the system design point of 2:00 p.m., winter solstice. The letters on the energy balance chart are keyed to the inset table which shows the system efficiency between the two locations. These system efficiencies relate to the discussion in paragraph II.B.1 on pilot plant performance factors.

Starting with the bus bar electrical output of 10 MWe at the bottom of the energy balance chart, 10 percent of the net electrical output is allowed for auxiliary power plant requirements, therefore requiring 11 MWe gross from the turbine/generator. The gross heat rate at this power output is 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 is 52.702 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, one can determine the area of mirrors required to support the design point. As discussed earlier, a solar insolation value of 0.8255 kW/m^2 is used. Based on the design size of the heliostat, the collector field requires 1718 heliostats to support the pilot plant. Should a higher value of solar insolation be used or should any of the efficiency factors shown in the inset of Figure II-5 improve, then the number of heliostats in the field will be reduced rather than providing a higher electrical power output at the bus bar.

3. Maximum Receiver Subsystem Size Requirement

For sizing the maximum receiver output requirements, the collector field size is fixed at the 1718 heliostats required for the 2:00 p.m. design point, and all heliostats are assumed to be operational. The energy balance chart shown in Figure II-6 depicts the efficiency factors used. For this purpose, since the number of heliostats is fixed, the calculations start at the top of the energy balance chart. The 65.286 MWth potential power to the system is established by the 1718 heliostats and a solar insolation of 1.0266 kW/m^2 , which is a realistic maximum throughout the year in the southwest. The next two factors on performance are a function of time of year; tower shadow, and effective cosine. They were combined over the year to determine when they would maximize. This occurred in January and the associated value for these two factors was used. Reflectivity was kept at the same value as for the 2:00 p.m. design point since an improvement in reflectivity will be used to reduce the field size rather than increasing the power output.

Figure II-6 Pilot Plant Performance - Maximum Receiver Output

The remaining efficiency factors were maximized for minimum wind conditions. The resultant output of the receiver for these conditions is 48.820 MWth with an input to the receiver of 51.939 MWth. Based on these data and the related enthalpies, the maximum flowrate from the receiver will be 69,400 kg/h (153,000 lb/h). With this capability, all available solar insolation throughout the year will be converted into steam for use by the electrical power generation subsystem and/or the thermal storage subsystem, as required.

4. Maximum Thermal Storage Subsystem Requirements

For sizing the maximum flowrate requirements of the thermal storage subsystem, the same approach was used as for the receiver subsystem. That is, the collector subsystem of 1718 heliostats were used and the efficiency factors were maximized to provide the maximum flowrates. The energy balance chart shown in Figure II-7 shows the related efficiencies used. The differences between the calculations for the receiver and the thermal storage subsystems include the solar insolation value, the tower shadow effect, and the effective cosine. The predominant factor is the solar insolation value of 0.9 kW/m^2 which is representative of a typical day as discussed earlier. The resultant output of the receiver for these conditions is 43.10 MWth with an input of 45.853 MWth input. The related flowrate for this mode of operation is 61,236 kg/h (135,000 lb/h). The thermal storage subsystem is sized to accept this typical day flowrate. In addition, sizing to this flowrate provides capability in flowrate on discharge above the 51,936 kg/h (104,000 lb/h) required for generation of 7 MWe net electrical power output and provides the capability to operate the turbine at its maximum admission flowrate of 57,379 kg/h (126,500 lb/h) for 9.4 MWe gross electrical power output.

C. TYPICAL DAY OPERATION

In order to understand the interaction between subsystems within the pilot plant, it is desirable to establish the operation on a typical day. 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 have 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

Figure II-7 Pilot Plant Performance - Maximum Thermal Storage Subsystem Charge

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-8 for December 24, 1963, at Inyokern as a representative day for insolation near winter solstice. Without further 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 subsystem, the typical day performance profile shown in Figure II-9 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 following paragraphs describe the system operation for those modes of operation shown in the inset of Figure II-9 and the related combination of flowrates and electrical power generated as shown on the curves. For convenience in following the operational description in the following paragraphs, refer to the system schematic shown in either Figure II-2 or Figure II-10. The following paragraph numbers and titles refer to the inset on Figure II-9 as well as the circled numbers shown at the bottom of the figure.

Figure II-8 Typical Day Insolation

Figure II-9 Typical Day Performance Profile

1. Receiver Startup

This mode of operation brings the receiver up from overnight temperature and pressure to the point where steam conditions are acceptable to the thermal storage system. This mode of operation requires considerable analysis to insure acceptable stress conditions in the receiver, particularly with respect to the superheater temperature. These concerns involve the flowrates within the superheater and the impinging flux. For purposes of this discussion, it is assumed that after an initial heating period, the receiver flowrate increases somewhat linearly for approximately one half hour until the output of the receiver is matching the solar flux input to the cavity.

After the receiver doors have opened, the heliostat executive control system will command the heliostats to acquire and then fine track. If required, heliostats will be brought up to the fine track mode in groups to avoid exceeding allowable tube temperatures and temperature rates in the boiler and superheater.

The initial steam flow from the receiver will be dumped to the condenser via the thermal storage charge and discharge valves (see schematic) and the admission steam bypass valve. This will heat the steam line to the thermal storage subsystem.

In this mode of operation, steam flow to the condenser is controlled to maintain acceptable temperature and pressure in the receiver. The condensate, booster, and feedwater pumps are operating to return feedwater to the boiler.

2. Thermal Storage Charge

When the steam pressure and temperature builds up to the point of being acceptable to the thermal storage subsystem, approximately 8826 kPa (1280 psig) superheated, steam flow is diverted from the condenser to the thermal storage subsystem. The thermal storage subsystem will be charged to provide at least one hour of generating capability before starting the turbine-generator. This one-hour of stored energy will provide assurance that the generator will not have to be taken off line because of a few passing clouds. During this period of operation, the TSS accepts all of the flow as shown in Figure II-9.

In this mode of operation, the pressure of the steam entering the thermal storage subsystem is controlled by the charge valve. The thermal storage subsystem accepts all the steam that can be provided by the receiver at the set steam conditions.

The thermal storage pumps are operating and the charge/discharge valves are in the charge position. Temperature of the steam going to the thermal storage condenser is controlled by regulating the feedwater flow to the desuperheater. Flow of the thermal storage fluids is controlled in response to the temperature of the steam and water at the outlet of the respective stages.

Feedwater flow from the thermal storage subsystem is controlled by pressure in the condenser, which positions the charge feedwater control valve. The feedwater goes to the suction of the high pressure boiler feedwater pump, for return to the receiver.

Just prior to starting the turbine, the main steam bypass valve is opened to the condenser to heat the lines to the turbine main steam stop valve.

3. Turbine Startup to Synchronization

At this point in time, the thermal storage subsystem has essentially been charged sufficiently to provide one hour's worth of discharge, if required; and the turbine is being brought up to synchronization from turning gear speed. During this mode of operation, the steam flow to the thermal storage subsystem will decrease as the flow to the turbine increases as shown in Figure II-9.

Steam is admitted to the turbine in a partial arc mode. The turbine is gradually brought up to speed according to a planned speed-time profile to avoid exceeding allowable limits on rate of increase on turbine speed. When the turbine-generator reaches rated speed, the generator is synchronized with and tied in to the electrical power network.

During this time, most of the steam being produced by the receiver is flowing to the thermal storage subsystem. Rated steam pressure at the turbine inlet is maintained by the charge valve. The thermal storage subsystem accepts all steam not required for bringing the turbine up to speed. Steam temperature is controlled to the rated value by the receiver attemperator control which controls the flow of desuperheating spray water.

Feedwater flow back to the receiver steam drum is controlled by the receiver three-element feedwater regulator.

4. Turbine Synchronization to Planned Load

After the generator is synchronized and tied in to the power network, the unit load is increased until it matches the flowrate capability provided by the receiver. Load is increased gradually using a planned load time profile to stay within allowable rates of increase of turbine temperatures. As steam flow to the turbine increases, flow to thermal storage decreases. For the typical day shown, Figure II-9, the turbine is capable of accepting the total flow from the receiver. At this point, the thermal storage subsystem is shut down and the turbine is switched from load control to a turbine following mode. The turbine throttle valve is controlled by inlet pressure to the turbine so that the turbine accepts all the steam available from the receiver at rated pressure. Turbine-generator speed control is provided by the speed governor and the tie to the electrical power network.

5. Generating from Receiver Alone

During this time period, operation continues in the turbine following mode, when the turbine accepts all of the steam available from the receiver. As shown in Figure II-9, the electrical power output increases as the receiver flowrate increases which is tracking the available solar insolation. This mode of operation continues until the flowrate available from the receiver reaches the maximum flowrate capability of the turbine.

6. Generating from Receiver and Charging Storage

For this typical day, the steam flow available from the receiver reaches the maximum capability of the turbine at about 10:30 a.m. at which time excess steam is fed to the thermal storage subsystem. Since there is a minimum flow that the thermal storage subsystem can accept, the turbine-generator load is cut back to provide that minimum flow to the thermal storage subsystem.

The turbine-generator is switched to load control and the electrical power generated remains constant at the required load. The thermal storage subsystem charge valve controls inlet pressure to the turbine, and the thermal storage subsystem accepts all the steam available from the receiver over and above the turbine-generator fixed load requirement.

This mode of operation continues until the receiver flowrate drops to the turbine-generator capability.

7. Generating from Receiver Alone

Just before 1 p.m. on the example day, the available steam flow from the receiver decreases to the point of being equal to the turbine capability. At that time, the thermal storage system is shut down and the turbine is switched back to the turbine-following mode. The turbine accepts all of the steam available from the receiver as the electrical power output decreases during the afternoon as the available insolation; and therefore, the available receiver steam flow decreases. As the generator output approaches 7 MWe, preparations are made for operating the thermal storage subsystem in the discharge mode to support the decreasing receiver output and maintain the 7 MWe output.

8. Generating from Receiver and Storage

At about 3:00 p.m. on the typical day shown in Figure II-9, the thermal storage subsystem is placed in the discharge mode to supplement the output capability of the receiver to maintain the electrical power generated at 7 MWe. As can be seen in Figure II-9, as the main steam from the receiver to the turbine decreases, steam from thermal storage to the admission steam throttle valve increases to maintain the desired electrical power output.

Prior to admitting the thermal storage subsystem steam to the turbine, the discharge valve and admission steam bypass valve are opened to heat the steam lines to the turbine admission valves. When the lines to the turbine are heated, the admission steam bypass valve is closed and steam is admitted to the turbine through the turbine admission steam valves. The admission steam throttle valve is on load control; the main steam throttle valve remains in the turbine-following mode. The turbine continues to accept all available steam from the receiver, and the load control gradually opens the admission steam throttle valve automatically to maintain constant generator electrical output.

The thermal storage pumps are operating, and the charge/discharge valves are in the discharge position. The temperature of the steam supplied to the turbine admission valves is controlled by regulating the flow of Hitec in the thermal storage subsystem output stage. Pressure of the steam supplied to the turbine admission valves is controlled by controlling the flow of thermal storage fluid to the evaporator.

Flow of feedwater to the thermal storage subsystem is controlled by the thermal storage subsystem three-element feedwater regulator. Feedwater flow from the electrical power generation subsystem splits with part going to the thermal storage subsystem and part going to the receiver.

9. Generating from Storage Alone

As the typical day approaches sundown, the receiver flow decreases to zero and the thermal storage subsystem flow increases to maintain the desired 7 MWe power output. The receiver is shut down, and the receiver doors are closed. The heliostats are commanded to their stowed position. The pilot plant remains in this mode of operation until the energy store in the thermal storage subsystem approaches the minimum level required for overnight operation.

10. Turbine Planned Load to Trip

When the thermal storage subsystem stored energy approaches the minimum level required for overnight functions, the electrical load will be gradually decreased according to a planned load-time profile to the point where the turbine trips off line. The turbine coasts down to turning gear speed, and the thermal storage subsystem is shut down except for supplying sealing steam to the turbine.

11. Shutdown and Overnight

In this mode of operation, the objective is to maintain the system in readiness for startup the next morning with minimum thermal transients. The heliostats are stowed, and the receiver doors are closed. The turbine is left on turning gear and the thermal storage subsystem supplies sealing steam to the turbine throughout the night.

D. PILOT PLANT INTERFACES

This schematic of the pilot plant, shown in Figure II-10, has circled numbers at the interface points for each subsystem. The circled letters represent the line segments between the interface points, with the exception of C, D, and E associated with thermal storage which are used to describe the flow for various operating modes. These identified locations represent the interface (I/F) points and pipe segments performance data shown in Table II-8. The data shown in Table II-8 represents conditions at the subsystem interfaces for some of the various operating modes of the pilot plant.

Figure II-10 Pilot Plant Interface

I/F Points	Max Interface Conditions		Design Point Receiver Gen		TSS Generation Only		Receiver & TSS Generation		TSS Charge Only		Max Rec/Max Turb & Charge TSS	
	Temp, K	Press, kPa	Temp, K	Press, kPa	Temp, K	Press, kPa	Temp, K	Press, kPa	Temp, K	Press, kPa	Temp, K	Press, kPa
1	789	9,137	789	9,137	--	--	789	9,137	789	9,137	789	9,137
2	784	8,826	784	8,826	--	--	784	8,826	784	8,826	784	8,826
3	783	8,791	783	8,791	--	--	783	8,791	--	--	783	8,791
4	672	4,137	--	--	672	4137	672	4,137	--	--	--	--
5	539	4,827	--	--	477	4827	472	4,827	539	--	495	4,827
6	539	13,445	482	12,756	--	--	472	--	539	--	495	13,445
7	536	11,722	479	11,032	--	--	469	--	536	--	493	11,722
8	536	10,859	479	10,170	--	--	469	--	536	--	493	10,859
Pipe Segments	Flow Rate, kg/h		Flow Rate, kg/h		Flow Rate, kg/h		Flow Rate, kg/h		Flow Rate, kg/h		Flow Rate, kg/h	
A	69,400		51,962		--		19,958		61,235		69,400	
B	58,967		51,962		--		19,958		--		58,967	
C	61,235		--		--		--		61,235		10,433	
D	61,235		--		47,174		22,680		61,235		10,433	
E	57,379		--		47,174		22,680		--		--	
F	61,235		--		47,174		22,680		61,235		10,433	
G	69,400		51,962		--		19,958		61,235		69,400	

International Units

I/F Points	Max Interface Conditions		Design Point Receiver Gen		TSS Generation Only		Receiver & TSS Generation		TSS Charge Only		Max Rec/Max Turb & Charge TSS	
	Temp, °F	Press, psig	Temp, °F	Press, psig	Temp, °F	Press, psig	Temp, °F	Press, psig	Temp, °F	Press, psig	Temp, °F	Press, psig
1	960	1325	960	1325	--	--	960	1325	960	1325	960	1325
2	952	1280	952	1280	--	--	952	1280	952	1280	952	1280
3	950	1275	950	1275	--	--	950	1275	--	--	950	1275
4	750	600	--	--	750	600	750	600	--	--	--	--
5	510	700	--	--	399	700	390	700	510	--	432	700
6	510	1950	408	1850	--	--	390	--	510	--	432	1950
7	505	1700	403	1600	--	--	385	--	505	--	427	1700
8	505	1575	403	1475	--	--	385	--	505	--	427	1575
Pipe Segments	Flow Rate, lb/hr		Flow Rate, lb/hr		Flow Rate, lb/hr		Flow Rate, lb/hr		Flow Rate, lb/hr		Flow Rate, lb/hr	
A	153,000		114,556		--		44,000		135,000		153,000	
B	130,000		114,556		--		44,000		--		130,000	
C	135,000		--		--		--		135,000		23,000	
D	135,000		--		104,000		50,000		135,000		23,000	
E	126,500		--		104,000		50,000		--		--	
F	135,000		--		104,000		50,000		135,000		23,000	
G	153,000		114,556		--		44,000		135,000		153,000	

English Units

Table II-8 Pilot Plant Interface Performance

E. SYSTEM EFFECTIVENESS

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.

1. Pilot Plant System Availability Allocation Analysis

The availability requirement specified for the pilot plant is 90%, or conversely, an outage rate of 10%. The requirement is interpreted in terms of inherent availability which is directly related to design capability versus an operational availability which accounts for downtime factors due to administrative causes and downtime for preventative maintenance. The inherent availability model assumes that the system is capable of producing 100% load at the generator output, excluding any outage due to poor weather conditions. The operational availability is expected to exceed the inherent availability through the utilization of the large amount of free time available daily for either corrective or preventative maintenance. The detailed analysis for the system availability allocation analysis has been documented in a separate report.

Figure II-11 depicts the relationship and the interaction between the availability analysis task and the design evolution tasks. The preliminary analysis represents the initial step of this interaction process which will lead to a system design which maximizes the system uptime in a cost effective manner.

Definitions of the terminology and concepts used in the availability allocation analysis are shown below:

- Pilot Plant Availability Requirement - "The pilot plant shall be designed to be available for the generation of electrical power a minimum of 90% of the planned time for power generation, based on reliability and maintainability considerations and exclusive of insolation conditions."

PP = Pilot Plant System

*Components for which reliability data are not available.

Figure II-11 Design Evolution/Availability Analysis Interaction

- Inherent availability models are used rather than "operational" or "use" availability modes on the basis that inherent availability refers primarily to the built-in capability of the system to operate satisfactorily under stated conditions. Inherent availability is mathematically defined below:

$$A_i = \frac{MTBF}{MTBF + MTTR}$$

where MTBF = Mean Time Between Failure

MTTR = Mean Time to Repair (or Mean Corrective Active Downtime).

The inherent availability figure of merit assumes perfect preventative maintenance and excludes any consideration of waiting time (logistics or administrative delay) associated with repair of failure or forced outages. Operational availability takes into account total downtime (corrective plus preventative). Inherent availability is associated with full capacity operation: 10 MW during collector operation; and 7 MW based on operating from discharge of the thermal storage subsystem.

- Allocation is defined as a process of apportioning or subdividing of a top goal to obtain lower level goals. The purpose of this allocation is to obtain lower level availability allocations and the related hardware characteristics: MTBF and MTTR, which can be more readily assessed and used in tradeoffs to optimize availability.

A summary of availability estimates and allocations by subsystem is shown below:

<u>Subsystem</u>	<u>Estimate</u>	<u>Allocation</u>
Electrical Power Generation	0.963	0.963
Receiver	0.993	0.990
Thermal Storage	0.960	0.959
Collector	0.999	0.999
Master Control	0.985	0.985
Total System	0.904	0.90

The basis for the above estimates is the initial conceptual design. It is not the intent to reallocate on the basis of further design development until the final design phase. During the detail design phase, the subsequent analysis task is oriented toward the assessment of the subsystem designs to determine the actions necessary to achieve the subsystem allocation. These actions include one or more of the following:

- Selection of detail design with known reliability statistics.
- Incorporating redundancy when the reliability estimates are too low or repair time is excessive.
- Trading off reliability for shorter repair time concepts to arrive at equivalent or improved availability.

The overall objective is to obtain an availability in excess of the pilot plant requirement of 0.90 within anticipated cost limitations. On the basis of these activities, the allocation will be revised and reflected in the final plant specifications. In addition, the final analysis will identify potential hardware changes which could further improve availability but have not been incorporated due to high program cost impact.

In addition to the collection of MTBF and MTTR data for the detail design, life estimate data will also be collected to aid in establishing planned preventative maintenance schedule.

Some of the major considerations identified in our preliminary analysis, from a repair or restoration viewpoint, include the following:

- Design concepts which allow for isolation of failed elements without significant impact on the subsystem or the pilot plant. Particular attention is required for failures which could require a cold start.
- Development of calibration procedures to improve turnaround in the heliostat field.
- Development of techniques which could be used to reduce waiting times such as cooling procedures to remove excessive heat.

a. Pilot Plant System Analysis - The model concepts and assumptions used in the availability analysis are developed below followed by a discussion of the system level data analysis and allocation.

1) Availability Model and Assumptions - As indicated earlier, "Inherent availability" is the figure of merit associated directly with the design characteristics of the system. In developing a model for preliminary analysis, the tradeoffs in model complexity (and cost) versus reasonableness of data for model inputs was considered. The following assumptions are employed:

- Inherent availability is associated with full capacity operation: 10 MW during collector operation and 7 MW during thermal storage discharge operation of the electrical power generation.
- The preventative maintenance period of 25 days per year is excluded in the model.
- The mean time to repair (MTTR) should take into account a time increment of 6 hours for failure which could shutdown the system and require a cold start.

The mathematical model is stated as:

$$A_{\text{SYSTEM}} = \prod_{N=1}^{N=4} A_N$$

N = 1 Receiver Subsystem

N = 2 Thermal Storage Subsystem - Charge or Discharge

N = 3 Collector Subsystem

N = 4 Electrical Power Subsystem

$$A_N = \prod A_i$$

where $A_i = \frac{\text{MTBF}}{\text{MTBF} + \text{MTTR}}$ for functional elements in each subsystem,

where MTBF is the Mean Time Between Failure,

MTTR is Mean Time to Repair (or Mean Corrective Action Downtime).

The above model does result in conservative estimates of system availability due to the numerous conditions where a subsystem downtime will not degrade system availability. This argument is based on the amount of free time available on a daily basis in which subsystem failures could be fixed without causing system downtime.

- 2) Data Analysis and Allocation - Each subsystem was analyzed to obtain an estimate of the inherent availability for that subsystem. The results of this analysis by subsystem are shown in Table II-9. To obtain an allocation by subsystem, an availability risk:

$$U_N = \frac{(1 - A_{\text{SUBSYSTEM}})}{(1 - A_{\text{SYSTEM}})} (1 - .90), \text{ where } A \text{ is the initial estimate.}$$

The resultant unavailability risks provide guidelines which can be adjusted up or down based on engineering judgment to arrive at a final allocated subsystem unavailability. The final allocations for each subsystem are shown in Table II-9.

Table II-9 Summary of Availability Estimates & Allocations

SUBSYSTEM	ESTIMATE	ALLOCATION
Electrical Power Generation	0.963	0.963
Receiver	0.993	0.990
Thermal Storage	0.960	0.959
Collector	0.999	0.999
Master Control	0.985	0.985
PILOT PLANT	0.904	0.90

Additional data elements which are considered necessary for final assessment of preventative maintenance and equipment spares are life estimates and duty cycles of individual hardware items.

To provide an insight into the sensitivity of an operational availability analysis to the use of free time for corrective maintenance, a gross system analysis was performed. This analysis is based on the following assumptions:

- Waiting and administrative downtime associated with corrective active downtime is equal to 50% of the mean corrective active downtime.
- Annual preventative maintenance downtime for each subsystem is as follows:

Collector - 480 hours, EPGS - 600 hours, Thermal Storage - 600 hours, Receiver - 120 hours, Master Control - 168 hours.

- Preventative maintenance is accomplished during free time.

The results of the gross analysis shown in Figure II-12 are not completely indicative of operational availability since not all maintenance can be practically performed during free time. However, the chart does indicate a trend for significant improvements in operational availability when this free time is factored in.

b. Electrical Power Generation - Subsystem Analysis - The historical availability of small fossil-fueled power plants has been very good, but this high availability is the result of designs that evolved slowly and were thoroughly cleaned-up over several iterations. The design of the electrical power generation subsystem portion of the 10 MWe pilot plant cannot be expected to do quite as well for several reasons:

- Steam conditions are higher than those usually encountered in power plants of a similar size with a record of high availability, so some innovation may be required.
- Diurnal cycling imposed on the turbine is expected to result in a reduction of the turbine life.

Figure II-12 Operational Availability Analysis

- In view of the dual steam sources, the control requirements of the system are more complex than has been customary.
- Probable location in a desert adds some complexity to the water supply and heat rejection systems.

Nevertheless, it appears likely that a reliability can be achieved for the electrical power generation subsystem that is substantially better than that of recent large turbine plant installations. This is due largely to the stability and reliability of design and performance achieved over many years for most components in a plant of this size, and also to the generally less complex nature of a small plant compared with the larger, more sophisticated designs.

The availability of the electrical power generation subsystem portion of the pilot plant will be almost identical, whether the receiver or the thermal storage system is used as the heat source. Support data has therefore been prepared only for the case where the receiver supplies the steam.

Areas where significant unknowns exist include the following:

- The coordinated control system may be complex and be demanding on the control valves it will employ.
- The turbine-generator may be subjected to thermal transients as a result of clouds crossing the sun and causing energy source transfer to the thermal storage system.

Therefore, its availability is estimated to be slightly poorer than that of similar-sized turbine-generators now in industrial use.

Redundancy is provided so that full load can be achieved despite casualties to some components. Where such redundancy is provided, provisions will be made to repair or replace the damaged component without shutting down the unit.

It should be emphasized that the failure rates assumed are not well documented. Sources include various published articles, an unpublished survey by an oil company, information from manufacturers and estimates based on in-house reports available to Bechtel equipment specialists. Further investigation during the course of the design will likely disclose better values. However, the order of magnitude is probably correct in most cases.

In some cases, MTBF and MTTR information was not available and availability information only is listed.

c. Receiver Subsystem Availability Analysis - The results of the preliminary availability allocation study for the receiver is divided into five areas: steam drum, evaporator, superheater, cavity and supporting structure, and auxiliary equipment.

In the calculation of the availability of the subsystem, only those failures which may lead to an unscheduled shutdown of the subsystem are considered. The frequency of such occurrences for each component is represented as MTBF. The numerical values are generated on the basis of the failure record data of fossil fuel power plant components, published in references 1 and 2 by Edison Electric Institute. Average outage time for each such unscheduled shutdown is estimated and reported as MTTR. Included in MTTR are shutdown, fault detection, parts awaiting, repair and startup times. The MTTR values were estimated on the basis of the information available for fossil fuel power plant components 1, 2, and discussions with experienced Foster Wheeler engineers.

No specific information on the life estimates was available. However, past experience on Foster Wheeler designed and fabricated boilers has shown that the life times for most components are much greater than 30 years.

In conclusion, some statements regarding the scheduled unavailability are germane here. Based on Foster Wheeler experience on fossil power plants, one week of scheduled shutdown annually is usual. This is needed for annual inspection and regular maintenance for the whole plant.

REFERENCES

- 1) Report on Equipment Availability for the Ten-year Period, 1964-1973, EEI Publication N.74-57, Issued December 1974, Edison Electric Institute, N.Y.
- 2) Equipment Availability Fossil Component Cause Code Summary Report 1973, to be Used in Conjunction with EEI Publication No. 74-57.

d. Thermal Storage Subsystem Availability Analysis - The analysis of the thermal storage subsystem is based on maintaining an availability state for both the "charge" and "discharge" cycles. The majority of subsystem hardware is common to both modes of operation.

Subsystem hardware elements were identified by Georgia Institute of Technology. Hardware mean time between failure estimates were obtained as follows:

- Selecting failure rates from RADC-TR-74-268, an RADC Non-electronic Reliability Notebook on EEI Publication No. 74-57, Equipment Availability Component Cause Code Summary Report (Fossil Fuel Power Plants).
- Applying engineering judgment to failure rates in the reference noted above to account for increased stresses anticipated with the Hitec and Therminol storage mediums.

Mean time to repair estimates are based on current judgment of the subsystem complexity and waiting times required to reduce temperatures to reasonable handling loads.

e. Collector Subsystem Availability Analysis - The collector subsystem has been analyzed in terms of maintaining a sufficient number of heliostats in an "up" condition to obtain the 10 MWe capability. For the collector field, the number of heliostats required for full capability is approximately 1708. The field is sized at 1718 which results in an estimated availability of 0.999. This calculation is based on a normal approximation to the binomial distribution.

f. Master Control Subsystem Availability Analysis - The master control subsystem major units are the minicomputer and the display and control rack. Peripheral equipment for the computer including tape drives, typewriters, etc., are excluded on the basis that their failure is not considered to be an outage which would result in a system outage.

2. Pilot Plant Failure Modes and Effects Analysis

As part of the system effectiveness task, the subsystem conceptual designs have been analyzed to identify equipment failure modes and their consequences of failure. For each failure mode, the fail safe provisions and failure warning provisions have been identified to the extent that the design has been firmed up.

This detailed analysis is documented in a separate report. Figure II-13 depicts the relationship and interaction between the failure modes and the design evolution tasks.

The basic format established for conducting the subsystem analysis includes the following data elements:

- Unit Nomenclature - Title of Hardware Element
- Probable Failure Mode - Identification of likely failure modes for each hardware element.
- Consequence of Failure - Identification of the effects of the failure mode on the subsystem operation.
- Fail Safe Provisions - Identification of subsystem design features which are incorporated to protect the subsystem from catastrophic damage if the failure mode were to occur (redundancy is an acceptable but not a necessary solution).
- Failure Warning Provision - Identification of displays or alarms which indicate the failure has occurred or is impending.

The ground rule for determining effect of failure is based on considering the failure in all possible modes of plant operation. For system level assessments, each hardware element is further categorized by a criticality code to obtain an overall system status and to determine priorities for subsequent action. The criticality codes assigned to hardware elements are based on the existence of one or more hardware element failure modes which could result in the following pilot plant effects:

	<u>Criticality</u>
Loss of Power Generation - Fail Safe Concept Under Investigation	1A
- Fail Safe Concept Identified: System Shutdown Procedure Required	1B
Degradation of Power Generation Capability - Fail Safe Concept Under Investigation	IIA
- Fail Safe Concept Identified	IIB

Figure II-13 Pilot Plant FMEA Design Evolution Interaction

		<u>Criticality</u>
No Effect on Power Generation	- Fail Safe Concept Under Investigation	IIIA
	- Fail Safe Concept Identified	IIIB
	- Fail Safe Concept Not Required	IIIC

Based on the analysis of the current conceptual design definition, summaries of selected criticality categories are provided as follows:

- Criticality IA Items - Hardware elements in which a failure results in loss of power generation capability and the fail safe provision is not identified include the cavity door actuating mechanism.
- Criticality IIA Items - Hardware elements in which a failure results in degradation of power generation capability and the fail safe provision is not identified include the steam drum internals, service water valves, some control air system components, and the Hitec throttle valve.

Although the design is not firm, these specific areas will be given special priority to assure appropriate design action is taken to develop fail safe provisions. Likewise, as the subsystem design detail is expanded, additional items will be added to these lists to assure project visibility.

It is also apparent from the analysis that the system shutdown or alternate procedures are required. This effort will be an ongoing effort in conjunction with the detailed design of the master control subsystem.

The analyses of the turbine-generator and steam system discloses that they are sensitive to carryover of boiler water and of water treatment chemicals from the receiver. It is extremely important that the receiver and thermal storage subsystem be so designated as not to permit moisture carryover or carryover of such solids as disodium phosphate. If such carryover does occur, the turbine plant availability will be adversely affected to a significant degree.

The main conclusions drawn by Foster Wheeler Energy Corporation from the receiver subsystem failure modes and effects analysis are as follows:

- The most critical failure mode in the receiver subsystem is the possible leakage of the superheater and evaporator tubes. The predominant cause of this failure is overheating. To prevent or minimize this failure, control devices could be installed interlocking the tube temperature profile to the heliostat position. This will be further investigated.
- Tube leakage may also occur due to cyclic creep and fatigue. This problem will be further studied by analytical methods as well as by test.

Based on the subsystem FMEA and an evaluation of operating stresses, a list of system concerns, Table II-10, was prepared. This list of concerns includes conditions which are likely to accelerate the occurrence of hardware failure modes or result in multiple hardware failures. For each concern, tentative solutions have been identified. The list will be expanded and retained as a checklist to assure that positive action is implemented in the final pilot plant design.

F. ENVIRONMENTAL CRITERIA

The specific location of the proposed pilot plant has not been identified; however, the region of likely candidates lies in the Southwestern United States, as shown in Figure II-14. This region meets the criteria of an annual mean daily total insolation of 5000 watt-hr/m²/day. The pilot plant design must be compatible with the general environments and site conditions within this region.

In addition to defining the environments, we have identified operational constraints or procedures required when selected environments exceed these values. When rated performance is a direct function of an environment, we have identified the specific value or range which is currently considered as a design point.

The environmental design requirements have been defined and documented in an environmental criteria for a plant design which will perform or survive (for a 30 year life) when exposed to the environments of the final plant site location. The operational limits or constraints indicate procedures to be implemented to

Table II-10 Pilot Plant System Concerns

Conditions Which are Likely to Accelerate the Occurrence of Hardware Failure Modes or Result in Multiple Hardware Failures in the Pilot Plant:	
<u>CONDITION</u>	<u>SOLUTIONS</u>
<ul style="list-style-type: none"> ● Thermal Cycling Resulting in Significant Changes in Plant Hardware ● Contamination of Feedwater or Steam ● Loss of Auxiliary Power Required to Operate Pilot Plant ● Accumulation of Sand and Dust in Pumps, Compressors, Bearings, Switches, etc. 	<ul style="list-style-type: none"> ● Use of Sealing Steam to Minimize Thermal Stress Changes ● Increased Insulation ● Feedwater & Steam Treatment ● Incorporate a Standby Diesel Generator Unit and an Emergency Battery Inverter Pack ● Use of Seals in Hardware Exposed to Direct Environment ● Use of Filters to Condition Air for Protection of Hardware Located Within Buildings ● Preventative Maintenance Inspections

Figure II-14 Pilot Plant Environmental Considerations

protect plant equipment during exposure to environmental extremes. The operating ranges for rated performance indicate current design point conditions for 10 MWe capacity. Plant operation above these ranges will generally result in reduced capacity. A summary of these environmental requirements, as reported in the environmental criteria, are shown in Table II-11. In addition to the environmental requirement shown in the Table, the plant will be designed for survival for seismic conditions of 0.30 g in both horizontal and vertical directions. The minimum allowable soil bearing capacity is assumed to be 1814 Kg/m² (2 ton/ft²). Site peculiar requirements will be revised upon selection of a specific site.

G. SYSTEM TEST AND EVALUATION

The pilot plant test program which follows has been updated based on the Preliminary Design Baseline Review (PDBR). The subsystem experiment test plans are included in their respective subsystems.

1. Engineering Tests

Engineering tests are conducted to obtain information to aid in the design and manufacturing processes. Engineering tests will be performed on components, assemblies, or subsystems where: 1) new design is required; 2) the results of the subsystem research experiments indicate that substantial improvements in efficiency, cost, design, or operation can be achieved; or 3) the design features were not demonstrated during Phase 1.

The most significant engineering test identified at this time is the heliostat structural integrity test. The approach to verifying structural integrity requires two test sequences: the first to corroborate analyses developed during the design phase by comparison of heliostat structural deflections resulting from an idealized test loading and analytical predictions for the same idealized loadings; and the second to test the yoke assembly with horizontal and azimuth drives installed to demonstrate that these mechanisms will function under load. These tests are planned to be conducted at the Denver Division's structures laboratory.

II-11 Summary of Environmental Requirements and Constraints

Environment	Worst Case Design Criteria	Operational Limits/ Constraints	Operating Range for Rated Performance
Ambient Temperature	253 K (-13°F) to 323 K (+122°F)	None	303 K (+85°F) at 2:00 p.m. on a Clear Day at Winter Solstice
Relative Humidity	<u>Surface (Unprotected) Limits</u> <ul style="list-style-type: none"> o 98 - 100% for 24 hours Vapor Concent. 0.9 gr ft⁻³ (2.1 gm⁻³) Air Temp. 261K (+11°F) o 15% for 6 hours each 24 hours (302K) 34% for Remaining 18 hours (289K) Vapor Concent. 2.0 gr ft⁻³ (4.5 gm⁻³) <u>Interior (No Conditioning)</u> <ul style="list-style-type: none"> o 2% for 1 hour Vapor Concent. 44 gr ft⁻³ 10.1 gm⁻³ o Linear Changeover 4 hours o 22% for 15 hours Air Temp. 311K (100°F) 	None	Same as Design Criteria
Ambient Pressure	940 mb (27.7 in ^{Hg}) to 1050 mb (31.0 in ^{Hg})	None	Same as Design Criteria

Table II-11 Summary of Environmental Requirements and Constraints (Cont.)

Environment	Worst Case Design Criteria	Operational Limits/ Constraints	Operating Range for Rated Performance
Wind	Peak Wind Speed of 45.6 m/sec @ 9.1 m (120 mph @ 30 ft)	Collector Heliostats Stowed and Recvr Doors Closed at Peak Wind Speed of 17.9 m/s (40 mph) @ 9.1 m (30 ft) (Heliostat Design for Structural Integrity up to 22.4 m/s @ 9.1 m)	Average Wind Speed of 0 to 3.4 m/s @ 9.1 m (0 to 7.5 mph @ 30 ft) for Receiver Supplied Steam Mode
Rain	30 min. of Rain $12.7 \pm$ 2.54 cm hr^{-1} ($5 \pm 1 \text{ in.}$ hr^{-1} ;	Collector Heliostats May be Stowed for Extended Periods of Rainfall	Thermal Storage Power Generation Only
Hail	Maximum Size - 2 in. (50 mm) Velocity of Fall 30.5 m/s^{-1} (100 ft sec^{-1}) Wind Speed 10 m/s^{-1} (33 ft sec^{-1}) Density 0.8 g cm^{-3} (50 lb ft^{-3}) Period of Large Hail 15 min.	Collector Heliostats to be Stowed	Same as Above
Snow	75 kg m^{-2} (15.4 lb ft^{-2})	Collector Heliostats to be Stowed	Same as Above
Lightning	Peak Value: 200 K amps Rise Rate: 100 K amps (sec^{-1}) Decay Rate: 2 K amps for 98 sec to 7 K amps Total Duration: 0.3 m/s Average Charge: 5 coulombs	Collector Heliostats to be Stowed	Same as Above

Table II-11 Summary of Environmental Requirements and Constraints (Concl.)

Environment	Worst Case Design Criteria	Operational Limits/ Constraints	Operating Range for Rated Performance
Sand and Dust	3m (10 ft) Above the Surface Wind Speed = 20 ms^{-1} (45 mph) Sand Flux Density = $.098$ $\text{g m}^{-2} \text{ s}^{-1}$ ($.14 \text{ lb/in}^{-2}$)	Stow Heliostats Based on	
Solar Insolation (Normal Incidence)	Maximum 1.0226 kW m^{-2} Summer & Winter	None	$\geq 0.8255 \text{ kW m}^{-2}$ at 2:00 p.m. on Clear Day at Winter Solstice
Ultra-Violet Radiation	33,000 Sun Hours Cumulative Dose	None	

The first test will be conducted with the heliostat base affixed to a stiff base fixture. Test loads for an idealized load case (perhaps loading only one quadrant of the assembly) will be applied to the back side of the heliostat with dead weight and a cable and pulley arrangement. Deflections and movement of the base of the heliostat will be measured to provide base fixture movement data that will be used to modify measured deflections at the mirror surface. All deflections will be measured with electrical deflection transducers with resolutions no greater than 0.0025 cm (0.001 in.) or dial gages. The deflection transducers will be mounted on a structure that is structurally independent of the base fixture or test loading apparatus.

The second test will be conducted with the heliostat yoke assembly, drive systems installed, mounted on the same base fixture as in the first test. A dummy mirror module assembly will be installed on the horizontal axis. The interface of the mirror module with the yoke will duplicate that of the deliverable hardware. Test loads simulating critical design conditions will be applied and the drive mechanisms activated to demonstrate operation. A few degrees of movement is anticipated to be adequate proof of operation, specifically to prove that the slew motor gear sets will operate under worst case winds to bring the heliostat to a safe position.

The components identified as criticality IA or IIA by the Failure Modes and Effects Analysis (FMEA) will be thoroughly investigated during this test phase. The specific objective of these investigations is to eliminate the component from being a criticality I or II item. The components which fall into these categories based on the preliminary FMEA are the cavity door actuation mechanism, control air compressor, deaerator, computer, steam drum internals, service water valves, control air receiver, control air plumbing, and the Hitec throttle valve.

2. Qualification Tests - Qualification tests are the formal demonstration that the design, implementation, and manufacturing methods have resulted in a system confirming to the specification requirements. Qualification tests will be performed on components, assemblies, subsystems, and system. The component- and assembly-level tests are primarily environmental exposure and life tests. The subsystem level tests are primarily repeats of the subsystem research experiment tests, with emphasis on new or modified design. Subsystem qualification tests may extend into the integrated system test phase to verify requirements that cannot be reasonably checked at a lower assembly level. A preliminary component qualification test matrix is shown on Table II-12.

TABLE II-12 COMPONENT QUALIFICATION TEST MATRIX

COMPONENTS	ENVIRONMENTS/TESTS					
	FUNCTIONAL	TEMPERATURE CYCLE	HUMIDITY	RAIN	VIBRATION	EMI
SENSOR COLLIMINATING TUBE	X	X	X	X		
SENSOR ELECTRONICS	X	X	X	X	X	X

Components will be acceptance tested prior to being subjected to qualification test.

Life tests will be performed on life limited components where historical data is insufficient to determine the life time of the component or the mean-time-between-failure. Where possible, life tests will be accelerated.

3. Acceptance Tests (Components)

Acceptance tests are performed to verify that the production hardware is free from defects and meets the specification requirements. A preliminary component acceptance test matrix is shown on Table II-13. The list of components was taken from the availability analysis and will be updated as the pilot plant design progresses.

All testing will be performed in accordance with approved operating instructions and the results included in the inspection records. The inspections and functional tests, as specified, will each be peculiar to the type of hardware being tested. Inspection includes x-ray where required by code. Coupons will be used to sample characteristics of the mirrors rather than testing each mirror. Burn-in tests will nominally be of 100-hour duration. Burn-in of the sensor electronics will include 5 temperature cycles from 248 K (-13°F) to 333 K (140°F). Where lot sampling is specified, an AQL level will be selected from MIL-STD-105D or the appropriate ASME code will be used. All components will be procured and acceptance tested to their respective codes, where applicable.

4. Installation Verification

Installation tests are to confirm that the system has been installed in accordance with the applicable drawings, specifications, standards, and codes. Installation tests will include power distribution, cable routing, wire continuity, insulation resistance, connector keying and cleanliness, pressure, leakage, alignment, bench mark location, line flushing and cleaning, and flow tests, where applicable.

5. Subsystem Functional Verification

After installation verification is complete, functional verifications will be made to verify the subsystem logic and operation prior to the generation of steam. The support subsystems comprising the balance of the plant, i.e., fire protection, switchyard, domestic water, drainage, etc., will be activated as required to support the following activities. Functional

TABLE II-13 COMPONENT ACCEPTANCE VERIFICATION MATRIX

ITEM	INSPECTION	FUNCTIONAL TEST	LEAKAGE	PRESSURE	LOT SAMPLE	BURN-IN
ELECTRICAL POWER						
GENERATION SUBSYSTEM						
TURBINE BYPASS VALVE	X			X		
CONTROL, THROTTLE, AND ADMISSION VALVES	X			X		
PRESSURE SENSORS	X	X				
FLOW SENSORS	X	X				
TEMPERATURE SENSORS	X	X				
TURBINE-GENERATOR	X					
STEP-UP TRANSFORMER	X	X				
GENERATOR CIRCUIT BREAKER	X	X				
AIR COOLED HEAT EXCHANGER	X					
SERVICE WATER PUMPS	X	X	X			
SERVICE WATER VALVES	X			X		
AIR COOLED MAIN CONDENSER	X					
CONDENSER FANS	X	X				
WELL WATER PUMPS	X	X	X	X		
WATER SUPPLY FILTERS	X					
WATER SUPPLY VALVES	X			X		
VACUUM PUMP	X	X	X			
MOISTURE SEPARATOR	X					
CONTROL AIR COMPRESSOR	X	X				
CONTROL AIR RECEIVER	X					
CONTROL AIR DRYERS	X	X				
CONTROL AIR VALVES	X			X		
CONTROL AIR FILTER	X					
DEAERATOR	X	X	X	X		
CONDENSATE AND FEEDWATER PUMPS	X	X	X	X		
PIPE	X					
EXHAUST DUCT DRAIN PUMP	X	X	X	X		
FEEDWATER HEATERS	X	X		X		
EXTRACTION STEAM VALVES	X			X		
FEEDWATER HEATER LEVEL CONTROLS	X	X				
MAKE UP DEMINERALIZERS	X					
CONDENSATE STORAGE TANK	X					
CONDENSATE DEMINERALIZERS	X					
RECEIVER SUBSYSTEM						
STEAM DRUM	X					
PIPE AND TUBING	X					
EVAPORATOR DRAIN VALVE	X			X		
ATTEMPERATOR	X			X		
SUPERHEATER DRAIN VALVES	X			X		
CAVITY DOOR ACTUATOR	X	X				
PRESSURE SENSORS	X	X				
TEMPERATURE SENSORS	X	X				
FLOW SENSORS	X	X				
SAFETY VALVES	X	X	X	X		
FEEDWATER REGULATOR VALVES	X			X		
STOP VALVES	X			X		
FEEDWATER CHECK VALVES	X			X		
LEVEL SENSORS	X	X				
FEEDWATER REGULATOR	X	X				
COLLECTOR SUBSYSTEM						
SENSOR ELECTRONICS	X	X				X
SOLAR CELL SUN SENSOR	X	X				
TRACKING MOTOR	X	X			X	
DRIVE MECHANISM	X	X			X	
LIMIT SWITCH	X	X				
ENCODER	X	X			X	
MIRROR	X				X	
MIRROR SUPPORT	X					
MASTER CONTROL SUBSYSTEM						
COMPUTER	X	X				X
DISPLAY/CONTROL RACK	X					
CABLING	X					
THERMAL STORAGE SUBSYSTEM						
HEAT EXCHANGERS	X		X	X		
PIPE	X					
VALVES	X			X		
PUMPS	X	X	X	X		
STORAGE TANKS	X					
LEVEL SENSOR	X	X				
TEMPERATURE SENSORS	X	X				
PRESSURE SENSORS	X	X				
FLOW SENSORS	X	X				
DESUPERHEATER	X		X	X		
CONDENSER	X		X	X		

verification will include the following tasks.

a. Heliostat Operation - Heliostat operation will be checked using the manual control capability. The checkout sequence will include all manual control operations. The response to the commanded operations will be compared against the specification response times.

b. Mirror Pointing - Mirror pointing will be accomplished by manually positioning the heliostat in the expected attitude for the day and time of day specified. The mirror alignment tool will be positioned in front of the center mirror in the heliostat so the incident beam will simulate the angle of incidence of solar radiation at the selected time. The mirror will then be adjusted manually so the reflected beam is pointed at the center of the alignment target near the receiver aperture location. The reflected image on the alignment target will be viewed directly by the technician during this adjustment. When the optimum mirror alignment is achieved, the laser will be translated to the next mirror of the heliostat. The procedure will be repeated until the entire heliostat is focused at the aperture. The light translation mechanism will be designed to maintain the same angle of incidence for all mirrors on the heliostat.

c. Software - The software will be completely checked out in Denver prior to use at the pilot plant site. Modular programming techniques will be used to facilitate software development. As modules are completed, testing will demonstrate specification compliance. Testing will consist of verification and validation as shown on Table II-14.

The verification phase will show that: the code complies with the software requirements document (SRD); the code is error free; and the code is complete. Manual verification will be performed before software system integration by extensive desk checking, visual code review, and correlation with the SRD.

Each module will be validated using open-loop checks of function code paths. Once the functional operation of each module has been shown, the modules will be integrated and system validation will be performed to show: 1) functional operation of all of the modules working together as an integrated software system; and 2) that functional performance of the integrated software system complies with the SRD.

TABLE II-14 SOFTWARE VERIFICATION AND VALIDATION

	VERIFICATION	VALIDATION
OBJECTIVES	PROVE THE CODING MATCHES THE DESIGN SPECIFICATION PROVIDE A FORMAL CHECK ON THE SOFTWARE BUILD PROCESS PROVE THE ELIMINATION OF CODING ERRORS	PROVE THE FUNCTIONAL REQUIREMENTS ARE MET PROVIDE A FORMAL CHECK FOR THE DESIGN PROCESSES USING END ITEM SOFTWARE PROVE THE PERFORMANCE REQUIREMENTS
TOOLS AND TECHNIQUES	PATH ANALYSIS CODING TO SPECIFICATION CORRELATION DIMENSIONAL ANALYSIS ACCURACY ANALYSIS TIMING ANALYSIS USAGE ANALYSIS CODING RULES COMPLIANCE	HELIOSTAT RESPONSE SIMULATORS PROTOTYPE COMPUTERS AND ELECTRONICS PERFORMANCE ANALYZERS
DISCIPLINES	SOFTWARE SYSTEM ENGINEERING	SYSTEM ANALYSIS

All normal operating modes, monitors, and emergency control modes will be verified. Selected malfunctions will be simulated during this test phase to verify the MCS logic and control functions. These malfunctions will be selected from the most critical failures identified in the failure modes and effects analysis.

d. Component Calibration and Functional Checks - Each individual part and component, such as instruments, valves, meters, etc., of the POCE system will be functionally checked. Instruments will be calibrated and set points for alarms, trips, etc., verified. Verification of control loops will be accomplished by applying test signals to note: 1) readings of all indicators and recorders in the loop; 2) positions of end devices such as control valves, dampers, etc.; 3) output signals of transducers and converters; and 4) trip and clear values of switches, annunciators, etc. Wherever possible, process variables will be used for the input test signal. Adjustments will be made and retests accomplished to ascertain proper calibration and functional operation of all parts and components.

e. Compatibility Tests - Compatibility tests will be performed to verify the pilot plant subsystem interfaces using the interface simulators. These tests will verify: 1) the command, control, and monitoring interfaces between the master control station and each subsystem; 2) the override and emergency control modes and responses; and 3) the mechanical and electrical interfaces between subsystems.

f. Subsystem Functional Tests - The pilot plant subsystems will be divided into functional groups. Each group will be tested to verify its performance as completely as possible without the use of steam generated by the receiver. The main purpose of this testing is to uncover and correct as many equipment deficiencies as possible before conduct of the acceptance test phases that follow. In particular, the tests include: 1) verification of cleanliness criteria; 2) obtaining starting and running current and voltage readings on large motors; 3) verification of pump head/capacity (HQ) curves; 4) obtaining vibration readings on rotating equipment; 5) measuring pressure differentials in piping and across heat exchangers; 6) verification of design objectives; 7) verification of correctness of design and construction; 8) proving of correct system actions with loss of services such as loss of control power and actuation air; and 9) determination of "off-design" performance.

A considerable number of temporary installations, such as instrumentation and "jumper piping", etc., will be required to derive maximum benefit from these tests.

6. Subsystem Marriage Tests

After the subsystem functional tests are complete, the functional groupings of subsystems will be incrementally married together. The test flow, shown in Figure II-15, basically follows the flow of energy through the system.

FIGURE II-15 SUBSYSTEM MARRIAGE TEST FLOW

a. MCS to Collector - The initial tests consist of exercising the positioning controls and responses for each heliostat. The following items will be tested: 1) the capability to drive the heliostat ± 1.92 radians (± 110 degrees) in azimuth at a rate of 0.70 radians per hour (40 deg/hr); 2) the capability to drive the heliostat 3.14 radians (180 degrees) in elevation at a rate of 0.17 radians per hour (10 deg/hr); 3) the capability to drive in azimuth and elevation at the fast slew rate of 0.63 radians per minute (36 deg/min); and 4) proper response to all other MCS generated commands, including the ability to control selected portions of the field.

After the MCS commands (executive control mode) have been verified, the automatic tracking capability will be tested using a selected number of heliostats. These tests will include: 1) the capability of the fine track sensor to acquire and command the heliostat to the proper position, starting with an offset error of approximately 0.21 radians (12 deg); 2) the capability to fine track to within $\pm 2.91 \times 10^{-4}$ radians (± 0.017 deg), in a 0 to 2.24 m/s (5 mph) wind; and 3) the capability to track with a commanded offset.

b. MCS/Collector to Receiver - The marriage of the MCS/collector to the receiver will be the first time steam is generated in the system. Prior to steam generation, the MCS/receiver interface will be verified. All controls and responses will be verified, including emergency actions.

The initial steam generation will be for boilout and cleaning with the steam being vented. This test will be followed by a steam blow of the main steam pipes. Initial vacuum will be attained in the condenser of the steam cycle. The equipment associated with condensate, feedwater, and heat rejection will be started up.

The collector subsystem will be used to incrementally increase the heat flux to the receiver cavity up to full availability with the steam flow rate, pressure, and temperature being verified at selected flux levels. A series of tests will be conducted using steam in varying amounts to simulate transient conditions and evaluate the response capabilities of the receiver and collector subsystems.

The following items will be verified by test: 1) the capability to control heat flux by moving heliostats; 2) receiver startup from a cold condition in 6 hours; 3) receiver startup from a hot condition in 20 minutes; 4) the capability to accept feedwater at flow rates up to 69,400 Kg/h (153,000 lb/hr) at 479 K (403.5°F) and 10,860 kPa (1575 psig); 5) the capability to generate steam at flow rates up to 69,400 Kg/h (153,000 lb/hr) at 789 K (960°F) and 9136 kPa (1325 psig); and 6) the capability to limit the saturation temperature drop to a maximum of 339 K (150°F) in 12 hours at an ambient temperature of 248 K (-13°F) and a wind velocity of 100 m/s (225 mph). The environmental conditions for overnight shutdown will probably not be reached during our test period requiring the test data to be extrapolated to these limits.

c. MCS/Collector/Receiver to Thermal Storage Subsystem - The thermal storage subsystem (TSS) will be verified before steam is applied to the turbine steam at 8826 kPa (1280 psig) and 784 K (952°F) will be supplied starting at the minimum rate of 2,268 Kg/h (5000 lb/hr). The flow rate will be incrementally increased until the maximum rate of 61,235 Kg/h (135,000 lb/hr) is reached. A series of TSS charge/discharge tests will be performed under varying steam conditions to evaluate TSS performance and transient operations. The following items will be verified by test: 1) the capability to accept steam at flow rates from 2,268 kg/h (5000 lb/hr) to 61,235 Kg/h (135,000 lb/hr) at 8826 kPa (1280 psig) and 784 K (952°F); 2) the capability to supply steam at 47,174 kg/h (104,000 lb/hr) at 4137 kPa (600 psia) and 672 K (750°F) for 6 hours; 3) the capability to supply saturated steam at 453.6 kg/h (1000 lb/hr) at 1034 kPa (150 psia) for 24 hours; 4) the time required to go from a charge to discharge

mode and a discharge to charge mode; and 5) the capability to fully charge the thermal storage system in 6 hours.

d. MCS/Collector/Receiver/TSS to Electrical Power Generation Subsystem (EPGS) On-line - The initial roll of the turbine generator will be accomplished with the receiver supplying steam in the bypass mode to the condenser. After performance checkout of the turbine generator, initial connection to a load simulator will be made. A procedure similar to that of a conventional power plant is envisioned for this initial roll. A series of tests will be performed on the turbine generator at different load plateaus, increasing in increments of about 1 MWe. Heat balance calculations will be made, and the system tuned for operation in this mode. System response to load transients and other load-following capabilities will be tested by using simulated load profiles and by making changes to the set point of the turbine generator controller. System responses to emergency shutdown at various load conditions will also be evaluated. The turbine will be tested in accordance with ASME Performance Test Code 6-1964.

The following operational modes will be tested:

- 1) Cold Start - The initial tests are anticipated to take longer than the 6 hour specification requirement. At each step of the way, the instrumentation will be carefully evaluated to verify the system is operating nominally and adverse stresses are not being introduced.
- 2) Hot Start - The initial hot start tests will also be carefully monitored and will probably take in excess of the 20 minute specification.
- 3) Operation from the receiver alone.
- 4) Operation from the receiver while charging storage.
- 5) Operation from the receiver and storage.
- 6) Operation from storage alone.
- 7) Transient operations, including mode switching.
- 8) Shutdown.

7. Acceptance Tests (Pilot Plant)

Pilot plant acceptance tests are to confirm that the hardware is designed and constructed in accordance with the applicable subsystem and system specifications. It is anticipated that the majority of the subsystem acceptance test requirements will be verified during the subsystem marriage tests. The primary system specification requirements to be verified by test include the following:

- a. The capability to deliver 10 MW net electrical power operating from the receiver at 2 p.m. on winter solstice with a minimum insolation of 0.825 kW/m^2 . Due to uncertainty in plant construction schedules, this requirement may be fulfilled by extrapolating the input/output data from the test days to winter solstice.
- b. The capability to deliver 7 MW net electrical power for a minimum of 6 hours while operating from a fully charged thermal storage system.
- c. The capability to deliver 7 MW net electrical power operating from both the receiver and thermal storage subsystems. This requirement will be verified at a minimum of three receiver/thermal storage power ratios.
- d. The capability to deliver 10 MW net in 6 hours after startup from a cold condition and in 20 minutes after startup from the hot standby configuration.
- e. The capability to demonstrate shutdown within specification limits under normal and emergency conditions. Shutdown under normal conditions will be verified by test. Shutdowns under emergency will be verified by analysis except in the cases where emergency shutdown is required to save the pilot plant. Real or simulated malfunctions will not be inserted into the system for the sole purpose of verifying emergency shutdown procedures.

After pilot plant acceptance, we anticipate there will be a continuing period of pilot plant operations, culminating with power being supplied to a utility. The purpose of these operations is to obtain the necessary data to determine the economic feasibility of a Central Receiver Solar Power Plant. The type of data will include, but not be limited to:

- a. Daily net electrical power produced;

- b. Scheduled and unscheduled maintenance;
- c. System availability;
- d. All costs to operate plant;
- e. Verification of system efficiency at various times of the year.

III. COLLECTOR SUBSYSTEM DESIGN AND ANALYSIS

A. INTRODUCTION

Since contract go-ahead, two phases in the development of the Collector Subsystem design have been started. The first phase, which consisted of the conceptual design of the heliostat field for both the 10 MWe pilot plant and for the subsystem research experiment (SRE), was completed in November 1975.

The second phase, which continues through February 1976, contains the detail design of the heliostats for the SRE. The results of this phase will include a complete set of drawings, specifications, and manufacturing plans to fabricate, assemble and install four heliostats at the Martin Marietta Solar Power Test Facility.

Section B describes both the analyses and tests conducted during the conceptual design phase and the preliminary baseline design of the collector subsystem for the 10 MWe pilot plant. Section C provides a brief summary of the subsystem research experiment (SRE).

B. PRELIMINARY BASELINE DESIGN OF 10 MWe COLLECTOR SUBSYSTEM

1. Performance Requirements

The basic collector subsystem performance requirements are described in the Collector Subsystem Design Criteria. This design criteria is based upon the Collector Subsystem Requirement Specification submitted with the proposal and is updated as required to provide an up-to-date design description for the purpose of defining performance and interfaces and to provide a common design point for trade studies and analyses.

a. Power Requirement - The collector subsystem is designed to provide 39.6 MWth to the receiver cavity aperture at 2 p.m. local sun time on a clear day at winter solstice under an insolation of 0.8255 kW/m^2 (0.0767 kW/ft^2).

b. Reliability Requirements - Additionally, two other factors are considered in the sizing of the heliostat field; availability and usability. Availability is the percentage of time that the collector subsystem can be expected to be operational without failures that would affect the whole field, e.g., a power failure. The availability is shown to be 0.999 which corresponds to 999 hours of operation out of every 1000 usable hours of sunlight.

Usability is defined as the percentage of heliostats in the field that are operational. A usability of 0.994 implies that ten heliostats, out of a total field of 1712, will be non-operational (in a maintenance condition) at any time. The executive control system, which directs all heliostat operations except for the fine tracking mode, has a separate usability factor of 0.985, which corresponds to 15 hours of down-time out of every 1000 hours.

c. Rotational Requirements - During the year, each heliostat is required to move at a wide range of speeds and gimbal angles. Figure III-1 shows the maximum required speeds and angular motion for both the azimuth and elevation axes for a flat field configuration located at 35° N. latitude. Since heliostats must be fully interchangeable for any location in the field, each heliostat is capable of meeting the rate and angular requirements of Figure III-1. A more detailed discussion of gimbal rates and limits of travel as a function of heliostat location in the field are contained in Section III-B8.

2. Error Allocation Summary

The summary presented in Figure III-2 results from a study performed to identify all static and dynamic elements in the collector subsystem that contribute to errors or losses in efficiency and to assign values to each contributor.

The total effect of all losses in efficiency is a reduction in the power delivered to the receiver of 23.1% for wind velocities up to 2.2 m/s (5 mph) and a reduction of 25% for wind velocities from 2.2 m/s to 13.4 m/s (30 mph). These are derived from the total collection efficiencies for each wind velocity at the bottom of the table (Figure III-2(a)).

The allowable tracking errors of 10.2×10^{-4} rad (3.5 arc-min) and 16.0×10^{-4} rad (5.5 arc-min) for the 0 to 2.2 m/s to 13.4 m/s velocity ranges, respectively, were devised from analyses of individual component capabilities and test data. The efficiencies corresponding to these errors were, in turn, derived from Figure III-2(b) which describes the effect of tracking errors on power output.

3. Baseline Collector Subsystem Configuration

a. Field Configuration - The collector field (Figure III-3) was sized and the heliostats arranged to meet the criteria of:

*Based on plant located 35 deg N latitude.

Figure III-1 Maximum Gimbal Rates and Limit of Travel for Baseline Heliostats

Error or Loss Source	Tracking Error		Efficiency Factor
	Radians	arc-min	
Heliostat Useability		-	0.994
Tower Shading		-	0.974
Cosine Effect		-	0.939
Reflectivity		-	0.85
Pointing and Tracking			
Pointing		0.5	
Sensor Alignment		0.5	
Hysteresis	$2.908 \cdot 10^{-4}$	1.0	
Circuit Calibration	$1.454 \cdot 10^{-4}$	0.5	
Drive Mechanism Backlash		-	
Sensor Wind Loading	$1.454 \cdot 10^{-4}$	0.5	
Structural Deformation:			
Wind Velocity = 13.4 m/s 30 mph	$7.272 \cdot 10^{-4}$	2.5	
Wind Velocity = 2.4 m/s 5 mph	$1.454 \cdot 10^{-4}$	0.5	
Foundation * Pedestal Deflection			
Total Tracking:			
2.4 to 13.4 m/sec (5-30 mph)	$1.599 \cdot 10^{-3}$	5.5	0.982
0 to 2.4 m/sec (0-5 mph)	$1.018 \cdot 10^{-3}$	3.5	0.988
Optical Inaccuracies			
Aberration		-	
Tangential and Focusing Error	$1.076 \cdot 10^{-3}$	3.7	0.988
Total Collection Efficiency			
5-30 mph			0.750
0 to 2.4 m/sec (0-5 mph)			0.763

Combined Tracking Error/Surface Error Effects on Power Output Performance

Figure III-2 Preliminary Error Allocation Summary for Collector Subsystem

Figure III-3 Collector Field Configuration

1) providing 39.8 MWth to the receiver cavity at 2:00 p.m. local sun time on winter solstice under an insolation of 0.8255 kW/m^2 ; 2) eliminating shading or blocking for a minimum sun angle of $\pi/6$ rad (30°) above the horizon; and 3) utilizing as high a packing factor as possible.

The field can conveniently be divided into three segments, each with its own rationale for arrangement. The first segment, which includes rows 1 through 13, was arranged so that no shading, i.e., shadowing of one heliostat from the sun by one in front, occurs for sun angles of $\pi/6$ (30°) or greater. The heliostats are spaced 9.8 m (32 ft) apart in parallel rows. The interrow spacing is 12.2 m (40 ft).

The second segment is also aligned in straight rows (14 through 25). Instead of the interrow spacing being constant, the spacing gradually increases from the front row (13.4 m or 44 ft) to the last row of the segment (23.5 m or 77 ft) to minimize blocking of the reflected rays by heliostats in front. Spacing between heliostats in a row remains constant at 9.8 m (32 ft) for this segment.

The last and furthest section of the receiver is arranged on radial lines so that the nearest blocking heliostat is two rows in front of any heliostat in order to obtain high packing densities. Although the heliostats still are aligned in rows, both the interrow and interheliostat spacing increases towards the back and sides of the field, respectively.

The total area of the collector field is approximately 0.311 km^2 (0.12 square miles), exclusive of boundary roads, fences, support buildings, etc. The number of heliostats is 1718, giving a total reflecting area of $63,843 \text{ m}^2$ ($687,200 \text{ ft}^2$).

The shortest focal distance in the field is 148 m (485 ft), while the longest focal distance is 693 m (2273 ft).

b. Collector Subsystem Seasonal Performance - Figure III-4 summarizes collector subsystem performance as a function of time of day for the summer and winter solstices and for the spring equinox. These data, derived from the 1963 Inyokern data, include the effects of varying cosines, the shading factor (loss of incoming insolation due to shading of heliostat by ones in preceding row) and the blocking factor (blocking of reflected light by heliostat in preceding row).

Figure III-4 Effective Collector Subsystem Performance for Summer and Winter Solstice and Fall Equinox

Table III-1

Estimated Collector Field Power Requirements for 10 MWe Pilot Plant

DUTY CYCLES

Winter solstice - average duty cycle for field	Azimuth: 9%
	- Elevation: 4.5%
Summer solstice - average duty cycle for field	Azimuth: 15%
	- Elevation: 4.5%
Average duty cycle for year	- Azimuth: 12%
	- Elevation: 4.5%

POWER

Normal operation (fine-track model)

Average power required over year	10.2 kW
Peak power demand during year	61.9 kW
Slew mode (fast slew at π rad (180°)/5 min)	
Steady state	247.2 kW
Inrush (staggered startup)	247.7 kW

The varying start and end times for each performance curve are based on the sunrise and sunset at that time of year. The noted fluctuations show the effects of shading and blocking on performance. These fluctuations are most pronounced for summer solstice.

The effects of shading and blocking produce, during winter solstice, a higher efficiency in the 3:00 to 4:30 p.m. period for the equinox curve. For a fall equinox curve, a similar period is observed between 7 to 9 a.m.

c. Estimates of the Collector Field Power Requirements - A summary of the estimated collector field power requirements is provided in Table III-1. These estimates largely result from the power requirements for the tracking and slew motors, which are described in Table III-2.

The highest power consumption occurs during slewing operations, when the entire collector subsystem is driven in elevation to a stow (mirrors down) position. Each heliostat requires approximately 15 minutes to reach a stowed position from the extreme elevation operating position. To minimize the effect of the high inrush currents to the elevation slew motor, a preprogrammed, staggered startup procedure, initiated by the executive control computer, would be used. A 100 millisecond period is required for the inrush current to decay to the steady state current. The startup period for the entire field of 1718 heliostats would be 172 seconds or approximately 2.9 minutes. Therefore, the total time for the collector subsystem to reach a stowed position against wind velocities up to 22.4 m/s (50 mph) is approximately 18 minutes.

Stowing the mirrors on each heliostat in a face down position would be adequate to provide protection against extreme weather, or in an emergency situation.

The non-uniform positions of the heliostats resulting from an emergency shutdown would, however, make access difficult for many of the proposed maintenance vehicles. For routine maintenance operations, therefore, it is necessary to slew the heliostats about the azimuth axis until uniform positioning is achieved. The maximum time to reach a uniform position, based on the maximum azimuth gimbal angles (Figure III-1), is approximately 12 minutes, including staggered startup.

Table III-2 Power Requirements for the Tracking and Slew Motors

Function of Motor	Drive Efficiency (%)	Horsepower Rating (HP)	Current at 120 VAC	
			Inrush	Steady State
Slew against m/s (50 mph) wind	39%	1/12	4.0	1.2
Tracking against m/s (30 mph) wind	37%	.01	NA	0.3
*Pulse duration 1 sec. Average duty cycle = 25% Maximum duty cycle = 50%				

Table III-3 Command Matrix for Normal Operating Modes

Operator Commands	Resulting Computer Commands to Heliostats
Clear	Clear registers. Motors off.
Acquire	Fast slew to computed coarse synthetic track position. Then continue synthetic tracking.
Fine Track	Fine track command to enable in-line sensor control loop. Error feed command (straight feed, crossfeed or combination) as determined by computer calculations.
Synthetic Track	Gimbal angle commands to the slew motor. Commands are calculated by the computer to offset the beam above the receiver.
Stow	Fast slew to stow position
Direct Slew	Turns on fast slew motor until commanded off by another command from the operator.

Power to operate the collector field under normal operations will be obtained from the pilot plant. Emergency operations require a backup diesel generator.

d. Executive Control Modes - The collector subsystem executive control modes can be grouped into three categories: normal control modes commanded through the executive control system and initiated either by the operator or by an automatic start-up or shut-down sequence; fail safe and emergency control initiated automatically by the executive control system in response to a sensed emergency; and manual control by an operator using a portable plug-in controller at the heliostat. These three categories are described in the following paragraphs.

- 1) Normal Control Modes - The normal control modes are commanded by the operator via the executive control system. The operator commands are initiated by switches on the control panel and are fed to individual heliostats, a group of heliostats, or the whole field as selected by the operator. These commands to the heliostats are shown in Table III-3.

The executive control system also provides the capability for automatic control of the start-up and shut-down sequence. For automatic start-up, the computer sends "acquire" and "fine track" commands to groups of heliostats in sequence to provide the proper build-up of heat input to the receiver.

- 2) Fail Safe and Emergency Modes - The executive control system automatically sends synthetic track commands, causing the heliostat beam to be offset above the receiver, or stow commands in case of an emergency such as overtemperature of the superheater tubes. The operator also has the capability to send a synthetic track command or a stow command to a group of heliostats or the whole field.

The direct slew command provides the operator with the capability to directly command the azimuth and/or elevation slew motors on, in case of loss of control due to failure of certain components such as the gimbal angle transducer or A/D converter.

Limit switches turn off either the azimuth or elevation drive motors if the gimbal angles reach predetermined safe limits of travel.

- 3) Local Manual Control - Each heliostat incorporates an external junction box which accepts a portable plug-in control box. By operating switches on the control box, the operator can turn on the slew motors directly and drive the gimbals to any desired position for performing maintenance operations.

e. Field Maintenance - Field maintenance of the collector subsystem includes such operations as the periodic calibration of electronic assemblies, lubrication of the drive mechanisms, washing of the mirrors, etc.

In the pilot plant, all heliostats would normally be stowed in a mirrors down position when non-operable. This practice, as demonstrated during several years of heliostat testing at MMC, greatly increases the time intervals between mirror cleaning operations.

An example of the relatively short exposure time necessary to accumulate enough contaminants on the mirror surface to affect performance is shown in Figure III-5. The prototype heliostat, which is stowed in a mirrors-up position, suffered significant degradation in only six days of exposure. The circular patterns, which are prints of the steel mirror warping structure, are caused by differential heating of the mirrors.

Water requirements to clean the heliostat field were evaluated to establish the maintenance schedule. A commercial glass washing company that has a 10-minute wash and rinse cycle using 29 gallons of water would be necessary to clean each heliostat. The recommended cleaning agent was a mild solution of trisodium phosphate.

Assuming that a mobile washing vehicle capable of cleaning two heliostats simultaneously were used (Figure III-6), the time required to clean a field of 1718 heliostats would be 143 hours or about 18 days. The total water requirement for the field would be 188,576 liters (49,822 gallons).

The mirror cleaning vehicle, as presently envisioned, will consist of a tank truck with two 3217-liter (850 gal.) tanks, pumps, plumbing, and high-pressure nozzles (Figure III-6). The truck will be equipped with a lift device capable of carrying two men to a height of 7.62 m (25 ft).

Figure III-5
Prototype Heliostat Showing Effects of Six Days of Exposure

Figure III-6 *Mirror Cleaning Equipment*

Each tank will be approximately 1.22 m (4 ft) in diameter and 3.04 m (10 ft) long. One tank will contain a mixture of soft water and trisodium phosphate for initial cleaning and the other will contain soft water for rinsing.

f. Relative Cost Breakdown for Individual Heliostats - The pie chart of Figure III-7 indicates that the structure of the heliostat and the fine tracking sensor tower account for approximately 61% of the cost of an individual heliostat, or over five times as much as the reflecting surfaces (12%) and the drive mechanism.

4. Preliminary Baseline Heliostat Design

The baseline heliostat (Figure III-8) for the 10 MWe pilot plant consists of nine mirrors with a total reflecting area of 37.2 m^2 (400 ft^2). The weight of the entire heliostat, including the reflective surface, is approximately 2041 kg (4500 lbs).

Each heliostat is capable of being driven in both elevation and azimuth to the rotational limits described previously by a pair of direct gear drive mechanisms. The error signals are supplied to these drive units through a closed loop control system operating from a tower mounted sensor.

a. Reflecting Surface - The reflecting surface consists of nine $2.03 \times 2.03 \text{ m}$ ($6.7 \times 6.7 \text{ ft}$) fixed curvature, second surface silvered, 3 mm thick white glass mirrors with a spectral reflectivity of 91%. The reflecting area of each mirror is 4.1 m^2 (44.9 ft^2). Each mirror on a heliostat is individually pointed.

The focal length of the mirrors is fixed for each heliostat in the field and is dependent upon the slant range between the center mirror of the heliostat and the receiver aperture in a number of increments which vary from 148 m (485 ft) to 693 m (2275 ft). All mirrors on a heliostat have the same focal length.

A typical fixed curvature mirror is shown in Figure III-9. A $2.03 \times 2.03 \text{ m}$ second surface mirror is bonded to a sandwich consisting of a layer of adhesive, a fiberglass sheet, a 5.1 to 10 cm (2 to 4 in.) thick sheet of honeycomb, a second layer of adhesive, and a final face sheet of fiberglass for back surface protection. The edges of the mirror are sealed by wrapping with fiberglass. Three mounting supports are inserted into the honeycomb layer to allow attachment to the mirror support assembly.

Figure III-7 Relative Cost Breakdown for Individual Heliostat--10 MWe Pilot Plant

Figure III-8 Baseline Heliostat for Pilot Plant

Figure III-9 Fixed-Curvature Mirrors

Table III-4 Load Criteria for Heliostat Drive Mechanism

Maximum Slew Torque at 22.35 m/s (50 mph) Wind Velocity	= 6,598 N-m (58,400 in.-lb)
Maximum Tracking Torque at 13.5 m/s (30 mph) Wind Velocity	= 2,373 N-m (21,000 in.-lb)
Base Bending Moment at 22.35 m/s (50 mph) Wind Velocity	= 53,698 N-m (39.600 ft-lb)
Base Shear at 22.35 m/s (50 mph) Wind Velocity	= 14,280 N (3,210 lb)

Further discussion of the characteristics of fixed curvature mirrors is contained in Section III-B5.

b. Drive Mechanism - Separate positioning systems for the azimuth and elevation axes of rotation are provided by direct gear mechanisms designed to minimize the tracking error associated with gear backlash and bearing deformation. Wind and gravity induced torques are prevented from back-driving the mechanism by introducing a worm gear reducer in the final stage of gear reduction. The principal factor governing drive mechanism design is the nonuniform distribution of wind-induced forces across the heliostat surface areas.

Since appropriate values of the aerodynamically induced torques could not be satisfactorily ascertained by analysis due to the absence of test data pertaining to the particular "slotted" configuration of the mirror assembly, wind tunnel tests (Section III-B6) yielded the tabulated load criteria for the drive mechanism (Table III-4).

The drive mechanisms for both azimuth and elevation positioning are basically identical with respect to motor and gearing requirements. Some minor differences exist in the ring gear configuration due to mounting constraints.

Figures III-10 and III-11 show the direct gear drive concept developed to meet the requirements of this proposal. Section III-B7 summarizes the tradeoff studies that led to this concept.

The design was mostly influenced by the azimuth drive since, in addition to operating against the aerodynamic torques previously defined, it must resist a large base bending moment with negligible deflection and anchor to the foundation. A further criterion was that both elevation and azimuth drive mechanisms use the same components to the greatest extent possible to facilitate maintenance and spare parts planning.

Another requirement is drive irreversibility, demanding a worm or similar reducer in the gear train as close to the azimuth axis as possible. In the azimuth axis, gear backlash is important since it induces dynamic errors whenever veering winds cause torque reversals. This does not occur in the elevation axis because aerodynamic and unbalance torques eliminate backlash errors. Backlash will not exceed 2 arc-minutes.

Figure III-10 Azimuth Drive Unit

Figure III-11 Elevation Drive Unit

Figure III-10 shows a nodular (ductile) iron casting to American Gear Manufacturers Association (AGMA) specification N15 that is used to provide not only the gear but also the foundation mounting flange and bearing mounting in one component. The casting is simple; the webbed and recessed areas may be formed by a single ring core.

This gear casting, heat treated to 255 BHN, is readily machinable and mates with a steel pinion of 300 BHN hardness, thus providing an optimum hardness difference between the tooth surfaces. Structural deflections that would significantly affect the tracking accuracy of the heliostat are held to an acceptable level.

The compactness of this design is largely due to the few tooth stress cycles throughout the required life, allowing a maximum tooth bending stress of approximately 65% of the yield stress of the gear material rather than the fatigue stress level that is the normal situation. This stress level when used in the AGMA strength formula, leads to a 5 DP tooth size.

The bearings are tapered single row, straight bore, roller bearings of 170 mm ID and 254 mm OD. This selection is based on a computer analysis conducted by the Timken Company optimizing bearing capability and cost for a given deflection. These bearings operate at 50% of their ultimate capacity at wind velocities of 22.35 m/s (50 mph) on a heliostat normal to that wind. This is the maximum load condition they will experience since at greater wind velocities, the heliostat will be in the stowed condition. At the maximum tracking wind velocity of 13.5 m/s (30 mph) they operate at 20% of ultimate capacity.

The bearings are preloaded in the azimuth drive by torquing the retaining plate screws to predetermined levels. Bolts are lock-wired to preserve this setting. Preloading is essential for gear mesh accuracy and elimination of bearing play.

The same tapered roller bearings are used in the elevation drive, not because the loads warrant such large bearings, but because they permit a through-shaft sufficiently large to facilitate attachment to the mirror module. These are high-production bearings and the most economical in their size and capacity range.

The final gear mesh is totally enclosed using a sheet metal or cast housing surrounding gear and pinion, and incorporating a peripheral seal, as shown in Figures III-10 and III-11, which seals against a machined diameter provided on the main casting. Precisely cut lengths of elastomeric seal material are inserted into the seal cavity and retained by means of helical garter springs rather than a continuous circular seal that would be difficult to replace.

Grease nipples are provided for direct access to both bearings and gear mesh. The initial lubricant charge will suffice for many years of operation. In view of the very low speed of operation, an EP grease having good oxidation stability and rust prevention qualities is used.

The dual electric motor drive system is shown in Figure III-12. The Slo-Syn tracking motor will drive through the slew motor and will be back-driven to 1700 rpm by the slew motor during slew operation.

The lower worm reducer efficiency under tracking conditions is due to the very low speed of operation where friction remains in the "breakaway" regime. This situation, however, has been optimized by using a 30:1 ratio worm reducer, which is the minimum ratio offering irreversibility, and hence, optimum efficiency.

The dual-motor approach is simple and electrically more efficient than a single variable speed motor and its associated control. Neither motor uses centrifugal starting switches or brushes, and both are totally enclosed, off-the-shelf items. The slew motor is a permanent split capacitor motor. The tracking motor is a Slo-Syn by Superior Electric. The Slo-Syn has numerous advantages--no inrush or stall currents, no rotor heating, and mechanical simplicity. All these features lead to high reliability and long life. Figure III-13 shows the drive mechanism and controls breadboard used to mock-up the drive assembly.

The means of mounting either an absolute encoder or a potentiometer to determine position is being resolved in discussions with several manufacturers.

c. Structure - The heliostat consists of the azimuth drive module, yoke module, and the mirror module. The mirror module includes the elevation drive unit as an integral part of the assembly. The modular approach for the baseline design greatly simplifies field installation, assembly, and maintenance and

EFFICIENCIES, HORSEPOWER AND RUNNING CURRENTS

	TRACKING	SLEW
FINAL GEAR DRIVE	96%	95%
WORM REDUCER	46%	61%
SLEW MOTOR GEAR HEAD AND SEAL	60%	60%
OVERALL EFFICIENCY	26.5%	34.5%
REQUIRED MOTOR HP	0.005	0.09
SELECTED MOTOR HP	0.01	0.083
MOTOR RUNNING CURRENT, A	0.3	1.2

Figure III-12 Drive Mechanism Characteristics

Figure III-13 Drive Mechanism and Controls Breadboard

permits interchangeability of entire assemblies as well as major components or subassemblies. Use of standard, commercially available structural steel sections, tubes, and accessories throughout minimizes procurement and fabrication time.

- 1) Strength Criteria - Conservative strength criteria have been established for use in design of the collector structure to preclude the need for in-depth structural integrity testing. The criteria are in the form of safety factors applied to limit loads and the interpretation of wind loading. The safety factors are 2.0 for allowable stress (yield) and 4.0 for allowable stress (ultimate). Wind gusts were treated as steady-state loads.

Because of the precise pointing accuracy required of the collector assembly, the heliostat supporting structure has been designed primarily for stiffness rather than stress considerations and the above safety factors have little effect on the design. The only areas they may affect are local bolted joints and connections. Even in these cases, however, deflection criteria tend to dominate.

The symmetrical arrangement of the baseline configuration has proved to be very beneficial in minimizing the effects of elastic deformation. Most of the structural deformation contributing to pointing error is in the yoke structure, accounting for the relatively large and heavy sections. In standard practice, where deflection requirements are critical and govern the structural design, deep built-up sections or truss-type structures are generally indicated. This results in the highest strength-to-weight ratio for a given application. However, due to increased fabrication and assembly time, this approach is more expensive and thus not appropriate for low unit cost mass-produced items.

- 2) Structural Distortion Analysis - Mechanical or structural distortion results from two environmental conditions, i.e., thermal deformation and deformation resulting from wind loading. Thermal deformation is relatively small in comparison to that caused by wind loading and has not been fully evaluated for the baseline design. A transient thermal analysis of the finalized design will be conducted to determine structural temperatures and temperature differentials. Temperature differentials will be translated to thermal deformations to determine overall distortion.

Structural deformations due to wind loading are a function of wind direction and velocity. Wind direction is important in determining whether the resulting deformation has any effect on mirror aiming accuracy. For example, a side wind will cause a sideward deflection of the vertical arms and vertical yoke members, but will not significantly affect mirror aim.

The worst case condition is a wind normal to the plane of the mirrors. The baseline was designed for a nonuniform wind velocity of 13.4 m/s (30 mph) at 9.14 meters above the ground normal to the plane of the mirrors.

The distortion or deflection analysis may be divided into two parts--deflection up to the reference point of the elevation axis encoder, and deflection within the mirror rack about the elevation axis. The first part amounts to a pointing error and includes deflection in the foundation, azimuth drive mechanisms, yoke structure, and elevation axis structure. Also included are errors introduced by gear backlash in the drive mechanisms. Gear backlash in the elevation drive mechanism is eliminated by an intentional structural unbalance in addition to wind torques as the mirrors rotate from a vertical to horizontal plane; thus the backlash will only be realized when the collector is being moved to the stowed position. The second part of the distortion analysis considers a defocusing because of out-of-plane deflection of the mirror support arms and elevation axis shaft.

Gear backlash in the azimuth drive mechanism adds vectorially to the primary deflection (rotation) of the elevation encoder. Although the effect is small, azimuth gear backlash must be included in the analysis as an additional error.

Preliminary distortion analysis illustrates the type and magnitude of mirror rotations that may be realized in a practical collector configuration. Contributions to the pointing error on the center mirror may be broken into the various contribution components as shown in Table III-5.

In addition to the above pointing error, the wind loading caused a racking, and therefore, defocusing of the entire mirror support assembly. Each mirror rotates to some degree about the azimuth (θ_y) and elevation (θ_x) axes. The magnitudes of rotation realized in this preliminary analysis are

Table III-5
Components of Center Mirror
Pointing Error

Component	Error Contribution, mrad (arc-min)
Collector Foundation	0.17 (0.60)
Azimuth Drive Mechanism	1.27 (4.48)
Yoke Structure	1.92 (6.79)

Figure III-14 Out-of-Plane Deflections of 9-Mirror Assembly

given in Figure III-14. These rotations are easily reduced by stiffening the mirror assembly.

- 3) Azimuth Drive Module - The azimuth drive module incorporates the azimuth drive mechanism, azimuth bearing system, and a mounting flange for securing the entire heliostat assembly to the foundation. This module is the first component of the heliostat assembly to be installed in the field. The unit is lowered over a ring of 2.54 cm (1.00 in.) diameter threaded anchor studs imbedded in the reinforced concrete foundation. These studs protrude through a leveling plate that provides a stable mounting base. The yoke module is then lowered by crane and attached to the azimuth drive module. The last item to be installed is the mirror module.
- 4) Yoke Module Assembly - The yoke module comprises the major structural element in the heliostat assembly and transfers wind-induced loads directly to the azimuth bearings. The vertical members of the yoke module are fabricated from wide flange sections (W12 x 31) welded to a horizontal member fabricated from 12 in. square commercial tubing (TS12 x 12 x .25). At the center of the square tube section, corresponding to the azimuth center of rotation, a steel tube section (TS7.75 OD x 3.5) is welded in place. The lower end of the tube bore is chamfered to provide for centering registration of the yoke module on the stub shaft of the azimuth drive module. A 1.27 cm (1/2 in.) thick steel plate welded to the top of the tube has holes to accommodate threaded studs and cone nuts. After the yoke module has been lowered into position, centered, and properly seated on the azimuth module, the cone nuts are installed and torqued to provide a rigid, self-centering connection to the azimuth drive unit.
- 5) Materials - The proper selection of materials for heliostat design is important, considering the severe environmental requirements and duration of exposure. Components sensitive to sand, dust, humidity, and corrosion are protected by environmental enclosures or sealed cavities.

In addition to a moderate thermal environment, components will be exposed to ultraviolet radiation. At the geographical location of the proposed test site, the annual number of usable sun hours varies from 2500 to 3100 hours per year. Therefore, during the 30-year lifetime of the pilot plant, materials will experience UV exposure of between 75,000 and 93,000

hours of radiation. Plastics, rubbers, and some elastomeric compounds deteriorate under long exposure of UV radiation. The use of these materials has been limited.

- 6) Mirror Module - The mirror module for the pilot plant heliostat consists of the 9 individual mirror assemblies, the horizontal tube assembly, the elevation drive mechanism, and the parallel pairs of vertical support tubes for each of the three rows of mirrors.

The selected heliostat design places the horizontal tube assembly approximately 3.71 m (12.5 ft) above the ground. This approach, with the horizontal pivot arm located at the center of the total area of reflective surfaces, tends to balance the torsional effects of wind load distributions. Another advantage of this approach is that the mirror surfaces are elevated above the zone where blowing sand and dirt are most dense.

The elevation drive tube assembly is supported near each end by the vertical members of the yoke module assembly. To minimize the bending moments and midspan deflections due to weight unbalance, the horizontal tube is supported inboard from each end of the tube so the vertical members of the yoke module occur in the first interior mirror space. The elevation drive module is integrated directly into the elevation drive tube.

The ring gear portion of the elevation drive mechanism assembly is attached to the web of the vertical yoke strut by three 19-mm (3/4 in. dia.) high strength bolts that are installed during field assembly. The horizontal tube is fabricated from a standard steel tube section (TS5.56 OD x 0.258). To facilitate assembly and accommodate the elevation drive module, the tube is fabricated in two sections and attached by flange connections to the double-ended output shaft of the elevation drive module. The other end of the tube is cradled by a cutout in the web of the vertical support member where two pairs of cam rollers provide the bearing interface. A separate plate attached to the web slightly above the horizontal tube retains the tube without introducing any additional friction to the rotating tube. This comes into play whenever aerodynamic lift forces exceed mirror module weight for high wind velocities when a heliostat is in the stowed position.

d. Control System - The collector subsystem consists of two basic control elements. The first is a fine-tracking system (Figure III-15), which uses the reflected beam from the heliostat to keep the sun's image centered on the receiver. The second element is an executive control system which is capable of overriding the fine-tracking loop and commanding the heliostat to modes other than the normal sun-tracking mode, i.e., synthetic tracking, acquisition, slewings, stowing, and maintenance modes.

The block diagram (Figure III-16) shows the heliostat control system including the fine tracking control loop and the electronic circuitry required at each heliostat for executive control.

The gimbals are driven by constant speed a-c motors. The "fast" slew motor is used for initial acquisition and for the stow operation. The tracking motor is used for all fine track modes and for coarse synthetic tracking.

- 1) Executive Control - The heliostat executive control system uses a digital computer and data bus to control the operating modes of the heliostat field. Commands are sent in sequence to each heliostat in the field and responses are returned from each of the addressed heliostats. The command to the heliostats is made up of characters containing 8 bits of data plus start and stop bits and a parity bit. Each command is made up of 5 characters--two for the heliostat address, one for the command code, one for the azimuth position command, and one for elevation position command. Each heliostat response is made up of 3 characters--one for azimuth position, one for elevation position, and one for heliostat status. Since each heliostat response immediately follows the command to that heliostat, there is no need to include the heliostat address in the response.

The actual heliostat gimbal angles contained in each heliostat response are compared to the gimbal angles computed by the coarse synthetic tracking algorithm. This provides a positive safety check on the operation of each heliostat. If the actual gimbal angles differ from the computed gimbal angles by more than a predetermined tolerance (less than $\pm .1$ radian), the computer actuates an alarm for the operator.

A single twisted, shielded pair is used for both the commands to the heliostats and the heliostat response.

Figure III-15
Simplified Block Diagram of Heliostat
Fine-Tracking Function

Figure III-16 Block Diagram of Heliostat Control System

Repeaters are used as required to provide acceptable loading on the data bus and maintain a high signal-to-noise ratio. The use of a single twisted shielded pair minimizes the labor cost of wiring the collector field.

A bus data rate of 10 K baud provides the capability to send a command to and receive a response from each heliostat in the field once every 20 seconds or less. A bus data rate of 20 K baud reduces the time to 10 seconds. These data rates and time intervals are adequate for communicating with the whole field. As a part of the collector experiment, the data bus will be operated at different data rates to check the effect on error rate.

Synthetic tracking commands are transmitted as 8-bit command for azimuth and 8 bits for elevation. This provides 256 steps; and since the total travel is less than 4.47 radians (256 degrees) in both axes, the step size is less than .0175 radians (one degree). The resulting error contribution is less than $\pm .009$ radians ($\pm \frac{1}{2}$ degree). The synthetic tracking commands are transmitted at three-minute intervals. The resulting contribution to azimuth error is equal to or less than .0175 radians (one degree), for a maximum azimuth rate requirement of .7 radians/hour (40 deg/hour).

The contribution to azimuth error will normally be much less than that, and the contribution to elevation error will be less than .0044 radians ($\frac{1}{2}$ degree).

- 2) Fine Tracking Control - Fine tracking is obtained through the use of an individual, closed loop system for each heliostat comprising a tower-mounted sun sensor driving self-contained error correction circuitry (Figure III-17). This circuitry, in turn, applies corrective signals to the drive motors as required to keep the reflected image of the center mirror of the heliostat on the receiver aperture within the required accuracy. The sensor, described in Figure III-18 in greater detail, utilizes two solar cell sensing pairs.

Concepts for using solar cells as the sensing elements in pointing systems for terrestrial solar power systems are not new. The feasibility of this type of sensor has already been demonstrated in both the United States and Russia. The remaining task is to adapt the principle to produce low cost, easily fabricated units capable of providing each heliostat in a field with the required pointing accuracy.

Figure III-17
Relationship between Heliostat
and Fine-Tracking Sensor Tower

Figure III-18 Details of Fine-Tracking Sensor

A series of breadboard fine pointing sensors and their accompanying control units (Figure III-19) have been developed at Martin Marietta to determine whether solar cell sensing elements could provide the required accuracy and to explore various means of achieving the low cost criteria. The results of initial testing indicate that acceptable accuracies can be obtained.

During normal tracking operation, each heliostat should be capable of pointing to within $\pm 2.9 \times 10^{-4}$ rad (± 1 arc-min) for wind velocities of 0 to 2.2 m/s (0 to 5 mph) and to within $\pm 14.5 \times 10^{-4}$ rad (± 5 arc-min) for wind velocities of 2.2 to 13.4 m/s (5 to 30 mph). At wind speeds higher than 13.4 m/s (30 mph), the collector subsystem will be stowed. However, the individual heliostat structure shall be capable of withstanding winds of up to 53.6 m/s (120 mph) while stowed and in an upright position.

e. Foundation - The load criteria for the foundation are summarized below for a uniform 22.3 m/s (50 mph) wind:

- o Base bending moment, 531,365 N-m (37,880 lb-ft);
- o Base shear, 14,145 N (3,180 lb);
- o Torque, 6916 N-m (5,100 lb-ft);
- o Dead load (axial), 19,523 N (4390 lb).

Each heliostat and its accompanying sensor tower will be mounted on a separate foundation, as shown in Figure III-20.

5. Tradeoff Studies and Analyses

During the conceptual design phase, a number of analyses and tests were performed to identify critical design characteristics and to optimize the cost and performance of various assemblies and components. These tests and analyses included those described in Sections 6 through 9, as well as this section.

a. Calorimetric Testing - A calorimeter has been designed, built, and used to test the efficiency of a fifteen mirror heliostat. This calorimetric testing program is the first known test of a concentrating heliostat, and the results substantiate analytical predictions.

(a) $\frac{1}{4}$ Scale Heliostat with
Advanced Breadboard
Sensor Installed

(b) Assembled View of Fine-
Tracking Sensor. Sensitivity
Greater than $\pm 2.909 \times 10^{-4}$
Radians (± 1 arc minute)

(c) Breadboard Control Electronics
for Fine-Tracking Sensor

(d) Disassembled Fine-Tracking
Sensor Showing Shadow
Column

Figure III-19
Details of Advanced Fine-Tracking Sensor and Breadboard Control Electronics

Heliostat Foundation

Sensor Foundation

Figure III-20 Heliostat and Sensor Tower Foundation Interface

Figure III-21 Prototype Calorimeter Schematic

- 1) Calorimeter Design and Operation - The calorimeter used for this initial test program was designed and built by Martin Marietta to substantiate approaches to be used for a large size calorimeter to be built under contract. This effort was accomplished by in-house funding. The calorimeter consists of a water tube receiver plate, a water storage vessel and circulating pump, and instrumentation to measure temperature and water flow rate. Figure III-21 shows a schematic diagram of the calorimeter, and Figure III-22 shows a photograph of the calorimeter during a test run. The water tube receiver plate was made by manifolding three type FS-7610-SW Roll Bond solar panels made by Olin Brass Company. The panels are made of soft aluminum, giving a good thermal conductivity and are finished with "Martin Black" which is a proprietary form of anodizing which provides an absorptance of better than 99%. The manifolding allows thermal expansion and contraction of the panels and equalizes the water flow through each panel. The backs of the panels and the manifolds are well insulated. Provision is incorporated for measuring the water inlet temperature and also the outlet temperature after passing the water through a mixing chamber to insure uniformity. Temperatures are measured by both conventional thermometers and also by platinum resistance temperature sensors that record on paper tape. Water flow is accurately measured by the well substantiated method of diverting the flow into a weigh tank and measuring the time required to flow a given mass such as 100 kg or 100 lbs.

Mirror efficiency is then determined by dividing the energy transferred to the calorimeter (output) by the energy received by the mirrors (input).

$$\text{Input} = (\text{Ins}) (A_m) \cos \epsilon$$

where Ins = Average Solar Insolation

A_m = Area of Mirrors

$\cos \epsilon$ = Cosine of the Angle Between the Sunline and the Normal to the Mirror

$$\text{Output} = (F) (\Delta T)$$

F = Flowrate of Water

ΔT = Temperature Rise Across the Calorimeter.

- 2) Correction of Losses - The receiver plates and the manifold tubing are insulated with fiberglass so that losses can be effectively minimized to radiation and wind conduction on

Figure III-22 Prototype Calorimeter during Test

the black face of the receiver plate. Ideally, both of these losses could be compensated by controlling the flow rate and inlet temperature so that, e.g., the temperature rise is 20° and the inlet temperature is 10° below ambient and the outlet 10° above. This refinement has not yet been incorporated. A correction, which approaches the ideal, is to move the sun off the calorimeter and then repump the hot water from the weigh tank through the panel and measure the loss. The temperature loss thus measured is then assumed to have also occurred during the run and is, therefore, added to the run differential temperature as a correction. This provides a means of correcting for losses with respect to ambient temperature wind velocity and wind direction.

- 3) Test Sequence - When it is desired to make a calorimetric measurement run, the calorimeter is positioned perpendicular to the mirror line at the preset focal distance from the mirror. The pump is turned on and the flow adjusted to a rate which will give approximately 20° temperature rise.

Solar insolation is measured during each run by using three Eppley pyrheliometers mounted on an equatorial tracking mount and positioned to track the sun. During a run, periodic measurements are recorded of the readings of each pyrheliometer.

The mirror is positioned so that the sun's image is focused on the center of the calorimeter panels. When the output temperature has reached a stable temperature, water flow is diverted to the weight tank. The time to flow 100 lbs is then recorded, together with temperatures (in and out) and pyrheliometer readings. 100 lbs is the mass usually flowed, but the criteria is to have a run of at least 100 seconds duration.

A repump correction run is then made in which the mirror is adjusted so that the sun's image is no longer focused on the calorimeter. The weight tank outlet is connected to the pump and valves adjusted to circulate weight tank water through the calorimeter without changing the throttling valve setting so that the flow rate will remain the same as during the measurement run. The temperatures in and out of the calorimeter panel are then recorded to provide the loss correction data.

- 4) Data Analysis - Four or five readings from each of the three pyrheliometers are recorded during a test run. The readings for each pyrheliometer are individually averaged, multiplied

by the calibration constant for the individual instrument, and the resultant values averaged to give the insolation during the run.

Thermometer and thermosensor readings are each individually averaged. The temperature difference (ΔT) is then computed separately for thermometers and for the temperature sensors. These two separate readings provide independent measurements that can be correlated to confirm accuracy.

The power received at the mirror (input) is

$$E_{in} = (Ins) (A_m) \cos \epsilon$$

$$E_{in} \text{ (watts)} = Ins \text{ (L)} 0.0698 \frac{W}{L \text{ cm}^2} \times 15 \text{ mirrors} \times (121.92 \text{ cm})^2 \times \cos \epsilon$$

$$E_{in} \left(\frac{BTU}{hr} \right) = Ins \text{ (L)} 221.296 \frac{BTU}{ft^2 \text{ hr L}} \times 15 \text{ mirrors} \times (4 \text{ ft})^2 \times \cos \epsilon$$

Ins = Average Insolation in Langleys

A_m = Total Area of Mirrors

$\cos \epsilon$ = Cosine of the Angle between the Sunline and the Normal to the Mirror.

MMC has a computer program which uses the arguments of mirror and receiver position, time of day, time of year, and latitude to give this angle (ϵ) and its cosine. This program (Solar Flux Program, MD285) is used in this case for a single heliostat.

The power received at the calorimeter (output) is

$$E_{out} = \text{Flow} \times \Delta T$$

$$E_{out} \text{ (watts)} = \frac{Kg}{sec} (\Delta T^\circ K) 4184.7 \frac{\text{watt sec}}{Kg \text{ Cal}}$$

$$E_{out} \left(\frac{BTU}{hr} \right) = \frac{lbs}{sec} \cdot \frac{3600 \text{ sec}}{hr} (\Delta T^\circ F)$$

It is the practice to compute this for both the delta temperature read and also the delta temperature corrected for repump losses as discussed previously.

The system efficiency is then computed by dividing the BTU/hr received by the BTU/hr available.

- 5) Data from a Typical Day's Test - The data shown on Table III-6 represents the values taken from the data sheets for seven runs made on October 27, 1975. The efficiencies shown on this data sheet for both thermometer and sensor measurements, both direct and corrected for losses, are shown plotted on Figure III-23. The line across the curve at 77.36% represents the predicted efficiency of the 15 mirror heliostat which is made of eight 6 mm mirrors and seven 3 plus 3 mm laminated mirrors. The measured reflective efficiency of the 6 mm mirrors is 72.16%, and that of the 3 plus 3 mm laminated mirrors is 83.36%. The line across the curve at 76.6% represents the predicted efficiency after assuming a one percent loss because the Martin Black is only 99% efficient as an absorber.
- 6) Determination of Energy Concentration in Beam - Tests were made to correlate the computed beam size, including aberration with actuals obtained with our 15-mirror heliostat focused at 102 ft. Aluminum screens were made to cover the front of the calorimeter with different size openings through which the beam could be focused. By making a calorimetric run with a full calorimeter opening and immediately making a second run with the beam passing through the shielding iris, the relative energy distribution in the focal zone can be determined. Calorimetric test pairs were made using iris openings of 15.5, 18, 21, and 24 inches. The results of these tests are shown plotted on Figure III-24. A photograph of the calorimeter with the 15.5 inch iris shield is shown in Figure III-25. In examining this figure, it should be noted that there is a fringe of light showing around the iris opening which appears to be accentuated because the reflectivity of the calorimeter panels is only 1%, whereas that of the aluminum shield is of the order of 50%.

During a calorimeter run, tests were also made with a thermovision infrared camera to determine the temperature gradients and flux patterns on the water filled plates. The thermovision, as used for this test, referenced all temperatures to an initial setting using the water inlet at the bottom of the calorimeter. The instrument was then adjusted to show isotherms or areas of constant temperature at varying values above the reference temperature. Figure III-26 shows a series of photographs indicating hotter and hotter isotherms

Table III-6 Calorimetry Data for Typical Day's Run

Run No.	1	2	3	4	5	6	7
Time of Day (MST)	1040	1050	1100	1121	1132	1151	1155
Avg Pyroheliometer (L)	1.338	1.339	1.340	1.332	1.333	1.337	1.340
Thermometer T (°F)	18.88	18.72	19.84	17.69	18.37	17.91	18.34
Sensor T (°F)	19.05	18.9	19.9	17.9	18.4	18.3	18.5
Thermometer Repump Corrected T (°F)	19.88	21.08	21.34	19.29	19.97	19.41	19.84
Sensor Repump Corrected T (°F)	19.75	20.65	21.2	19.6	19.95	20.18	20.78
H ₂ O Mass (lb)	100	100	100	100	100	100	100
Flow Time (sec)	144.2	144.9	153.1	136.8	140.8	137.8	138.5
Btu/hr at Mirror	71062.41	71115.52	71168.63	70743.75	70796.86	71009.3	71168.63
Cosine	0.935	0.936	0.397	0.939	0.939	0.939	9.939
Corrected Btu/hr at Mirror	66443.35	66564.13	66685.01	66428.38	66478.25	66677.73	66827.35
Btu/hr at Cal. Thermometer	47134.54	46509.32	46651.86	46552.63	46968.75	46789.55	47670.76
Efficiency (%)	70.94	69.87	69.96	70.08	70.65	70.17	71.33
Btu/hr at Cal. Sensor	47558.9	46956.5	46792.95	47105.26	47045.45	47808.42	48086.64
Efficiency (%)	71.5	70.5	70.17	70.91	70.77	71.70	71.96
Btu/hr at Cal. Thermal Corrected	49631.06	49490.6	50178.96	50763.15	51059.66	51439.76	52284.47
Efficiency (%)	74.69	74.35	75.24	76.42	76.8	77.14	78.23
Btu/hr at Cal. Sensor Corrected	49306.5	49192.55	49849.77	51578.94	50752.84	51727.14	51985.55
Efficiency (%)	74.2	73.9	74.75	77.64	76.34	77.58	77.79

Figure III-23 Calorimetric Measurement of Mirror Efficiency

Figure III-24 Determination of Energy Concentration in Beam

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Figure III-25 Calorimeter with 39.37 cm (15.5 in.) Iris

Figure III-26 Thermovision Pictures of Solar Image on Calorimeter

in progression from left to right. The isotherm is defined by the white pattern within the black background and represents a thermal band of approximately 10°C in width for the particular setting of the instrument used for this test. Evaluation of the data from these photographs across the middle elevation of the solar image provides a temperature pattern as shown in Figure III-27. The abscissa represents the width of the calorimeter plate at the center of the beam in elevation. The ordinate is then the temperature as measured.

These preliminary tests with the thermovision show that it is a potential tool for evaluating flux patterns and also for evaluating measurement losses.

- 7) Problems and Solutions - Problems that have been encountered and solved during the prototype calorimetric testing have provided the knowledge that will insure minimum problems when the full scale calorimeter is built. Some of these problems are discussed below.

a) Thermocouples - Copper constantan thermocouples were tried to measure temperature differences. The thermocouple is a valid means of making differential temperature measurements although thermocouple wire with accuracies greater than 1°C must be special-ordered or individually calibrated. The major problem of using thermocouples is the electronics to read to microvolts potential difference. Digital voltmeters initially rack mounted on the calorimeter were required to be in a temperature controlled building to achieve stable operation. Later, a digital system using platinum resistance sensors capable of obtaining an instantaneous permanent record printout was used.

b) Output Water Mixing - The water from each of the three panels entered the output manifold and even though it traveled some distance after the last entry point before reaching the temperature sensors, was not always well mixed. Thorough mixing was accomplished by a combination of inserting flow-spoilers in the tube and inserting two 90° bends prior to temperature sensing.

c) Discontinuous Data - It was necessary to periodically stop testing to empty the weight tank. During the emptying process, the mirror was deoriented to prevent continual heating of the supply water. To eliminate this problem, a dump tank was installed into which the weigh tank can be

Figure III-27
Temperature Profile Across Solar Image

Figure III-28
Specular Reflectivity Test Rig Using
Normal-Incidence Pyrheliometers,
Profile View

Figure III-29
Specular Reflectivity Test
Rig Using Normal Incidence
Pyrheliometers, Front View

quickly emptied to permit a succeeding run. The dump tank can be stored or pumped back to the supply tank at a more convenient time.

b. Mirror Tradeoff Study - In an effort to obtain comparative data for several candidate mirror surfaces with specular collimation similar to that of the solar monitoring instrumentation, the reflectivity test rig shown in Figures III-28 and III-29 was assembled. The reflected beam from the test mirror is monitored by the reverse-mounted pyrhelimeter. To avoid shadowing, the mirror sample is tested with a 10-deg tilt ($\cos = 0.98481$). The reference insolation input to the mirror is monitored by the pair of independently tracked pyrhelimeters shown in Figure III-29.

The baseline design specular reflectivity necessary to be consistent with the pilot plant field sizing is 85%. For a given flux into the receiver cavity, the field size is approximately inversely proportional to the spectral reflectivity of the mirror (Figure III-30). Although the mirror with the highest reflectivity has the highest cost per unit area, the net cost saved by the reduction in number of heliostats required justifies its use.

The results of specular reflectivity tests completed to date are shown in Table III-7. From the samples tested, it is apparent that a specular reflectivity of 85% is only attainable with either first-surface aluminized teflon or with a laminated second-surface mirror with glass of improved transmissivity.

The second surface silvered white glass laminated mirrors were chosen over the aluminized teflon for reasons of producibility and reliability. First-surface mirrors, such as the aluminized teflon, show a greater surface degradation rate than a second-surface silvered glass mirror. Also, special cleaning fluids and techniques would probably be necessary to avoid damage of the aluminum surface. Samples of second-surface silver on teflon have not been available from the manufacturer, although diligent efforts to obtain them have been made over a 13-month period.

The laminated white glass mirrors, which exhibit considerably higher spectral reflectivity will, however, be available in the United States within a one to two year period. Therefore, the subsystem research experiment will use two variations of heliostat to verify different parameters. The first variation, using 25 1.2 x 1.2 m variable focus mirrors, will measure performance

Figure III-30 Effect of Mirror Reflectivity on Field Size

Table III-7
Results of Terrestrial Spectral Reflectivity
Tests of Candidate Mirror Surfaces as Compared
with Cost

Description of Mirror Sample	Test Date	Specular Reflectivity, %			Cost, \$/M ² (\$/ft ²)
		High	Low	Average	
First-Surface Aluminized Teflon (Reference Surface)	10/01/74	86.5	86.1	86.27	Unknown, But Is Probably Greater Than \$10.76/m ² (\$1/ft ²)
	10/08/74	86.6	83.5	85.11	
Second-Surface Silvered Commercial Float Glass Laminated Mirror, 3-mm Front to Mirrored Surface	09/20/74	83.1	82.7	82.90	\$18.84/m ² (\$1.75/ft ²) from Gardner Mirror
	09/23/74	83.9	83.3	83.56	
	10/08/74	84.1	82.7	83.47	
Second-Surface Silvered White Glass Laminated Mirror, 3-mm Front to Mirrored Surface	03/13/75	90.7	89.7	90.2	Estimated \$18.84/m ² (1.75/ft ²) from Gardner Mirror and Forcut Company
	03/13/75	93.9	90.8	92.3	
	03/13/75	94.4	91.2	92.8	
Second-Surface Silvered Commercial Float Glass 6-mm Thick	09/20/74	72.2	70.5	71.27	\$10.76/m ² (\$1/ft ²) Gardner Mirror
	10/08/74	73.3	72.7	73.05	
Second-Surface Silver On Acrylic, 3.2-mm Thick	09/23/74	79.3	78.3	78.7	Unknown, But Is Probably Greater Than \$10.76/m ² (\$1/ft ²)

using the high reflectivity glass (91%). The second variation, which is representative of the proposed pilot plant design, uses nine 2.03 x 2.03 m (6.7 x 6.7 ft) fixed curvature mirrors. Using conventional float glass mirrors with spectral reflectivities of 71 to 73%, this heliostat will be used to analyze the producibility and structural and mechanical characteristics of the proposed design for the pilot plant.

c. Structural Tradeoffs - All structural members in the heliostat design have been subjected to optimization to maximize performance and decrease cost. An example of cost optimization during the conceptual and detail design phases is illustrated by the proper selection of a structural cost section to meet requirements for a member in the yoke assembly.

For a specific moment of inertia, which was the desirable characteristic being evaluated, the relative direct costs and resulting weights of the various cross-sections are shown in Tables III-8 and III-9. The favorable cost and weight characteristics of the I-beam were a major factor in its selection for the vertical yoke member.

Another example of performance optimization is shown in Figure III-30. Originally, the mirror support for each mirror consisted of a single vertical beam as shown in Figure III-31(a). Actual experience with the single beam concept proved that this design did not provide either the required rigidity or ease of pointing.

A second design (Figure III-31(b)) used two vertical beams to support each mirror column. The three point attachment scheme provides two degrees of rotation about orthogonal pivot axes and is simple to adjust. Additionally, the total weight of the mirror support assembly decreased, since each of the two members could be significantly smaller and lighter than a single member in order to retain the required stiffness. The two beam concept also increased the natural frequency of the structure considerably.

d. Performance and Cost of Fixed Curvature vs. Variable Curvature Mirrors - The estimated production cost of a typical 2.03 x 2.03 m honeycomb-substrate mirror is shown in Table III-10. The estimated cost of \$33.74/m² (\$3.13/ft²) was derived from the individual costs of each of the component materials, assuming that several thousand units would be fabricated.

Table III-8 Example of Structural Cost Optimization

OR		OR		OR
Yoke Assembly							
Moment of Inertia 158 (Ix)			150 (Ix)		148 (Ix)		161
kg/m (lb/ft)	43.18 (29)		52.82 (35.49)		47.96 (32.23)		60.24 (40.48)
kg (lb)	158.76 (350)		194.59 (429)		176.45 (389)		221.81 (489)
Cost/kg (Cost/lb)	\$0.46 (\$0.21)		\$0.53 (\$0.24)		\$0.53 (\$0.24)		\$0.53 (\$0.24)
Cost	\$73.03		\$103.13		\$93.52		\$117.56

Table III-9

Cost and Performance Optimization of the Vertical Yoke Member

Cross-Section	Relative Material Cost	Relative Member Weights
I-Beam	1	1
Rectangular Tube	1.40	1.22
Square Tube	1.27	1.11
Round Tube	1.59	1.40

Figure III-31 Mirror Assembly Support/Positioning Tradeoff

Table III-10

Estimated Production Cost of 2.03x2.03 m Honeycomb Substrate Mirrors

Component	Description	Weight, kg (lb)	Cost (\$)
Mirror	3 mm, 2.03x2.03 m (80x80 in.), Rear Surface Silvered Glass	31.3 (69)	\$ 44.44
Paper Core	12.7 mm (½ in.) cell, 0.076x2.03x2.03 m (3x80x80 in.)	6.4 (14)	18.80
Glass Cloth	3 Layers, 0.38 mm (0.015 in.) F1564 Cloth	8.6 (19)	23.23
Inserts	3 Pcs, 6061 Bar, 76x76x76 mm (3x3x3 in.)	3.7 (8.1)	8.10
Adhesives	Epoxy Resin	7.6 (16.7)	16.77
Total Material		57.5 (126.8)	111.34
Estimated Assy (Labor)		--	28.00
Total*		57.5 (126.8)	\$139.34*
Resulting Cost = \$139.34/4.129 m ² = \$3.14/ft ² * = \$33.74/m ² *			
*Does not include estimated cost for capital equipment, profit, etc.			

Table III-11 Comparison of Adjustable and Fixed Curvature Mirror

Characteristic	Adjustable Curvature 1.2x1.2 m (4x4 ft)	Fixed Curvature - Honeycomb 2.03x2.03 m (6.7x6.7 ft)
Weight	26.7 kg/m ² (5.47 lb/ft ²)	13.95 kg/m ² (1.30 lb/ft ²)
Cost	\$55.21/m ² (5.13/ft ²)	\$33.80/m ² (43.13/ft ²)

A comparison of the estimated costs for production of a 1.2 x 1.2 m (4 x 4 ft) variable-focus mirror (see Section c), a 1.2 x 1.2 m honeycomb substrate mirror, and a 2.03 x 2.03 m honeycomb substrate mirror is shown in Table III-11. The two honeycomb substrate mirrors are of fixed curvature.

6. Wind Tunnel Tests

Wind tunnel tests were required to determine the wind loads imposed on the Martin Marietta configuration of the collector (heliostat) in the central receiver solar thermal power system program. Five 1/10th scale collector models were tested in the Colorado State University Industrial Aerodynamics Wind Tunnel at Fort Collins, Colorado. The models comprised simulated metal-backed variable curvature and honeycomb backed fixed curvature mirrors in 25-mirror and 9-mirror configurations, as well as a large rectangular single-mirror configuration.

Sufficient results were obtained to define critical loads on the baseline configuration and on configurations similar to the tested configurations. Particular emphasis was made to obtain data that could be used to verify and/or modify existing methods of load analyses. This was accomplished and the results of the modified analysis are presented.

At the time of this test, the Martin Marietta baseline pilot plant configuration for the collector for the central receiver solar thermal power system was essentially a "billboard" consisting of an array of either 25 or 9 mirrors. These mirrors were arranged in a rectangular pattern with axes of rotation about the horizontal and vertical centerlines of the rectangle. Since the "billboard" is exposed to the environment, and in particular, wind loads, determination of realistic wind loading was important for a cost effective design for the collector.

The collector extends 6.8 m (22.5 ft) above the ground and in the operational mode may experience wind up to 22.35 m/s (50 mph) measured at 9.06 m (30 ft) above the ground. This wind velocity may occur with the collector rotated to any combination of elevation and/or azimuth angles relative to the wind.

Maximum base loads such as base shear and base bending may be conservatively calculated by assuming a uniform wind normal to the plane of the mirrors. This is not true, however, of other important wind loads. Base or azimuth torque, elevation torque, and lift require knowledge of the resultant normal force and

center of pressure of this force on the mirror assembly. The effect of a nonuniform vertical wind profile on elevation torque may also be significant.

The most appropriate method of load analysis available, prior to this test, was to treat the mirror array as a low-aspect-ratio rectangular wing. The analysis is empirical and is based on the results of wind tunnel testing that had been done as far back as 1930. Results of the analysis include coefficients of lift, drag, and normal force, and the center of pressure at which the normal force acts. From these could be determined the required load on the collector structure; however, the analysis required an assumption of the angle of attack at which flow separation occurred. This assumption has a significant effect on the resulting loads as illustrated in Figure III-32. Varying the separation angle from 0 to $\pi/6$ rad (30 deg) in increments of $\pi/18$ rad (10 deg) increases the torque 41, 193, and 339%, respectively. These loads directly affect the design of the collector drive mechanisms. Therefore, it became necessary to experimentally determine the correct separation angle for the proposed mirror array.

The wind tunnel test was conducted to determine the answer to the question concerning separation angle. In detail, the objectives of the wind tunnel tests were to:

- o Establish the correct separation angle for the collector configuration;
- o Evaluate the Reynolds number effect, if any, on the collector configuration;
- o Obtain data to verify and/or modify the existing method of load analyses;
- o Evaluate the effects of minor changes to the baseline mirror array such as different size mirrors, varying the space between mirrors, etc.

a. Wind Tunnel Test Facility - The test facility used was the Industrial Aerodynamic Wind Tunnel at the Fluid Dynamics Laboratory at Colorado State University in Fort Collins, Colorado. The wind tunnel provided a 1.8 x 1.8 m (6 x 6 ft) test section. Vertical velocity profiles in the test section were thoroughly surveyed and modified by the addition of roughness of the floor of the tunnel in front of the model. The profile selected for

Figure III-32 Analytical Wind Load Predictions

Figure III-33
Wind Tunnel Testing - Test Velocity Profile

this series of test is illustrated in Figure III-33. This figure also shows the standard wind profile predictions for desert ($V = V_{36} (Z/36)^{.14}$) and suburban ($V = V_{36} (Z/36)^{0.233}$) locations. The test profile falls between these two profiles. Since smooth floor profile approaches a uniform wind, it was felt it would have a conservative and undesirable effect on elevation torque.

Velocity measurements in the tunnel were provided through a pitot tube mounted 121.9 cm (48 in.) above the tunnel floor and immediately above the model. Scale-wise, the velocity measurement should have been at 91.4 cm (36 in.) so corresponding adjustments of test velocities on the models were required.

A turntable was provided in the floor of the tunnel. The model was mounted at the center of the turntable allowing a 360 degree azimuth rotation, if desired. Instrumentation wires were fed out through the base of the turntable.

The base model was a 1/10 scale model of the 25-mirror configuration collector (Figure III-34). The primary structure was made of solid steel round or rectangular bars, made as closely as possible to 1/10 scale, from material readily available.

The mirrors were sheared from 0.15 cm (0.060 in.) steel sheet stock. Rings were soldered to the backs of the mirrors and wooden diagonals bonded to the backs to simulate actual structure. The mirrors were attached to the primary structure by screws through the center of the mirrors. The collector yoke was attached to a six-component Task balance and, in turn, to the turntable in the tunnel floor. The six-component balance was oriented so the N-forces were normal to the plane of the mirrors.

In addition to the base model, four modifications of the base model were tested. The first modification consisted of replacing the 25 metal mirrors with 25 thick mirrors. The mirrors were made from 3/16 fiberboard and simulated proposed sandwich-backed mirrors. Modifications 2 and 3 were 9-mirror configurations. The vertical arms of the collector yoke were each moved in 7.62 cm (3.0 in.). Two mirror support arms were removed and the remaining arms were used to support the mirrors through center screws. Two mirror configurations were used--metal mirrors constructed as described for the 25 metal mirror configurations and 1/4-inch fiberboard mirrors. The last configuration consisted of 60.9 x 60.9 cm (24 x 24 in.) piece of 1/4 in. plywood. The plywood was attached to the five mirror support arms of the base configuration.

25-Mirror Model

9-Mirror Model

Figure III-34 Wind Tunnel Model and Test Setup

Figure III-35
Wind Tunnel Test Results - 22.4 m/s (50-mph Wind)

b. Test Setup - The test setup consisted of mounting the model on the turntable in the floor of the wind tunnel. Instrumentation wires ran from the Task balance through the bottom of the turntable mechanism to an eight channel Brush recorder adjacent to the wind tunnel.

Zero azimuth angle was established utilizing a laser beam. The laser was positioned at a reference point at the far end of the tunnel test section. The beam was reflected off a mirror on the elevation axis of the model back to a target at the laser.

Zero elevation angle was established utilizing an adjustable combination level-square. Each arm was checked for zero elevation angle and attached rigidly to the elevation axis. All elevation rotation subsequent to this was accomplished by rotating the entire mirror assembly as a unit.

The task balance was calibrated by applying loads to the model and adjusting the recorder response to the scale desired. Reactions N_A and N_F were calibrated by applying a force parallel to N_A and N_F at the center of the elevation axis. Reactions Y_A and Y_F were calibrated by applying a force parallel to Y_A and Y_F at the end of the elevation axis. Base torque (R) was calibrated by applying two loads to the upper ends of the model yoke. The loads were applied normal to the plane of the yoke and in opposite directions, resulting in a torque without lateral load in the balance. Lift (X) was calibrated by applying a downward force at the center of the model yoke.

The baseline model was mounted on the wind tunnel turntable with the six-component Task balance in place. The model was positioned at zero azimuth and elevation angle and the tunnel started. The tunnel wind velocity was stabilized at a reference velocity of approximately 13.5 m/s (30 mph) and the wind velocity recorded on the stripchart. The model was then rotated in azimuth to the angles 0, 20, 40, 50, 60, 70, 80, 90, 100, 110, 120, 130, 140, 160, and 180 deg. At each azimuth angle (α), there was a pause in azimuth rotation and the angle was recorded on the stripchart. Rotation in azimuth continued to an angle of $\alpha = 180$ deg. At this point, the model was rotated continuously back to $\alpha = 0$ deg. During this rotation, the recorder remained on and the azimuth angles were indicated on the stripchart. There was no pause at the various azimuth angles, but a second "check" reading was made available in this manner.

After the above azimuth survey, the tunnel was stopped and the model positioned to a new elevation angle. This required entering the tunnel and manually positioning the model to the desired elevation angle. The tunnel was then started, stabilized, and the model rotated in azimuth as described above.

The above procedure continued until elevation angles (β) of 0, 20, 40, 60, 70, 80, 90, 200, 220, 230, 240, 250, 260, and 270 deg were tested. The entire series was conducted at a reference velocity of approximately 13.5 m/s (44 fps). Tunnel velocities and temperatures were recorded for each β setting and are reflected in the reduced data.

The above β angles actually encompass the entire range of angles, including those from 90 to 200 and 270 to 360 deg. Testing at $\beta = 20$ deg gives the results for $\beta = 160$ deg, with essentially a reverse shift in α . That is, the results of $\alpha = 0$ and $\beta = 20$ are the same as $\alpha = 150$, $\beta = 160$ deg. Since each of the β angles tested has a "mirror" angle, the entire range was tested.

A test was then conducted to evaluate the Reynolds number effect on the base model. The model was returned to the original position ($\alpha = \beta = 0$ deg). The tunnel velocity was increased to approximately 17.7 m/s (40 mph). This represented an upper load limit on the six-component balance because of the manner in which it was mounted and calibrated. After tunnel stabilization, the model was rotated in azimuth as in the original survey with the corresponding stripchart recordings.

Four modifications of the base model were then introduced and each was tested to this procedure. As before, reference velocity and tunnel temperature were recorded on the stripchart for each setting.

c. Test Results - Test results are given in Figure III-35. Results have been scaled up to a full-size collector as well as corrected to a wind velocity of 22.4 m/s (50 mph) at 9.2 m (30 ft) above the ground. The lift curve shown in Figure III-35 indicates that the separation angle must be equal to, or fairly close to, zero. A nonzero separation angle would show a spike in the data on either side of $\alpha = 90$ and 270 deg. Lack of this spike was apparent as the data were being recorded. The N_A , N_F , Y_A , Y_F , and R readings from the six-component balance would have shown this type of response, but none were evident.

Although the azimuth torque curve (Figure III-35) could be interpreted to show a peaking of the data at $\alpha = 80$ deg and thus a separation angle of 20 deg, this is inconsistent with the lateral and vertical force data and the high level of data between 30 and 60 deg on this plot. Continuing the assumption of a separation angle = 0 deg appears to give an analytical prediction curve that best fits the available data.

d. Load Analysis Program Modification - The original computer program was written to determine force coefficients for a low-aspect-ratio rectangular wing. The initial modification was to include the capability of calculating the desired forces and torques on the collector for a nonuniform velocity profile. The nonuniform velocity was handled by using an integrated average velocity based on the elevation angle. Since the wind tunnel test established the separation angle to be 0 deg, the load curve shapes indicated in Figure III-35 were established.

The final program modification was to add multiplying factors to the elevation and azimuth torque calculations. These multiplying factors were 0.9528 and 0.9189 for azimuth and elevation, respectively. This reduced the calculated torques to the test values shown in Figure III-35.

The modified analysis program was used to calculate loads for various collector configurations and for various velocity profiles. The results for the baseline are given in Figures III-36 and III-37.

e. Conclusions - The wind tunnel tests demonstrated that the configurations tested had a separation angle of 0 deg and that there was no discernable Reynolds number effect. The modified load program may be used to predict loads for similar collector configurations. Important factors in using this program are to include the entire frontal area of the collector and to use a similar mirror spacing. The mirror spacing tested was a scaled 7.62 cm (3.0 in.). The one test conducted to evaluate mirror spacing was inconclusive, but did indicate an area of concern. If it is necessary to increase mirror spacing appreciably, care should be taken in applying the predicted loads. If possible, additional testing would be desirable.

7. Design Mechanism Tradeoff Study

The many possible design solutions for the drive mechanisms fall into two basic categories characterized by the type of tracking

NOTE: AZIMUTH & ELEVATION TORQUES: 22.35 m/s (50 mph)

Figure III-36 Design Loads, 25-Mirror Configuration

NOTE: BASE BENDING MOMENT: 50 mph
25 MIRROR CONFIGURATION

NOTE: LIFT & BASE SHEAR - 50 mph

LEGEND:
--- BASE SHEAR lbs
— LIFT, lbs

NOTE: 25-MIRROR CONFIGURATION,
 $V_z = 50(z/30)^{0.14}$

NOTE: 25 MIRROR CONFIGURATION,
 $V_z = 50(z/30)^{0.233}$

Figure III-37 Design Loads

error they introduce:

- o Backlash Error Systems - Where a tooth mesh involves backlash but negligible elasticity;
- o Elastic Error Systems - Where a roller chain, ball screw, cable, flex spine, or volume of hydraulic fluid involves elastic deflection. The results of this tradeoff study are summarized in Table III-12.

The direct gear drive was shown to be the most cost effective solution. The primary reason for this was the high cost involved in beefing-up the elastic members of competitive designs to achieve acceptable deflection characteristics.

A previously undefined advantage of the direct gear drive is that from a dynamics viewpoint, the involvement of backlash, which is normally nullified by aerodynamic torque, is preferable to the involvement of elastic elements.

a. Design Requirements - The wind tunnel testing reported in Section III-B6 established the performance requirements presented in Table III-13.

Figure III-38 illustrates those torque requirements in the form of tangential load versus radius. The supplementary abscissa scales relate worm reducer capability. The right hand ordinate defines the amount of circumferential motion at a given radius equal to 1 arc minute, and thus, gives an immediate idea of the extent of backlash or elasticity that can be tolerated at a given radius.

The chain pull abscissa indicates the considerable increase of tooth load capability using a hollow shaft reducer (providing rated torque is not exceeded).

Figure III-39 applies strictly to a geared final drive system, illustrating the degree of tracking error at a 13.41 m/s (30 mph) wind velocity that could be introduced by gear mesh clearance. This error would be realized only during the limited periods when wind veering in conjunction with heliostat angle would cause a reversal of aerodynamic torque.

The design operating points given for the various sizes of gear reducer are based on the use of unheat-treated commercial gears as discussed in the following subsection. The reflected back-

Table III-12 Comparison of Attributes of Various Drive Systems

Type of Tracking Error	Backlash		Elastic Deflection					
Drive Method Attribute	Direct Gear Drive	Gearbox Mounted	Chain and Sprocket	Wire Rope	Harmonic Drive	Linear (Preloaded) Ballscrew	Hydraulic: Rotary Vane Actuator with Metering Pumps	Hydraulic: Dual Cylinder
Hardware Development Effort	Minimal	Zero	Medium	Medium/High	Medium/Low	Minimal	High	High
Potential Problem Areas	None Anticipated	None Anticipated	Tensioning Mechanisms to Code with Tracking Accuracy, Bidirection Operation, & Veering Winds		Compliance of Flex Spine	Difficult to Apply to 3.84 rad (220 deg) Angular Motion	Requires Constant Pressurization & Low Operating Pressure for Tracking Accuracy Due to Oil Compressibility	
Adaptability to Elevation Drive	Excellent	Poor	Acceptable	Poor	Poor	Good	Good	Medium
Size	Excellent	Ok for Azimuth	Satisfactory	Cumbersomely Large	Requires 2 ft Extra Height	Excellent	Excellent	Satisfactory
Size of Work Reducer, CD	8.89 cm (3 1/2 in.)	20.32 cm (8 in.)	8.89 cm (3 1/2 in.)	7.6 to 8.89 cm	5.08 cm (2 in.)	Included in Actuator	N/A	N/A
Maintenance Effort	Minimal	Minimal	Medium	High	Minimal	Minimal	?	?
Tracking Capability, arc-min	1 1/4 Backlash	1.6	2 to 7 at 13.5 cm (30 mph)	3 to 5 1/2 at 13.5 cm (30 mph)	Flexspline Elasticity, TBS	Very Low ~1	Depends on Actuator Size, ~2	Depends on Actuator Size ~2
Cost Factor	1	2.0	1.25 to 1.6	1.3 to 1.4	2.0	1.7	?	?

Table III-13 Performance Requirement for Drive Mechanism

Base Bending Moment (Azimuth) at 22.3 m/s (50 mph)	456.39 kg-m (39,600-lb)
Aerodynamic Torque at 22.3 m/s (50 mph)(Slew Only)	673.06 kg-m (58,400 in.-lb)
Aerodynamic Torque at 13.5 m/s (30 mph)(Max Tracking Torques)	242.02 kg-m (21,000 in.-lb)
Aerodynamic Lift at 22.3 m/s (50 mph)	Positive 634.2 kg (1400 lb) Negative 634.2 kg (1400 lb)
Slew Speed	13.4 rad/hr (755 deg/hr)
Tracking Speed, Azimuth	1.5 rad/hr (89 deg/hr)
Angular Displacement of Heliostat, Azimuth	$\pm 1.91 \pm 0.09$ rad ($\pm 110 \pm 5$ deg)
Angular Displacement of Heliostat, Elevation	-4.71 ± 0.09 rad (-270 ± 5 deg)
Tracking Increment	1 arc-min
Weight to be Supported	2718 kg (6000 lb)
Drive Must Be Irreversible (Cannot Be Back-Driven)

Figure III-38 Tangential Load vs Radius

Figure III-39 Backlash vs Pitch Radius

Figure III-40
Direct Gear Drive Concepts Using Yoke
Mounted Reducers

lash from the worm reducer is also indicated and is additive to that defined in the abscissa.

b. Direct Gear Drive Concepts

- 1) Direct Gear Drive Concepts Using Yoke-Mounted Reducers - The design points defined in Figures III-38 and III-39 are exemplified in Figure III-40 using yoke-mounted reducers and stationary unheat-treated gears (pinions heat treated). The backlash-induced tracking errors delineated assume a maximum tooth clearance of 0.025 cm (0.010 in.), which is probably optimistic for such large commercial gears. Obviously, these concepts are adaptable for such large commercial gears.

These particular versions of the direct gear drive, utilizing commercially available gears, pinions, and worm reducers, did not prove cost effective in comparison with the use of turntable-type bearings having integral high strength gear teeth (Figure III-41) or the modular approach of Figure III-42.

- 2) Direct Gear Drive Using Turntable Bearing - The concept shown in Figure III-41 represents a very conventional approach for precision tracking systems. The turntable-type preloaded bearing, however, is quite expensive and must be mounted without distortion, which is the reason for introducing the stiff machined mounting ring. Similar precision must be maintained on the mounting plate attached to the yoke.

With this arrangement, a mesh clearance of 0.007 cm (0.003 in.) maximum is practical, representing a $1\frac{1}{2}$ arc-minute potential tracking error at a 13.4 m/s (30 mph) wind velocity (when backlash is taken up by reversing aerodynamic torques).

- 3) Low Duty Cycle Consideration - The total number of tooth meshes involved in normal duty plus emergency stowage operations and maintenance amount to approximately 26,100. Bearing in mind that tooth load is proportional to wind velocity, Table III-14 defines load distribution based on the most severe wind velocity history of any recorded site in the southwest desert terrain of the United States.

In the light of these statistics, it is unnecessary to utilize the fatigue stress values normally associated with AGMA tooth strength calculations. Instead, values approaching the normal yield may be used, e.g., 65% of yield.

Figure III-41 Direct Gear Drive Using Turntable Bearing

Figure III-42
Direct Gear Drive Using Preloaded
Tapered Roller Bearings

Table III-14

Gear Meshes as Function of Wind Velocity

Wind Velocity	Gear Meshes
14% at Winds Reaching 1.78 m/s (4 mph)	3,645
58% at Winds Reaching 7.15 m/s (16 mph)	15,066
23% at Winds Reaching 14.3 m/s (32 mph)	5,589
7% at Winds Between 14.3 m/s (32 mph) and 22.35 m/s (50 mph)	1,800

Figure III-43 Typical Chain and Sprocket Drive

- 4) Direct Gear Drive Using Preloaded Taper Roller Bearings - This concept which is shown in Figure III-42 utilizes a nodular iron casting (ductile iron) to AGMA grade N15 and tooth strength equated to 65% of yield as previously discussed. It provides a very rigid combination mounting and final drive of utmost simplicity. The same casting also houses two preloaded tapered roller bearings. The bearings cost an estimated 1/7 of the turntable bearing previously illustrated (because they are a high production item) and incur less deflection. For the cost balance, the remainder of the drive can be provided in a configuration readily adaptable to the elevation drive. The nodular iron casting is heat treated to 225 BHN and the steel pinion to 300 BHN, leaving an optimum hardness differential. An accurate gear geometry and pitch line concentricity will be assured by precision gear finishing techniques.

An important advantage of the gear box concept, compared to chain and cable systems, is that it comprises an independent interchangeable functional module. Eight bolts attach it to the foundation, and it is then operational. No time consuming on site assembly operations are involved. The drive will be enclosed and completely sealed from the environment.

Lubrication nipples will service both bearings and gear mesh.

c. Chain and Sprocket Drives - The determining criterion for a chain drive system (Figure III-43) is chain elasticity and strain energy, as is in the case of a cable system. Irreversibility of the drive gearbox is, of course, essential in any case; and the chain represents an elastic linkage to the heliostat that the aerodynamic torque will strain proportionally to wind velocity. An increase in wind velocity will create a tracking error by stretching the chain; a reduction in wind velocity reduces chain tension and permits the strain energy of the chain to diminish chain stretch and again effect a tracking error.

The incorporation of a brake is of no advantage in a chain drive system. (In a nonelastic gear driven system, it would successfully eliminate backlash.) The brake would presumably be set to maximum tracking torque at 13.4 m/s (30 mph). The drive mechanism torque would then have to be twice this value and chain strain energy will also be doubled, thus reintroducing the wind velocity sensitivity previously described. These elasticity induced errors are in direct proportion to the size of chain selected and size of sprocket.

An apparent simplification that could be incorporated would be to use a toothless sprocket. Unfortunately, the additional length of chain now exposed to tension introduces greater elastic stretch and tracking errors. The use of an even larger chain failed to produce a cost effective solution.

These complications, combined with the need for a bidirectional tensioning device, a well-sealed enclosure, and assured lubrication, plus a high cost estimate, gave this type of system a low rating.

d. Cable and Capstan

- 1) 3 Arc-Minute Accuracy - Figure III-44 schematically illustrates a relatively simple arrangement for tracking control. A virtual sheave comprising nonrotating pulleys is provided on the foundation. The reducer-driven capstan mounts to the yoke and winds itself around the stationary pulleys. The arrangement, however, is particularly inconvenient for adaptation to the elevation axis.

Appreciable slack side tension must be provided to minimize the necessary wraps on the capstan. This is complicated (even in the case of azimuth drive where tracking accuracy is necessary in only one direction of rotation) due to the reversibility of aerodynamic torques. Cable creep can also prove troublesome.

- 2) 5 Arc-Minute Accuracy - Figure III-45 schematically illustrates a tensioning arrangement that could satisfy the bidirectional requirement previously discussed. A dual spring-loaded pulley arrangement such as used on ferris wheels might be a satisfactory alternative. This figure also depicts a smaller sheave that could be adaptable to the elevation axis drive. However, it would still be clumsy and involve considerable shadowing of mirrors.
- 3) Tracking Error vs. Sheave Diameter - Figure III-46 illustrates the relationship between cable elasticity in terms of arc-minute tracking error and sheave diameter for a 1.42 cm (9/16 in.) diameter unlubricated wire rope. A tension ratio of 5.5 was selected as a compromise between number of capstan wraps and "slack side" tension. Note that it is the deflection (stretch), not strength, of the rope that is important.

Figure III-44
Cable and Capstan Drive - 3 arc-minute Accuracy

Figure III-45
Cable and Capstan Drive - 5 arc-minute Accuracy

Figure III-46
 Tracking Error vs Sheave Diameter for Cable and Capstan Drives

Figure III-47
 Cost vs Tracking Error, Gear, Cable and Chain and Sprocket Drives

In general, it was concluded that although the tracking error at low wind velocities is less than the potential backlash error of a geared system, the unwieldiness of a cable system, combined with higher cost and its poor adaptability as an elevation drive mechanism eliminates it from serious consideration. Also, the difficulty and expense of enclosing it from the environments renders it a high maintenance risk.

e. Cost vs. Tracking Error Comparison - Figure III-47 shows that although the systems subject to elasticity errors, i.e., the chain and cable systems can yield relatively low errors at the lower wind velocities, these are indeed errors.

The nonelastic gear systems are merely subject to potential backlash errors of the magnitude shown. The magnitude shown of $1\frac{1}{2}$ arc-minute cannot be directly compared with the elastic errors of the other systems, since this will only be realized on reversal of aerodynamic torque. As long as the direction of the torque remains constant, regardless of wind velocity, there will be no induced error from the gearing (the elasticity of the teeth being insignificant). However, on reversal of aerodynamic torque, the backlash will introduce a tracking error. The extent, however, is limited to $1\frac{1}{2}$ arc-minute regardless of the magnitude of wind velocity, whereas, in the case of the elastic systems, the total error could be twice that illustrated. In steady winds, the torque reversal will occur once as the heliostat tracks through the zero angle of attack. Unsteady winds will produce elastically induced tracking errors equal to those given on the graph, whereas the geared system is wind-velocity insensitive.

The cost factor points to the direct drive gear system as the most cost effective.

f. Hydraulic System Evaluation - Like the cable and chain drive systems, the hydrostatic design approach suffers from elasticity shortcomings due to oil compressibility. The analysis of Figure III-48 simply shows that because a given tracking accuracy and bulk modulus predetermines the maximum permissible operating pressure for any particular operating radius, the minimum size of operating cylinder is also fixed.

If the required motions of the two axes had not exceeded approximately 110° , a simple hydraulic cylinder operated crank would have been a viable solution using a pressurized return and miniature metering pumps operating through check valves into either side of the piston. To accommodate the required 270°

Force Induced
by Aerodynamic
Torque

Induced Deflection = Allowable
Tracking
Tolerance

$$\text{Volumetric Strain} = \frac{\text{Allowable Tracking Tolerance}}{\text{Total Heliostat Motion}}$$

$$\text{Bulk Modules} = \frac{\text{Stress (Pressure)}}{\text{Volumetric Strain}}$$

Note: Error/Pressure Relationships vs
Moment Arm & Cylinder Bore at
13.41 m/s (30 mph) Wind Velocity
& 21,109 kg/cm² (300,000 psi)
Bulk Modulus.

Figure III-48 Hydraulic Compressibility-Induced Errors

Figure III-49 Direct Coupled Gearboxes in Comparison with Baseline

motion, a dual cylinder arrangement or a vane type actuator can be used. Both are regarded as sufficiently developmental to preclude serious consideration at the present.

g. Direct Coupled Gearboxes - Comparison with Baseline Design - Probably the simplest drive solution is to mount the heliostat on a large worm reducer with output shaft bearings of sufficient capacity to accept the base bending moment of 5476 kg-m (39,600 ft-lb). This approach, shown in Figure III-49, is compared with a harmonic drive installation and the direct gear drive system, previously described, using back-to-back tapered roller bearings. The harmonic drive will introduce flex spline compliance (of an undetermined amount) and is not irreversible, necessitating a worm reducer to drive it.

The compactness of the direct gear drive system, as opposed to the bulkiness of the harmonic drive system, is obvious. The costs are also much greater for these two direct coupled arrangements compared to the baseline design.

h. Bearing and Bearing Mount - Elastic Deflection - The bearing deflection data given in Figure III-50 were obtained from the bearing manufacturers. In the case of the tapered roller bearing concepts, the manufacturer ran extensive computer programs to optimize the selection of bearing size, bearing center distance, deflection, and cost. This resulted in the back-to-back arrangement of concept C. In comparison, it will be observed that the considerable structural deflection of concept C has been eliminated.

It is to be noted that azimuth bearing deflections affect elevation tracking accuracy, but the reverse does not hold.

8. Control System Analyses

a. Fine Tracking System Stability Analysis - A computer program has been developed to study the stability of the heliostat fine-tracking control system. The simulation model is a simplified one; the primary objective was to study the effect of cross-coupling on the fine tracking control system.

Study results show that the fine-tracking control system operates very satisfactorily for all heliostats during the middle part of the day and for the heliostats in the center and back part of the field throughout the day. However, modification to the control system is required for heliostats in the front corners

		A.	B.	C.
Bearing Arrangement
		arc-min	arc-min	arc-min
6.70 m/s (15 mph)	Bearing Deflection Only	0.47	0.28	0.6
	Bearing & Structure Deflection	0.48	1.29	0.61
13.41 m/s (30 mph)	Bearing Deflection Only	13.0	0.66	1.6
	Bearing & Structure Deflection	13.4	4.71	1.7

Figure III-50 Deflections of Bearings and Associated Structure

Figure III-51 Fine-Tracking Simulation

of the field early in the morning and late in the afternoon. Heliostats in these positions are unstable or operate in a limit cycle--a continuous oscillation at a constant amplitude.

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 after 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. 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.

An alternative solution to the cross coupling instability problem would be to resolve the sensed errors into theoretically exact feedback signals to drive the azimuth and elevation gimbals. This could be done by using sine and cosine resolvers on the gimbals and analog computing elements or by using a digital microcomputer. The recommended solution of crossfeeding the fine-tracking signals is much less expensive to incorporate and should provide completely satisfactory operation.

- 1) Simulation Program - Figure III-51 is a block diagram of the simulation of the fine-tracking system. The program includes the equations that define the motion of the beam reflected from the heliostat as the sun and the heliostat gimbals move. The output of the fine-tracking sensor is assumed to be a linear function of the change in beam angle. The hysteresis function causes the gimbal drive motor to turn on when the pointing error of the beam exceeds 2.909×10^{-4} rad (1 arc minute) and to stay on until the pointing error is driven to zero. The motor is a constant-speed motor; the motor, gears, and gimbal inertia were simulated as an integrated and a first order lag. The time constant used for the lag was 0.1 sec.

In the simulation, the tracking motor drove the azimuth gimbal at 1.396 rad/hr (80 deg/hr), the elevation gimbal at 0.349 rad (20 deg/hr). The latest hardware configuration produces 1.5705 (90) and 0.5235 rad/hr (30 deg/hr); the simulation results will not be significantly changed.

- 2) Simulation Results - Figure III-52 shows fairly typical operation of the fine tracking control system. The time histories shown are for a heliostat near the center of the field, winter solstice, 9 a.m. The azimuth error builds up more rapidly than the elevation error as the sun moves. When the azimuth error gets to one arc minute, the azimuth motor turns on and drives the azimuth gimbal 1.455×10^{-4} rad (0.5 arc minute). This reduces the azimuth error to zero, and the motor turns off. This motion of the azimuth gimbal moves the reflected beam slightly in elevation and produces a small elevation error. The elevation error does not become large enough to turn on the elevation motor until about ten seconds after the start of the run. The elevation motor then turns on and drives the elevation error back to zero. This motion of the elevation gimbal produces an azimuth error that is large enough to turn the azimuth motor on in the negative direction. The azimuth motor then continues to pulse in the positive direction, driving the azimuth gimbal to keep the reflected beam aimed at the receiver as the sun moves.

a) Limit Cycling and Unstable Operation - Figure III-53 illustrates a limit cycling type of operation. This heliostat location is in the east front corner region of the field; the time is just after sunrise (just after 6 a.m.) at the equinox. The time histories show that azimuth error oscillates between $+5.818 \times 10^{-4}$ rad (+2 arc min) and -2.909×10^{-4} rad (-1 arc min), elevation error between +2 and -4 arc minutes. Both the azimuth gimbal and the elevation

Figure III-52 Typical Fine-Tracking System Operation

Figure III-53 Limit-Cycling Operation

gimbal track in the correct direction, but they oscillate around the desired angles. The gimbal angles are plotted in radians. In the zero azimuth position, the vector normal to the mirror surface points south; in the zero elevation position, the normal vector is straight up.

The type of operation shown in Figure III-53 is unacceptable because of the pointing errors and because the motors and gears are running almost continuously, producing unnecessary wear of gears and bearings and wasting electrical energy.

Figure III-54 illustrates unstable operation. The heliostat location is the east front corner region; the time is 6 a.m., summer solstice. The oscillations in elevation error and elevation gimbal angle are diverging.

b) Effect of Crossfeed - Figure III-55 shows the type of operation obtained using total crossfeed. The heliostat location and time are the same as for the unstable case shown in Figures III-54, but in this case, the sensed elevation error is used to drive the elevation gimbal. The plots show that this crossfeeding of the error signals produce satisfactory operation.

For heliostats in the front west corner of the field, crossfeed with opposite signs is required for late afternoon operation. That is, the sensed elevation error with sign reversed is used to drive the azimuth gimbal and the sensed azimuth error is used to drive the elevation gimbal.

c) Combination of Straight Feed and Crossfeed - There are cases for which the operation of the fine-tracking control system is marginal using either straight feed or total crossfeed. This is illustrated by Figures III-56 and III-57. There is no limit cycle in the time responses obtained from the simulation using straight feed of the error signals, as shown by Figure III-56. However, Figure III-56 shows that the oscillations in azimuth error periodically peak at -2.618×10^{-4} rad (0.9 arc min). A small amount of noise on the error signal would cause the azimuth error to exceed the -2.909×10^{-4} rad (-1.0 arc min) threshold, and a limit cycle would result. Since there will certainly be some noise on the error signal, the actual system represented by Figure III-56 would very likely operate in a limit cycling mode.

Figure III-54 Unstable Operation

Figure III-55 Total Crossfeed Example

Figure III-56 Marginal Operation Using Straight Feed

Figure III-57 Limit-Cycling Operation Using Total Crossfeed

Figures III-57 are time histories for the same heliostat at the same time of day and year using total crossfeed of the error signals. Total crossfeed in this case is worse than the straight feed case and results in a limit cycle.

The use of a combination of straight feed and crossfeed for this case results in satisfactory operation with error signals that vary only between zero and one arc minute in the desired turn-on direction, as shown in Figure III-58. Since the error signal excursions are much smaller than in the case using straight feed, the system will be much less susceptible to spurious oscillations or limit-cycling due to noise on the error signals.

The signal driving the azimuth gimbal is half of the sensed azimuth error plus half of the sensed elevation error. The signal driving the elevation gimbal is half of the sensed elevation error minus half of the sensed azimuth error.

The use of three modes of error signal feed--straight feed, total crossfeed, and a half-and-half combination--should provide satisfactory operation of all heliostats for all times of day and times of the year. The combination of straight feed and crossfeed will be used as a transition mode of operation in going from crossfeed to straight feed or from straight feed to crossfeed. The time at which switching occurs should not be at all critical; a tolerance of a few minutes should be adequate.

- 3) Portion of Collector Field Affected - Computer runs were made to determine approximately what part of the collector field has a stability or limit cycle problem when only straight feed of the error signals is used. Four different times were investigated--summer solstice at 6 a.m. and 9 a.m., equinox at 6 a.m., and winter solstice at 8 a.m. The results are shown on Figure III-59 through III-62. These figures show the west half of the collector field where the stability problem occurs in the late afternoon. The plots show the heliostats affected at 6 p.m., 3 p.m., 6 p.m., and 4 p.m. The mirror image heliostats in the east half of the field are affected at 6 a.m., 9 a.m., 6 a.m., and 8 a.m. These figures show that the stability problem is more severe in the summer than in the winter and more severe near sunrise and sunset as compared to midday.

Figure III-58 Combined Straight Feed/Crossfeed Example

Figure III-59 Portion of Collector Field Affected--Summer Solstice, 6 PM

Figure III-60 Portion of Collector Field Affected--Summer Solstice, 3 PM

Figure III-61 Portion of Collector Field Affected--Equinox, 6 PM

Figure III-62 Portion of Collector Field Affected--Winter Solstice, 4 PM

b. Determination of Gimbal Angles and Rates for Pilot Plant Heliostats - Figure III-63 through III-68 show azimuth and elevation gimbal angles, as a function of time of day for the baseline collector field. The gimbal angles are shown for a site located at a latitude of 35 degrees north for winter solstice, equinox, and summer solstice. For each time of year, the gimbal angles are plotted for the heliostats in the front and back corners of the field and in the center of the front and back rows.

The azimuth gimbal angle is measured from the south. The elevation gimbal angle is measured from a reference where the normal to the mirror surface is vertical.

The heliostats in the front corners of the collector field have the largest gimbal travel requirements. The largest gimbal travel occurs at summer solstice; the maximum azimuth gimbal angle at 6 a.m. and 6 p.m. is 1.65 radians. Using a maximum azimuth gimbal angle capability of 1.94 radians allows some margin for other site locations and for the possibility of plant operation before 6 a.m. or after 6 p.m.

The maximum azimuth rate requirement is 0.63 radians/hour and occurs at summer solstice for the same heliostats. For a plant located at 21 degrees north latitude (Hawaii), the maximum azimuth rate is 1.4 radians/hour.

The maximum travel of the elevation gimbal is 2.94 radians from the stow position (i.e., the vector normal to the mirror surface is 0.2 radians from vertical) for the .611 radian north latitude location. For the 21 degree north latitude location, the maximum travel is 3.05 radians from the stow position.

There is less variation in the peak rate of the elevation gimbal angle. The peak rate is less than .17 rad/hour for all locations.

9. Pointing and Alignment

The pointing and alignment process affects two primary areas of concern, the heliostat and mirror alignment and the fine tracking inline sensor alignment. During the initial phase of this contract, technique and feasibility studies and testing were accomplished using both collimated white light and an HeNe laser. The principal emphasis was placed on heliostat mirror pointing and alignment, with the evaluation also considering the application of the technique for the inline sensor alignment. The

Figure III-63
Gimbal Angles, Winter Solstice, Front Row

Figure III-64
Gimbal Angles, Winter Solstice, Back Row

Figure III-65 Gimbal Angles, Equinox, Front Row

Figure III-66 Gimbal Angles, Equinox, Back Row

Figure III-67 Gimbal Angles, Summer Solstice, Front Row

Figure III-68 Gimbal Angles, Summer Solstice, Back Row

various techniques had a single criteria for evaluation purposes, the ability to point and align all mirrors of a heliostat on a single target with an accuracy of $\pm 2.909 \times 10^{-4}$ rad (± 1 arc-min). The following paragraphs will discuss each technique evaluated.

The definition of an optimum technique for pointing, aligning, and focusing multiple mirror systems for solar power concentration is a critical problem for large solar power heliostat arrays. This problem is further complicated when adjustable-focus mirrors are used. In an effort to define an optimum technique, Martin Marietta evaluated several approaches to this problem. A brief discussion of the most promising methods, their advantages and disadvantages, and the rationale for rejecting each for this program is discussed.

a. Laser Techniques - The invention of lasers has provided a most powerful tool for the alignment of optical systems. Several laser techniques were evaluated during the study. The major disadvantage of using lasers in this application is that the narrow laser beams illuminate only a small portion of the total mirror surface. Therefore, the reflected beam represents only the optical properties of this small area. If prefocused well-figured mirrors were used, this surface sampling would be the most accurate. However, focusable mirrors of the type used during the collector subsystem experiment do not provide a consistent precision figure for which a single or even several small area elements will represent the correct focal point or total mirror performance. Therefore, both individual mirror focusing and intermirror alignment require that a large percentage of the focusable mirror surface be illuminated with collimated light or actual solar radiation. Such illumination provides both an optimum focusing and alignment image and the actual image produced by the full mirror.

- 1) Single Laser Approach - This approach, as shown in Figure III-69, uses a single laser positioned at the receiver aperture location (the point at which the solar beam will be focused by the mirror). The laser beam is then directed sequentially to several premasked points (5 minimum) on each heliostat mirror and the reflected beam displayed on a target. The target is placed at a simulated solar angle. When the mirror is properly focused, the laser images at the target should be spaced identically to the masked points on the mirror indicating parallel light. However, in addition to illuminating only a very small percentage of the mirror surface, this technique suffers from geometric insensitivity since the laser-to-mirror distance is much larger than the

Figure III-69 Single Laser Test Setup

Figure III-70 Laser Mobile Tower

mirror-to-target distance. Therefore, the relative image locations on the target must be precisely adjusted. Such adjustment is complicated because the laser beam is Gaussian and is diffused by this geometry. Locating this image with the precision required for pointing and alignment, therefore, becomes impractical. This is further complicated by the precise pointing requirement place on the laser.

- 2) Multiple-Laser Approach - This approach inverts the geometry of the single laser approach described above by positioning several (5 minimum) intercollimated lasers in front of each mirror so their angle of incidence simulates the equinox noon position of the sun. The mirror is then adjusted until the five images are superimposed at the receiver aperture. The prime reason for eliminating this technique was the irregularity of the focusable mirror figure discussed above. This leads to a high probability that the five-point or multiple-point focus will not be the true focal points of the entire mirror. This technique also requires the same precise collimation of the incident laser beam from one mirror to the next that is required for the recommended approach. The laser technique does, however, have a much stricter requirement for reflected beam interference, which eliminates positioning devices such as a bridge crane concept.

b. Collimated White Light Techniques - The results just discussed point out that the optimum light source for heliostat focusing was a collimated beam of light that would flood the heliostat. Since the heliostat is approximately 6.7 m (22 ft) square, a collimated light source of that size is impractical both to manufacture and position. Therefore, a collimated white light source that floods a single mirror and is sequentially positioned in front of each mirror so the collimation is preserved becomes an achievable solar simulator.

Although simulation of the solar noon equinox position of the sun is the most straightforward technical approach, accurate positioning of the light source is a difficult problem. This requires that the light source be located on a mobile positioning device that could be moved in front of each heliostat in the field. The light would then to move to 9 positions and retain collimation from position to position.

c. Selected Pointing and Alignment Technique - The selected technique to accomplish heliostat mirror alignment and pointing, after careful consideration of all of the above techniques appears to be a combination of best qualities of all the considered techniques. The following paragraphs will provide a description of and the rationale for both the heliostat mirrors and the fine-tracking assembly pointing and alignment.

- 1) Heliostat Mirror Pointing and Alignment - Several critical aspects have entered into the selection of technique to accomplish heliostat mirror alignment and pointing. The fact that the baseline mirror configuration has been selected as a pre-focused mirror, the single laser approach will yield more accuracy capability. However, during performance of this technique, it was noted that displacements representing 2.9×10^{-4} rad (± 1 arc-min) cannot be detected without providing an electronic detector which increases the alignment system cost. It was then decided that this problem could be relieved by relocating the laser source such that it would provide the solar simulation. This change allows the 2.9×10^{-4} rad (± 1 arc-min) to represent a much larger deflection at the receiver aperture which would be visually detectable on a reflective target.

The selected technique will be accomplished by providing a mobile tower assembly much like Figure III-70, with a single or multiple laser attached to simulate the selected solar angle. The laser beam will then be reflected on a target at the receiver aperture. The minimum deflection experienced at receiver would result from heliostats located nearest to the receiver. This minimum deflection would be approximately ± 43.9 mm (± 1.73 inches), which is easily detected visually.

- 2) Fine-Tracking Sensor Pointing and Alignment - The fine-tracking sensor pointing and alignment technique will utilize the same setup as the heliostat mirror pointing and alignment technique and will be accomplished in conjunction with alignment of the center mirror on the heliostat. The primary difference in the two techniques is that the error allocation associated with the fine-tracking sensor is $\pm 1.45 \times 10^{-4}$ rad (± 0.5 which at the sensor location is a deflection of $\pm .62$ mm ($\pm .024$ inches). Figure III-71 illustrates the technique for accomplishing this alignment. The technique will require that the center mirror be first aligned to the receiver using the selected solar angle, and then the fine-tracking sensor being aligned to that reflected laser beam. The $\pm 1.45 \times 10^{-4}$

Figure III-71 Fine-Tracking Sensor Pointing and Alignment

Figure III-72 Laser Pointing and Alignment Setup

rad (± 0.5 arc-min) deflection cannot be detected visually; therefore, a 4-element spot series position sensing detector integrated into the boresight tube of the fine-tracking sensor will be required. This device will be tuned to the HeNe laser wavelength and will have the capability of resolving the center of the laser beam to $\pm 2.54 \times 10^{-4}$ mm ($\pm 1.0 \times 10^{-5}$ inches) which is more than adequate for alignment to a maximum deflection of $\pm 6.2 \times 10^{-1}$ mm ($\pm .024$ inches).

- 3) Experiment Pointing and Alignment - The selected pointing and alignment techniques for both the heliostat mirrors and the fine-tracking sensor discussed above will be verified at site No. 4 of the collector subsystem experiment. Due to the severe terrain conditions existing at the subsystem experiment location, only site No. 4 provides compatible terrain conditions for verification of these techniques. This verification will not require fabrication of actual hardware visualized for the pilot plant. The concept verification will require only the setup of a scaffold in vicinity of site No. 4 to simulate the function of the mobile laser tower and a pointing target will be incorporated on the calorimeter protective closures. Figure III-72 illustrates the test setup that will be used during verification of the pointing and alignment techniques.

The remaining heliostats and fine-tracking sensors will use actual sun conditions to perform pointing and alignment procedures.

C. SUMMARY OF SUBSYSTEM RESEARCH EXPERIMENT (SRE)

1. Objectives

The primary objectives of the SRE to be performed at the MMC Solar Test Facility in Denver are to verify the calculated performance of the collector subsystem and its components, determine the efficiency with which solar energy can be concentrated into a cavity receiver and to prove the structural, mechanical and electrical design of the individual heliostats. Parallel efforts under MMC funding will evaluate the data bus method of executive control.

2. Experimental Approach

To attain the objectives described above, two types of heliostat will be tested. These two heliostats, which are variations of

the basic design for the pilot plant, represent a combination of the materials and processes available in early 1976 that will confirm the pilot plant design planned for 1977. The pilot plant will use a nine mirror heliostat with prefocused mirrors made of water white glass, giving a reflectivity of 91%. The nine mirror configuration requires mirrors 2.03 meters (6.7 feet) square; however, water white glass will not be available in this size for another year, although commercial plate glass is currently available in this size. Therefore, one heliostat of the nine mirror configuration will be built using commercial plate glass and the preformed honeycomb support structure. Water white glass is currently available in widths up to 1.2 meters (4 ft). The other three heliostats will be built with twenty-five mirrors each using the 1.2 meter water white laminated glass having the 91% reflectivity. Table III-15 lists the difference between the two heliostat types used for the experiment and contrasts these with the heliostat planned for the pilot plant. It may be noted that the nine mirror configuration provides a weight savings, materially reduces the number of adjustments, and consequently, saves material cost as well as setup and maintenance time.

The experiment will establish heliostat efficiency through the use of the three heliostats with 91% reflectivity glass and will provide structural and mechanical verification of the nine mirror configuration as experienced under the extremes of the Colorado environment.

3. Test Plan Summary

The methods by which the degree of success in attaining the objectives will be evaluated are contained in the following Test Plan summary.

This section describes the testing currently planned for the collector subsystem experiment. The test program will be comprised of four major increments, these being component, assembly, subsystem, and special tests. The component test program will primarily be directed toward performance verification of the components prior to installation at the assembly level. The assembly level testing is also an incremental verification of the major assemblies associated with the collector subsystem experiment which includes wiring and cabling continuity checkout, functional and control interface checkout, and pointing, alignment, and focusing technique verification. The subsystem level testing will be directed toward subsystem level performance verification, with primary emphasis placed on

Table III-15

Tabulation of Differences between the Two Proposed Experimental Heliostat Types and the Proposed Baseline Pilot Plant Heliostat Design

Characteristic	Experimental Type Heliostats		Baseline Pilot Plant Heliostat
	25-Mirror	9-Mirror	
Overall Heliostat			
Weight	2331 kg (5139 lb)	2306 kg (5084 lb)*	1991 kg (4390 lb)*
Number of Mirrors	25	9	9
Envelope Dimensions	6.9x6.7 m (22.5x22 ft)	6.7x6.9 m (22x22.75 ft)	6.7x6.9 m (22x22.75 ft)
Reflecting Area	37.16 m ² (400 ft ²)	37.16 m ² (400 ft ²)	37.16 m ² (400 ft ²)
Mirror Assembly & Supports			
Size	1.2x1.2 m (4x4 ft)	2.03x2.03 m (6.7x6.7 ft)	2.03x2.03 m (6.7x6.7 ft)
Focusing	Variable	Fixed (400 m or 1312 ft)	Fixed (148 to 693 m)
Substrate Material	Tubular Steel	Honeycomb	Honeycomb
Glass Thickness	6.3 mm (0.25 in.) 3 mm Face Sheet	6.3 mm (0.25 in.)	3 mm (0.118 in.)
Spectral Reflectivity	91%	73%	91%
Number of Support Struts/ Heliostat	10	6	6
Number of Pointing Adjustments	50	18	18
Number of Focusing Adjustments	125	None	None
Main Structure			
Yoke Width	4.1 m (13.5 ft)	2.4 m (7.8 ft)	2.4 m (7.8 ft)
Yoke Height	3.7 m (12.0 ft)	3.6 m (11.9 ft)	3.6 m (11.9 ft)
Yoke-Vertical Member	0.3 m (1 ft) I-Beam	0.25 m (.85 ft) I-Beam	0.25 (.85 ft) I-Beam
Yoke-Horizontal Member	Steel, 0.3x0.3 m (12x12 in.)	Steel, 0.3x0.3 m (12x12 in.)	Steel, 0.3x0.3 m (12x12 in.)
Drive Mechanism/ Bearings	Same	Same	Same
Foundation Interface	Same	Same	Same
Rotational Limits			
Azimuth	±110° Maximum	±110° Maximum	±110° Maximum
Elevation	±180° Maximum	±180° Maximum	±180° Maximum
*Difference in weight due to use of thinner glass in baseline pilot plant design.			

collector (heliostat) efficiency and tracking accuracy. The special test program is any test(s) which are not necessarily an intrinsic part of the normal subsystem buildup. These tests will include both subscale and assembly level testing such as, but not limited to, the following:

- 1) Mirror specular reflectivity testing related to exposure to environmental conditions.
- 2) Fine-tracking unit cross-over tracking due to special siting locations.
- 3) Environmental monitoring and data collection for heliostat and tower long term exposures.

a. Component Testing - The collector subsystem experiment component testing will be limited to functional performance type of testing. This testing is to provide assurance that the subject component is capable of performing all defined design functional parameters within the limits specified. During this phase of testing, the components will be periodically inspected and tested during performance of the special environmental monitoring and data collection activities.

This testing will be accomplished either by the component supplier or Martin Marietta, depending on the extent of supplier testing accomplished. As a minimum, MMC will test all components to assure functional operations prior to installation of the component in its applicable assembly.

b. Assembly Level Test - The assembly level testing will be directed toward three major assemblies: site cabling and wiring; the fine tracking assembly; and the collector (heliostat). This test verification will be accomplished in increments which follow the logical buildup of the collector subsystem experiment hardware. This testing will include verification of ancillary functions such as pointing, alignment, and focusing. The various assembly level test, by necessity, will have to be combined in order to establish complete functional verification. The following paragraphs, in general, will define the types of tests associated with each major assembly.

- 1) Cabling and Wiring Assembly Level Test - The cabling and wiring assembly level test will be primarily continuity, insulation resistance, and grounding testing. This testing will be accomplished in accordance with the subsystem experiment cabling and wiring diagrams and engineering standard procedures and techniques. These tests will provide assurance that all interface cabling is properly terminated for the control, power, and grounding systems. In addition, this same cabling will be used during performance verification of the heliostat and fine tracking assembly testing to assure subsystem interface compatibility.
- 2) Collector (Heliostat) Assembly Level Test - The heliostat assembly level testing will be accomplished in increments beginning with the installation of the heliostat yoke mechanism and its drive assemblies. After completion of this installation, all functional parameters associated directly with this mechanism and its drive assembly will be tested in all modes of operation prior to progressing to the next level of assembly. These tests will include, to the extent possible, verification of fast slew and fine tracking rates, drive to associated limits, error allocations, and other parameters deemed pertinent. After completion of this testing and assurance of proper operation is established, the next level of assembly will be accomplished. This method of alternating assembly and testing will continue until the entire heliostat assembly is completed. The testing accomplished during this phase will also provide assurance that all interface requirements existing between manual and executive control are properly implemented. The final increment of testing associated with heliostat assembly will be performance of pointing, alignment and focusing operations, and calorimeter assessments of heliostat efficiency.
- 3) Fine-Tracking Assembly Level Test - This assembly level testing will be primarily directed toward assessment of the collector subsystem experiment tracking accuracy. The testing activities will be accomplished in the same manner as the heliostat assembly level verification in that the necessary testing will be accomplished in a logical build-up order such that at completion of the tower/sensor build-up, all pertinent functional parameters will have been verified for proper operations. Again, the final activity will be pointing and alignment of the fine-tracking sensor to the receiver (calorimeter) and the heliostat with subsequent verification of tracking accuracy.

c. Subsystem Level Test - The subsystem level of testing is the most important phase of testing associated with the collector subsystem experiment program. This phase of testing will demonstrate the collector subsystem's capability to deliver the proper solar thermal energy to the receiver and performance of the subsystem in all operational modes.

The testing accomplished during this phase will be primarily directed toward collector efficiency and tracking accuracy of individual and groups of heliostats. In addition, empirical performance data will be collected during critical periods of the solar year, such as summer solstice, equinox, and winter solstice. Further, operation of the experiment as a subsystem will provide the necessary data to establish preliminary design requirements, support equipment requirements, operational and emergency procedures.

d. Special Engineering Tests - The special engineering tests currently planned for this phase are designed to evaluate such things as degradation of mirror surfaces as an effect of environmental exposures, effects of wind and thermal loading on heliostat and fine-tracking assembly structures, and effects of critical solar angles (early morning and late afternoon) on the fine-tracking assembly. The following paragraphs provide a general description of this testing activity.

- 1) Specular Reflectivity Test - This test will be accomplished by using a collimated specular reflectivity measuring technique as illustrated in Figures III-28 and III-29 after varying environmental conditions which assess the effects of contamination, long term exposures to sand and dust, and moisture on the specular reflectivity of the heliostat mirrors. This test will be accomplished on coupon samples taken from mirror materials to be used on the collector subsystem experiment. These tests will be accomplished at either the MMC Denver facilities or the ERDA/Sandia Solar Test Facility at Albuquerque, New Mexico.
- 2) Environmental Monitoring and Data Collection - This series of tests will be accomplished at the MMC solar test facilities and will consist of exposing a nine mirror configuration heliostat and a fine-tracking assembly to the natural environmental conditions existing at this location. The location of this test setup was selected due to the normally severe wind conditions existing at this location. The heliostat and the fine-tracking assembly will be fully instrumented

with strain gages, accelerometers, and thermocouples for purposes of measuring thermal loading (temperature gradients) and dynamic loading. The primary concern will be directed toward structural deflections due to thermal and wind loading. The data collection associated with this testing will be accomplished on a continuous basis for the duration of the testing. In conjunction with the collection of assembly environmental data, weather data for this location will also be collected on the same continuous basis to provide direct correlation between the natural environment conditions and the assembly response data.

- 3) Fine-Tracking Assembly Axis Cross-over Test - The primary objective of this test will be to demonstrate that the executive control system can command cross-over error signals from the fine tracking sensor that will result in acceptable beam positioning accuracy for certain heliostats during critical solar angles occurring during early morning and late afternoon hours.

e. Test Documentation and Controls - The collector subsystem experiment test program will be controlled by a subsystem integrated test plan which establishes the test objectives and requirements for all testing from the component through subsystem level of testing. Further, all tests will be controlled and conducted by use of a written test procedure.

Test controls such as instrument and test tolerance, as well as test methods, will be controlled by MMC engineering standard procedures.

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 reflected from a north side heliostat field. A description of the cavity configuration is presented in Section IV-A-2 of this report.

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 three major elements of a natural circulation boiler are: 1) the boiler tubes which generate saturated steam from feedwater; 2) the steam drum which separates the steam/water mixture from the boiler tubes; and 3) the superheater which further increases the temperature and energy content of the steam to the specified exit condition. The three major elements are linked together by a system of feeders, risers, headers, and downcomers, which provide the necessary internal fluid flow paths.

The main reason the natural circulation type of boiler was selected was because it is a well proven, highly reliable concept which has been successfully used in many different applications over a period of many years. Furthermore, it is deemed to be entirely compatible with the Central Receiver Solar Thermal Power System operating requirements. The physical and operational characteristics of the pilot plant boiler are discussed in detail in Section V-A.

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.

Table IV-1 Receiver Requirements

Aperture Opening	7.498 m (24.6 ft)		
Tilt of Receiver Centerline	0.349 ± 0.087 rad (20 ± 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 (960°F)		
Pressure	9136 kPa (1,325 psig)		
Minimum Receiver Efficiency	<u>m/s</u>	<u>Wind Speed (mph)</u>	<u>Efficiency</u>
Heat into Steam	0	0	95
Heat into Cavity	4.47	10	94
	6.71	15	93
Cycle Life			
Diurnal Cycles	13,000		
Cold Starts	1,300		
Startup Time			
Over Night	20 min		
Cold	6 hr		
Winds	33.5 m/s (75 mph) Max Operating		
	100.6 m/s (225 mph) Max		

2. Cavity Configuration

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 factor which influenced the size, shape, and orientation of the cavity was the heliostat field layout. Other important considerations were cavity efficiency and heat flux profiles.

Figures IV-1 and IV-2 depict the selected cavity configuration. Dimensions shown are for inside surfaces at ambient temperature. Also shown in Figures IV-1 and IV-2 are the projections of heliostat solar beams which bound the directly insolated surfaces of the cavity. These figures vividly illustrate that active steam generating tubing must occupy most of the two 6.706 m (22 ft) wide side walls 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.

The depth and width of the cavity, and the orientation of the 3 active walls with respect to the aperture, were established to attain a reasonable compromise between receiver size and active surface heat flux distribution. A larger cavity could result in lower peak heat fluxes and a somewhat more uniform distribution of energy over the active walls, but would also be heavier (a detriment to tower design) and less efficient due to increased heat losses through the insulated walls.

The front face of the cavity was extended beyond the dimensions of the aperture to a size of 9.144 m (30 ft) square. This was done to permit a slight enlargement of the aperture even after fabrication is well under way should the need arise due to unforeseen circumstances such as changes in heliostat design or field layout might dictate.

Figure IV-1 Martin Marietta Pilot Plant Receiver, Top View

Figure IV-2 Martin Marietta Pilot Plant Receiver, Side View

3. Solar Heat Flux Analysis

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 complexity and quantity of calculations to be performed in an accurate heat flux distribution analysis dictates the use of a computerized math model. The MMC-developed Thermal Radiation Analysis System (TRASYS) was used as the basis for the model. However, at the outset of this contract, TRASYS lacked certain capabilities necessary to simulate heliostat tracking and focusing characteristics. Since the ongoing conceptual design effort required some knowledge of heat flux profiles, a task to generate preliminary estimates by means of simplified hand calculations was initiated in parallel with the generation of the needed additions to the TRASYS program.

a. Preliminary Flux Profile Estimate - The hand calculated heat flux analysis was geared to fulfill two prime objectives: 1) to insure that the total pattern of direct solar energy encompassed the three cavity interior walls designated as active steam generation surfaces; and 2) to get a preliminary idea of the magnitudes and variations in heat flux at various locations on those three surfaces. It should be noted that early in the program when this analysis was performed, the MMC baseline heliostat field was the terraced concept.

The analysis was performed for a solar noon time of day to take advantage of the symmetry of the heliostat field and cavity. Therefore, only the east half of the heliostat field and the west half of the cavity was analyzed. A noontime insolation value of 1.02 kW/m^2 was assumed. It was also assumed that each heliostat was aimed and focused upon the geometric center of the aperture. The combined effects of spherical aberrations, aiming inaccuracies, focal length tolerances, and mirror contour imperfections were simulated by using a slightly larger solar disc (included angle of 42 minutes rather than 32 minutes) and then adding 1.219 m (4 ft) to the calculated diameter of each projected image.

The centerline of an idealized circular beam from each heliostat was projected upon the plane of the cavity interior surface by solving the simple trigonometric relationship between heliostat, aperture, and surface. Then the effective beam diameters were calculated using the previously mentioned correction factors. This resulted in an estimate of the average heat flux and area of influence of the reflected beam from each heliostat. By totaling the flux contributions of each image impinging upon a given point within the cavity, the heat flux at that point was determined.

The results of this analysis are illustrated in Figure IV-3. Two very important facts were derived from this study. First, there were very steep heat flux gradients imposed upon the three active cavity surfaces. This knowledge flagged the criticality of a careful design and stress analysis of superheater tubes which might be subjected to much higher heat fluxes than would normally be encountered in a conventional fired boiler. Secondly, these preliminary results demonstrated that the complete pattern of reflected energy significantly exceeded the originally proposed rear wall height of 13.66 m (44 ft, 10 in.), so that an increase to 15.24 m (50 ft) was necessary.

b. Thermal Radiation Analysis System (TRASYS) Math Model - TRASYS is a generalized thermal radiation computer program which has been developed and used at MMC for several years. TRASYS, and its predecessor MTRAP, was used with great success in solving radiation problems for the Skylab, Titan III, and Viking space projects. The two major outputs of the program are: 1) radiation interchange factors (also called gray-body view factors) for a given system of surfaces; and 2) energy incident upon each surface by direct and reflected radiation from specified heat sources.

When TRASYS was designated for use in calculating heat flux patterns within the Central Receiver Solar Thermal Power System receiving cavity, it lacked the capability for calculating heliostat aiming and tracking parameters, and also, the logic to simulate focusing of the radiant energy from the curved mirror surfaces. These capabilities were added during the first three months of the CRSTPS contract. TRASYS locates the sun at any given time of day and day of year, orients each heliostat in azimuth and elevation to project the focused solar image toward the specified aiming point (i.e., center of cavity aperture), and then determines the resulting heat flux distribution upon the receiver surfaces. Receiver and heliostat mirror surfaces can be divided into any desired number of finite sized elements. If, in the interest of run time savings, it is desired to use only a single

Figure IV-3
Preliminary Heat Flux Profile Estimated for Maximum Flux Case

surface element for each heliostat, the effect of spherical aberration upon projected solar image size can be approximated by a programmed factor based upon mirror size and angle of incidence.

c. Pilot Plant Receiver Heat Flux Distribution - 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) 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 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.

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		0.982	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		0.986	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	0.85	0.992	0.805	52.55
(System Design Case)	1400	0.826	0.927	0.85	0.991	0.780	41.09

Direct solar heat flux incident upon the three active steam generation surfaces of the receiver for a minimum flux case, maximum flux case, and a more nominal flux case are shown in Figures IV-4, IV-5, and IV-6.

It is noted in Figure IV-5 that the peak maximum heat flux occurs just below the horizontal centerline on the cavity rear surface. For this case, the vertical heat flux gradients are quite sharp, while the horizontal gradients are much less pronounced.

The minimum heat flux case (Figure IV-4) also produces the most asymmetrical flux patterns within the receiver, as can be seen by comparing fluxes on corresponding surface elements on either side of the vertical centerline.

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 (section V-A-4) is then sized to accommodate the maximum anticipated proportion of energy on the superheaters. Figure IV-7 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%.

4. Cavity Thermal Analysis

There are two types of receiver thermal analysis discussed in this report. First is an overall thermal balance of the receiver for the purpose of predicting effective interior surface temperatures and determination of the various cavity heat losses. This analysis combines a very detailed simulation of cavity inter-surface and boundary heat transfer characteristics with very simplified calculations of tube interior thermodynamics. Secondly, a more detailed heat transfer and thermodynamic analysis of receiver

Aperture Efficiency: 0.9657
 Energy into Receiver: 9.9 MW
 Flux Values in kW/m² (Btu/hr-ft²)

Day: 172
 Time: 0600

Insolation: 0.304 kW/m²
 (96.5 Btu/hr-ft²)
 Latitude: 33.6°N
 Field Average Cosine: 0.6237

Figure IV-4 Pilot Plant Receiver Heat Flux Profile

Insolation: 1.023 kW/m²
(324.2 Btu/hr-ft²)
Latitude: 33.6°N
Field Average Cosine: 0.9543

Day: 355
Time: 1200

Aperture Efficiency: 0.9920
Energy into Receiver: 52.55 MW
Flux Values in kW/m² (Btu/hr ft²)

Figure IV-5 Pilot Plant Receiver Heat Flux Profile

Insolation: 0.919 kW/m^2
 $(291.7 \text{ Btu/hr-ft}^2)$
 Latitude: 33.6°N
 Field Average Cosine: 0.9049

Day: 80
 Time: 1000

Aperture Efficiency: 0.9880
 Energy into Receiver: 44.6 MW
 Flux Values in kW/m^2 (Btu/hr-ft^2)

Figure IV-6 Pilot Plant Receiver Heat Flux Profile

Figure IV-7 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	9.156	29.75	30.8
	1000	2.368	2.736	2.453	2.586	2.205	1.036	13.334	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
Winter Solstice (355)	1200	2.218	2.650	2.218	2.484	2.484	1.031	13.085	45.28	28.9
	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

tubes and contained fluid is performed, using simplified boundary conditions for external tube surfaces. The former type of analysis is described in this section. The later analysis is discussed in Section V-A-2.

The overall receiver thermal balance is performed primarily with the Martin Improved Thermal Analysis System (MITAS) computer program. MITAS is a very versatile, general purpose thermal analysis program developed by Martin Marietta Corporation. Like TRASYS, MITAS has been used for highly accurate thermal analysis of spaceborne systems over the past several years. MITAS uses the heat flux and radiation interchange output data from TRASYS together with receiver lumped parameter nodal data, conductance and convective characteristics to calculate heat transfer rates and temperatures for the various parts of the receiver.

The present MITAS model of the receiver is capable of steady state analysis only. The determination of necessary heat capacity properties must be made in order to extend the model to the simulation of receiver transient operation. In its present state of development, the MITAS model calculates receiver cavity surface temperatures, energy input to boiler fluid, energy input to superheated steam, steam temperatures throughout superheater panels, and aperture energy losses by convection and radiation. It should be noted that the aperture convective loss logic is a very simplified approximation based upon limited small scale model free convective tests performed at MMC last year. The true convective characteristics of the cavity receiver are very complex, particularly for the more realistic case with wind effects, and have yet to be investigated in sufficient detail to formulate meaningful methods of prediction.

The MITAS model has been run for the maximum pilot plant receiver heat input case (noon on the winter solstice, insolation of 1.02 kW/m^2). A summary of predicted interior surface temperatures is presented in Table IV-4. More detailed temperature distribution data are presented in Figure IV-8. Nominal cavity efficiency (ratio of energy into steam to energy entering cavity) is 93% at the pilot plant system design point, at a wind velocity of 6.7 m/s (15 mph). Radiation loss back through the aperture is less than 3%, convection losses total 3.5%, and loss through cavity wall insulation is 0.5%.

B. 5 MW RESEARCH EXPERIMENT RECEIVER

A reduced scale model research experiment receiver is to be designed, fabricated, and tested for the purpose of verification

Table IV-4 Preliminary Pilot Plant Receiver Temperature Summary

CONDITIONS

Maximum Incident Heat Flux Case

Steady State Operation - Steam Output of 69,400 kg/h (153,000 lbm/hr)

Interior Surface Radiation Properties: Active $\alpha_s = 0.9$

$$\epsilon = 0.8$$

Uncooled $\alpha_s = 0.3$

$$\epsilon = 0.8$$

SUPERHEATER TEMPERATURE GAINS

Pass:	1	2	3	4	5	6
ΔT (K)	281	288	296.5	298	299	280
ΔT (°F)	46	58	74	77	78	45

STEAM OUTLET TEMPERATURE

Calculated: 801.5 K (983°F)

Required: 789 K (960°F)

UNCOOLED SURFACE TEMPERATURES

Insulated Side Walls: 589 - 1089 K (600 - 1500°F)

Remaining Sides, Top and Bottom: 628 - 678 K (670 - 1760°F)

(a) Sidewall and Rear Active Surfaces

(b) Top and Bottom Surfaces

(c) Uncolled Sidewalls

Figure IV-8 Maximum Predicted Internal Surface Temperatures, K (°F)

Figure IV-8 Maximum Predicted Internal Surface Temperatures, K (°F)

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.

Although our present Central Receiver Solar Thermal Power System contract defines no requirements for operation of the research experiment receiver in an actual solar collection field, we recognize that the Energy Research and Development Administration does plan to subject this receiver to such operation in the forthcoming 5 MW Solar Thermal Test Facility. We are, consequently, orienting the design to be compatible with such operation insofar as cost and schedule limitations permit.

1. Cavity Scaling Criteria

It is important that the research receiver cavity be a geometric scale model of the pilot plant design for several reasons. First, the predicted heat flux intensities and gradients are somewhat unique as compared to normal boiler conditions, and a geometrically similar cavity facilitates duplication of these heat flux patterns. Also, the attitude of the boiler tubes, headers, downcomers, and steam drum, with respect to the vertical, is an important parameter. Lastly, the attainment of meaningful operational data from testing in an actual heliostat field is very dependent upon the similarity of cavity geometry.

The next question to be answered then was what scale factor should be used. Since one of the most important parameters to be simulated is energy per unit area of receiver cavity surface, it seemed logical to scale the research receiver cavity according to the relationship

$$L_{\text{Res. Rec.}} = L_{\text{p.p.}} \sqrt{\frac{\text{Energy into Research Receiver}}{\text{Energy into Pilot Plant Receiver}}}$$

The total energy available to the research receiver (numerator in the above equation) was established at 5 MW by power supply limitations at the designated test facility. The denominator in the above equation was chosen to be energy into the receiver at the pilot plant system design point for two reasons. First, the system design point naturally is the focal point for most performance analyses. Secondly, but perhaps more importantly, basing the scale factor upon the system design point results in a larger cavity and aperture than would result from referencing to the maximum heat flux point. This is important in considering the operation of the receiver in an actual solar thermal test facility, wherein more of the heliostats are further from the aperture than the square root scaling law would indicate. The reason is because the tower height, distance from tower to the first row of heliostats, and the spacing between heliostats all are greater distances than the square root law would establish in order to maintain minimum blocking and shadowing among heliostats. This results in proportionally larger solar images projected at the aperture plane, and a lesser percentage of the energy entering the cavity to generate steam. The described effect is illustrated in Table IV-5, which compares aperture efficiencies of the pilot plant receiver and research experiment receiver. The research experiment receiver data were determined from a TRASYS analysis of an MMC proposed 5 MW solar thermal test facility. Although the field average cosines are very comparable for the two receivers, the research experiment aperture

Table IV-5

Comparison of Receiver Solar Collection Characteristics for Pilot Plant and Research Experiment

Day of Year	Solar Time (Hour)	Field Average Cosine		Aperture Efficiency	
		Pilot Plant	Research Exp	Pilot Plant	Research Exp
Spring Equinox	1000	0.905	0.914	0.988	0.955
	1200	0.942	0.944	0.990	0.966
Summer Solstice	1000	0.844	0.844	0.983	0.924
	1200	0.874	0.873	0.985	0.940
Winter Solstice	1000	0.927	0.948	0.991	0.965
	1200	0.954	0.975	0.992	0.970

efficiency is significantly lower than that of the pilot plant in each case. If the research receiver scale factor had been based upon maximum energy into the pilot plant receiver, a smaller aperture would have resulted. The aperture efficiencies would, therefore, be even lower than for the selected design, and the attendant high heat fluxes impinging upon receiver external surfaces would pose a formidable thermal protection problem.

Final sizing of the research experiment receiver cavity, then, was established by the relationship

$$L_{RE} = L_{PP} \sqrt{\frac{5 \text{ MW}}{39.6 \text{ MW}}} = 0.3553 L_{PP}$$

where L_{RE} is a research experiment dimension which corresponds to the dimension L_{PP} in the pilot plant receiver.

2. Preliminary Test System Concept

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 graphite resistance elements will provide the radiant heat source. The radiant heater arrays will be 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 will be 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-9. Superheated steam exiting the receiver flows through a heat exchanger which reduces steam temperatures to very near the saturation point, while heating the boiler feed-water 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

Figure IV-9 .5 Mwt Research Receiver Test System Schematic

pressure and temperature, and is again converted to superheated steam in the receiver.

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.

SECTION V

RECEIVER SUBSYSTEM DESIGN

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7. RECEIVER SUBSYSTEM DESIGN

A. 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 Figures V-1 and V-2, respectively. 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. This construction was adopted because it is structurally rigid, may be handled in shipment and during erection with relative ease, and is impervious to solar flux.

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.

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

Figure V-1 Side Elevation View - Pilot Plant Receiver

Figure V-2 Plan View - Pilot Plant Receiver

platens. These tube platens are placed in front (on the cavity side) of the waterwall panels, as shown in Figure V-3. All superheater headers and transfer lines between passes are located behind the waterwall panels. Superheater 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. These openings are made by offsetting a suitable number of adjacent waterwall tubes outward into the cavity a sufficient distance to provide clearance for the superheater tubes. The offset waterwall tubes serve to shield the openings from the incident solar flux. The layout of superheater tubes and wall openings is shown in a subsequent section of this report in Figure V-12.

The inner surfaces of the cavity not lined with boiler tubes are lined with either steel coated with reflective material or refractory, depending on the intensity of incident solar flux on the particular surface. The outside surface of the entire cavity, as well as the steam drum and exterior piping, is encased in weatherproofed insulation.

The aperture of the cavity is to be provided with an insulated door (not shown in the accompanying figures) which can be closed to minimize heat loss and resultant cooling down when the receiver is out of service.

The receiver cavity and pressure parts are suspended from a structural steel framework and are free to expand laterally and downward. The suspended assembly is suitably tied back to the structure to effect the desired 20-degree tilt. Bumpers located on the cavity casing transmit horizontal loads caused by wind or earthquake to the supporting structure.

Essential instrumentation, valving, and control features of the receiver are shown schematically in Figure V-4. 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. Attenuation is required during start-up not only to control steam temperature but also to prevent excessively high tube temperatures. During normal operation it is required because the fraction of the total solar flux incident on the superheater varies with the time of day and season of the year. The attemperator between the fourth and fifth passes is used during start-up only.

Figure V-3 Arrangement Of Superheater Passes - Pilot Plant Receiver

SYMBOLS

Controls are provided to de-focus the heliostats immediately in the event of high superheater outlet steam temperature, high steam drum pressure, or low water level in the steam drum. The receiver is provided with one superheater outlet safety valve and two drum safety valves as required by the ASME Code.

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. A detailed discussion is not intended here. Rather, only their names and sources are identified. For heat transfer calculation, Dittus-Boelter's correlation^[1]* was used in the sub-cooled water region, Thom's^[2] in the sub-cooled boiler region, Chen's^[3] in the nucleate boiling region, and Heineman's^[4] in the superheated steam region. Correlations for predicting departure-from-nucleate boiling (DNB) are lacking for a tube heated from one side, in the pressure range considered in the pilot receiver design. Therefore, design equations developed by General Electric Company^[5] for uniformly heated tubes were used in estimating DNB heat flux at the pilot receiver design conditions. Ample margin was then included in designing the boiler circuitry. For pressure drop calculation, a conventional method by Moody^[6] was employed for a single-phase flow region and a Martinelli-Nelson correlation^[7] for a two-phase flow region.

The pilot plant receiver operating and design conditions are summarized in Table V-1. The maximum insolation condition corresponds to the heat input to the receiver at 1200 on winter solstice at an insolation of 1.023 kW/m^2 , and the rated condition (also termed the system design point) corresponds to that at 1400 on the same day at an insolation of 0.826 kW/m^2 . The preliminary pilot plant receiver design was based on the maximum insolation condition. The heat flux distributions on the active cavity walls at this condition are shown in Figure V-5. The locations represented by these plots were numbered and depicted in the insert in this figure. Design pressure was determined by adding 6 percent to the maximum drum operating pressure to account for the drum safety valve setting and then rounding to a larger value.

The physical dimensions of the pilot plant receiver were built up around the cavity internal dimensions shown in Section IV. 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, enclosure and insulation were then added to box up the receiver, and finally, a steel structure was designed to support the complete receiver assembly.

b. Parametric Studies The anticipated cyclic and asymmetric heating of the boiler and superheater imposes severe stresses on the

*References are listed in Section V.C

Table V-1 Pilot Receiver Operating And Design Conditions

	MAXIMUM INSOLATION CONDITION	RATED INSOLATION CONDITION
INSOLATION, kW/m ²	1.023	0.826
FLOW RATE, kg/h (lb/h)	69,400 (153,000)	51,962 (114,556)
SUPERHEATER OUTLET CONDITIONS		
Pressure, kPa (psig)	9,136 (1,325)	9,136 (1,325)
Temperature, K (°F)	789 (960)	789 (960)
APPROX. ΔP THRU SUPERHEATER,		
kPa (psi)	1,724 (250)	1,034 (150)
DRUM OPERATING CONDITIONS		
Pressure, kPa (psig)	10,860 (1,575)	10,170 (1,475)
Saturation Temp., K (°F)	590 (603)	586 (595)
FEEDWATER TEMP., K (°F)	479.5 (403.5)	479.5 (403.5)
TOTAL THERMAL DUTY, MW	49.05	36.7
SUPERHEATER DUTY, MW	13.78	10.11
SUPERHEATER DUTY/TOTAL DUTY, %	28.09	27.56

Design Pressure: 10,860 x 1.06 = 11,512 kPa. Use 11,722 kPa
(1,575 x 1.06 = 1,670 psi. Use 1,700 psi)

Design Temperature: (1) Superheater tubes at maximum mean wall temperature of 894 K (1150°F)
(2) Other pressure parts at their respective maximum expected temperature.

(Maximum Insolation Condition)

1.0226 kW/m^2

Day: 355 (Winter Solstice)

Time: 12 00 (Solar Time)

Figure V-5 Pilot Plant Receiver Heat Flux Profiles At The Maximum Insolation Condition

tubes (particularly the superheater tubes) which have to be accommodated in the receiver design. Therefore, parametric studies were performed first to determine the effect of heat flux intensity and its asymmetric distribution on various sizes of tubes at various fluid conditions in the tubes. The parameters considered were tube material, tube size, heat flux intensity, bulk fluid temperature, and fluid heat transfer coefficient (which is related to fluid mass velocity). Table V-2 summarizes these parameters and their respective ranges analyzed for both superheater and boiler tubes.

Thermal stress calculations were performed using the computer program ANSYS. In conjunction with this program, two-dimensional finite element models for superheater and boiler tubes as shown in Figure V-6 were set up. Because heat input is axisymmetric, only one half of a tube was modeled. A cosine heat flux distribution as shown in the figure was assumed for the tube surface receiving heat. In the case of the boiler tube, the front surface of the welded fin received the full incident flux intensity. With input comprising a certain combination of thermal boundary conditions, tube material, and tube size, the program calculated the temperature distribution of the tube. Assuming a plane strain boundary condition, the program also calculated thermal stresses. These stress values were then corrected for the actual tube support conditions. The detailed method of stress calculation and the results of parametric studies are presented in Section A.3, Structural Analysis and Design.

c. Superheater/Boiler 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. As shown in Table V-1, the superheater requires 28 percent of the total absorbed energy and the boiler, 72 percent. Since the superheater tubes contain steam at relatively high temperature and the inside heat transfer coefficient is relatively low, the temperature of the tubes tends to be high and very sensitive to the heat flux. Therefore, the arrangement of the superheater is rather critical. Because of the uneven heat flux distribution on the heating surfaces, as depicted in Figure V-5, surface arrangement became very difficult, 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.

Table V-2 Summary Of The Parameters And Their Ranges
Analyzed In Parametric Studies

SUPERHEATER TUBE

- Tube Material:	SA-213 TP304H; SA-213 TP316H SA-213 T22; SB-163-800 GR 2
- Tube O. D.:	19.05; 25.4; 31.75 mm (0.75; 1.0; 1.25 in)
- Heat Flux:	From 94,638 to 520,510 W/m ² (From 30,000 to 165,000 Btu/h-ft ²)
- Bulk Fluid Temperature:	From 803 to 672 K (From 985 to 750°F)
- Film Coefficient:	From 2,612 to 5,394 W/m ² -K (From 460 to 950 Btu/h-°F-ft ²)

BOILER TUBE

- Tube Material:	SA-210 A1
- Tube O. D.:	38.1 mm (1.5 in)
- Heat Flux:	441,643 and 694,010 W/m ² (140,000 and 220,000 Btu/h-ft ²)
- Bulk Fluid Temperature:	583 K (590°F)
- Film Coefficient:	5,678; 17,035; 34,070 W/m ² -K (1,000; 3,000; 6,000 Btu/h-°F-ft ²)

(a) SUPERHEATER TUBE

(b) BOILER TUBE

Figure V-6 Finite Element Models Used In Parametric Study

To assist in arranging the superheater passes, the heat energy was integrated over the rear and side walls and plotted against the height of each wall as shown in Figure V-7. Different schemes of superheater surface arrangement were evaluated and are shown in Figure V-8, where the superheater passes are numbered in the order in which steam flows through them. Dotted lines with arrows indicate the connections between the passes and flow directions. Schemes 1, 2, and 3 were based on preliminary flux distribution curves developed by Martin Marietta early in the program which resembled the updated curves shown in Figure V-5 except that peak flux values were smaller and the elevations of the flux peaks were lower. Scheme 1, with all the superheater passes located on one area of the rear wall, presented a simple and neat arrangement, but it was abandoned because thermal stresses in passes 1 and 2, which traverse the peak flux zone, were excessive. In Scheme 2, although two superheater passes were shifted lower to avoid the peak flux zone, portions of passes 1 and 2 were still exposed to flux intensities high enough to cause unacceptable thermal stresses. Further investigation of the lower heat flux zone at the bottom of the rear wall proved that the energy there was not sufficient to meet the superheater energy requirement. Scheme 3, which has the superheater surfaces located outside high flux zones of both rear and side walls, would be the selected scheme if the initial flux distribution were not updated. As shown in Figure V-5, the peak heat flux elevation in the updated flux distribution shifted upward to approximately 45 percent of the wall height from the bottom. To accommodate this change, spaces between passes 1 and 5 and between passes 3 and 4 were provided in Scheme 4 to keep superheater passes away from the peak flux region. Further evaluations revealed that the heat flux at the lower portions of the upper side wall passes (4 and 5 as shown in Scheme 4) were too high to keep the tube temperatures in these areas within material design limits. Consequently, flow path arrangement was modified in the final scheme, which is shown in Figure V-3. In this final selected scheme, each side wall pass was divided into two sub-passes, and steam flow was split into one upper and one lower sub-pass 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.

d. Design Analyses of the Preliminary Baseline Design

After the preferred surface arrangement was selected for the pilot plant receiver preliminary baseline design, more detailed design and analyses were performed for both superheater and boiler sections.

The arrangement of superheater passes is shown in Figure V-3. Superheater tubes are 25.4-mm (1-in) O.D. x 2.77-mm (0.109-in) minimum wall and are on a 28.58-mm (1.125-in) centerline. SA-213 TP304H stainless steel tubes are used in passes 1-A, 1-B, 2-A, 2-B, and SA-213 TP316H tubes

Figure V-7 Energy Distributions Along The Height Of Receiver Walls

SCHEME 1

SCHEME 3

SCHEME 2

SCHEME 4

Figure V-8 Superheater Surface Arrangements Approached

in the remaining passes. Eight 88.9-mm (3.5-in) O.D. saturated steam feeders are provided from the drum to the superheater inlet header. All connecting headers and transfer pipes are 273.1-mm (10.75-in) O.D.

Pressure drop and temperatures of steam and tube wall were then calculated for the entire superheater using Foster Wheeler computer programs. For this preliminary design, 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). The steady-state tube wall and steam bulk temperature profiles along the length of the superheater are shown in Figure V-9. In calculating these temperatures, the following flux conditions and flow unbalance effects were considered:

- Tube wall temperature calculation based on maximum attainable flux density,
- Steam temperature calculation based on maximum attainable flux densities less convection, radiation, and insulation losses,
- Flux variation among tubes of the same pass,
- Flow unbalance because of tolerance of tube wall thickness (+ 10%, -0% on minimum wall), and
- Flow unbalance because of connecting headers.

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. The general procedure is summarized as follows:

- Divide boiler section into different circuits having similar heat absorption characteristics.
- Select sizes of downcomers, feeders, headers, and risers.

(Maximum Insolation Conditions)

Figure V-9 Superheater Metal And Steam Temperature Profiles - Pilot Plant Receiver

- Calculate ΔP (head required) for all upflow circuits over a range of flow rates.
- Calculate the head available in downcomer circuits that feed boiler tubes involved.
- Plot the head required and the head available. The intersection between these curves indicates the point at which the system will balance.
- Calculate the entrance velocity, exit steam quality, and circulation ratio and determine if the circulation is adequate.
- If not, change circuitry sizes and/or arrangement and repeat the same calculation procedure until satisfactory circulation is achieved.

The boiler circuitry selected for the preliminary baseline design is shown in Figure V-10. The material list of this arrangement is given in Table V-3. The surfaces at the far side of each side wall, shown slant-shaded in Figure V-10, receive less than 2.5 percent of the total incident energy. Since this amount of energy would not promote adequate circulation should boiler tubes be located there, these surfaces are covered by refractory material. Results of the circulation analysis for this arrangement are summarized in Table V-4. As indicated in this table, all circuits have satisfied the circulation design criteria.

3. Structural Analysis and Design

In order to ensure the structural integrity, lifetime requirements, and performance characteristics of the receiver, a detailed structural analysis of the pressure boundaries, viz., superheater, boiler (waterwall), steam drum, headers, risers, downcomers, and other parts was done. The methods used in structural analysis, the computer programs, and the criteria used in the evaluation and the important results are described below.

The standards and documents used for structural evaluation are discussed in Section (a) below. A parametric study was done to evaluate the effect of important parameters such as heat flux, film coefficient, bulk fluid temperature, and material properties on the structural behavior of the tubes. The results of this study are reported in Section (b). The superheater is one of the most critical components of the receiver because it is subjected to high metal temperatures and stresses, and hence a large bulk of the structural analysis effort was

- LEGEND**
1. Feedwater Pipe	7. Rear Boiler Tubes
2. Steam Drum	8. Side Screen Tubes
3. Downcomer	9. Rear Screen Tubes
4. Lower Header	10. Corner Screen Tubes
5. Feeder	11. Upper Header
6. Side Boiler Tubes	12. Riser

Figure V-10 Boiler Circuitry Of The Pilot Plant Receiver

Table V-3 Material List Of The Pilot Receiver Boiler Circuitry

Item No.	Name	No. Req'd	Material	Description
1	Feedwater Inlet Pipe	1	SA-106 C	114.3 mm (4.5") O. D.
2	Steam Drum	1	SA-516 GR 70	1220 mm (48") O. D.
3	Downcomer	6	SA-106 C	168.3 mm (6.625") O. D.
4	Lower Header	3	SA-106 C	219.1 mm (8.625") O. D.
5	Feeder	28	SA-210 A1	101.6 mm (4") O. D.
6	Side Wall Boiler Tubes	208	SA-210 A1	38.1 mm (1.5") O. D. by 3.76 mm (0.148") Min. Wall on 44.45 mm (1.75") centerline
7	Rear Wall Boiler Tubes	128	SA-210 A1	"
8	Side Wall Screen Tubes	16	SA-210 A1	"
9	Rear Wall Screen Tubes	8	SA-210 A1	"
10	Corner Screen Tubes	6	SA-210 A1	"
11	Upper Header	3	SA-106 C	219.1 mm (8.625") O. D.
12	Riser	40	SA-210 A1	101.6 mm (4") O. D.

Table V-4 Pilot Plant Receiver Boiler Circulation Summary

CIRCUIT IDENT.	FLOW RATE kg/h (lb/h)	VELOCITY ENTERING m/s (ft/s)	STEAM EXIT QUAL. % Wt/Vol	STEAM GENERATED kg/h (lb/h)
Rear Wall Panel	259,650 (572,420)	1.17 (3.83)	13.44/62.89	34,897 (76,933)
Rear Wall Screen Tubes	18,670 (41,160)	1.34 (4.40)	14.82/65.51	2,767 (6,100)
Rear Wall Downcomer	278,320 (613,580)	3.32 (10.88)	-	-
Side Wall Panel 2	241,814 (533,100)	0.99 (3.26)	8.97/51.82	21,691 (47,819)
Side Wall Screen Tubes	18,734 (41,300)	1.35 (4.42)	10.45/56.02	1,958 (4,316)
Side Wall Downcomer	260,548 (574,400)	3.10 (10.18)	-	-
Side Wall Panel 1	122,127 (269,240)	1.03 (3.39)	4.29/32.86	5,239 (11,550)
Side Wall Screen Tubes	16,901 (37,260)	1.21 (3.98)	5.91/40.68	999 (2,202)
Corner Screen Tubes	17,328 (38,200)	1.66 (5.45)	8.74/51.11	1,515 (3,339)
Side Wall Downcomer	156,356 (344,700)	1.86 (6.11)	-	-

Total Circulation Rate: 695,224 kg/h (1,532,680 lb/h)

Steam Generation Rate: 69,065 kg/h (152,259 lb/h)

Flux Condition: Maximum Solar Intensity (1.0226 kW/m²)
Day of Year: 355 (Winter Solstice)
Time of Day: 12 00

Overall Quality - % by Wt: 9.93

Overall Circ. Ratio: 10.07

concentrated on the superheater. This analysis is described in Section (c). The remaining Sections (d), (e), and (f) discuss the waterwall analysis, analysis of other pressure parts, and transient analysis.

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.[8] The design philosophy of Section I is essentially to fix the thickness of pressure boundaries such that the hoop stress is kept within the allowable limit. There are no requirements that the detailed stress distributions within the structure be considered. Localized stresses which may cause fatigue, therefore, cannot be evaluated by this approach. In solar receivers creep and fatigue are important modes of failure 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.[9] 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.[10] Because this is a Code for Nuclear Components, Code Case 1592 has, of necessity, very stringent requirements, and it will be too uneconomical for solar receivers. However, in this report the approach used in Code Case 1592 is used in evaluating the structural integrity of the superheater tube.

• Section VIII, Division 2, Code Limits - The requirements of Section VIII, Division 2, may be summarized as follows:

(i) General primary membrane stress intensity $P_m \leq S_m$. Stress intensity at a point is defined in the Code as twice the maximum shear stress at that point. A primary stress is the stress developed by the imposed loading, which is necessary to satisfy the simple laws of equilibrium of internal forces and moments. S_m is the allowable stress. Generally, it is 2/3 yield stress for ferritic materials and 0.9 yield stress for austenitic materials.

(ii) Local primary membrane stress intensity $P_L \leq 1.5 k S_m$. ("k" is a factor based on the type of loading, and the values of "k" are listed in the Code.[9])

(iii) Primary membrane plus primary bending stress intensity $P_L + P_B \leq 1.5 k S_m$.

(iv) Primary plus secondary stress intensity $P_L + P_B + Q \leq 3 S_m$. A secondary stress is a normal or shear stress developed by the constraint of adjacent parts or by self constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Thermal stresses are secondary stresses.

(v) Primary plus secondary plus peak stress intensity $P_L + P_b + Q + F \leq S_a$ (Fatigue limit). F is the peak stress, the basic characteristic of which is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. S_a is the fatigue allowable stress corresponding to the given number of cycles.

Code.[9] More details about the above limits may be found in the

• Additional Criteria from Code Case 1592 - Three additional criteria from Code Case 1592 relevant in the evaluation of components subjected to high temperature are:

(i) Primary membrane stress intensity $P_m < S_{mt}$: S_{mt} is an allowable stress which is a function of the life of the structure and the operating temperature.

(ii) Ratcheting strain limit: Ratcheting is a progressive cyclic inelastic deformation. The limits on accumulated inelastic strains are:

- Strains averaged through the thickness - 1 percent
- Strains at the surface because of an equivalent linear distribution of strain through the thickness - 2 percent
- Local strains at any point - 5 percent

These limits apply to the maximum positive values of the principal strains. It may be noted that these limits are not necessarily quantitatively related to specific failure modes.

(iii) Creep-fatigue damage limit.

$$\sum_{j=1}^p \left(\frac{n}{N_d} \right)_j + \sum_{k=1}^q \left(\frac{t}{T_d} \right)_k \leq D$$

where:

- D = Creep-fatigue damage limit (usually 1 or less)
N_d = Number of allowable cycles for any given type of strain cycling
T_d = Creep rupture time at a given stress intensity

Creep-fatigue damage evaluation in Code Case 1592 treats compressive and tensile stresses identically. The Code, however, recognizes that for austenitic steels compressive stresses are much less damaging than tensile stresses. In the solar receiver the stresses at elevated temperature (in regions where creep damage is significant) are mostly compressive in nature. Hence this limit may be relaxed.

b. Parametric Study A parametric study was conducted to evaluate the effect of all relevant parameters on the stresses, strains, creep, and fatigue of the tube and thus to select the suitable material and tube size. The parameters considered were (a) heat flux, (b) film coefficient, (c) bulk fluid temperature, (d) tube thickness, and (e) tube material. A summary of the range of parameters considered is given in Table V-2. Four materials were considered for the parametric study, viz., SA-213 TP304H (304 Stainless Steel); SA-213 TP316H (316 Stainless Steel); SM-163-800 Gr. 2 (Incoloy 800) and SA-213 T22 (2-1/4 Cr. 1 Mo). The SA-213 T22 material was eliminated after preliminary consideration because of its poor creep properties.

Two types of superheater tube support schemes were considered. The first scheme, viz., the stringer and sleeve support scheme (Figure V-11) is better from the structural rigidity and vibration points of view. However sleeves prevent the tubes from bowing out and consequently introduce high thermal stresses. The second scheme, viz., the 'floating tube scheme' (Figure V-12) allows the tubes to bow out; consequently, the thermal stresses are reduced considerably. The latter scheme was finally chosen for the pilot plant receiver. However, the parametric study reported in this section was done with the stringer tube arrangement.

Table V-5 shows a typical thickness calculation using ASME Code Section I.[8] The Code formula for thickness is:

$$t = \frac{PD}{2 SE + P} + .005 D + e,$$

where:

- t = minimum required thickness, P = pressure,
D = outer diameter, S = allowable stress,
e = a constant given in the Code (=0 in this case) and
E = efficiency of the joint (=1 for seamless tubes).

Figure V-11 Superheater Support Scheme (Stringer And Sleeve Scheme)

SECTIONAL VIEW OF SIDE WALL

SECTIONAL VIEW OF REAR WALL

Figure V-12 Superheater Support Scheme (Floating Tube Scheme)

The thickness calculations are shown for three different materials and for two different design temperatures. Similar calculations were made for two more outer diameters, viz., 25.4 mm (1 in) and 19.1 mm (.75 in).

- Thermal Stress Analysis - The thermal stress analysis was done using the finite element program ANSYS. This program calculates both temperature distribution and stresses. The finite element model used is shown in Figure V-6. One half of the tube is divided into 90 isoparametric elements. A cosine distribution of the flux is assumed. A plane strain analysis, i.e., assuming that the tube is not allowed to bend or expand axially, is done by using the program. The boundary conditions used in this analysis are listed in Table V-6.

Figure V-13 is a computer plot of the temperature contours (isotherms) in the superheater tube for a typical case. It may be noted that the front side of the tube is subjected to the largest temperature gradient through the thickness. The back of the tube remains approximately at the fluid temperature.

The model used in the computer analysis is a plane strain model. The ends of the tube are prevented from rotating and expanding axially. However, the actual superheater tube has three intermediate supports (stringer and sleeve support scheme), and the ends are allowed to expand and rotate. To correct for this, an axial force and moment as shown in Figure V-14 are applied. A continuous beam analysis is then done, and the stresses are added algebraically to the plane strain solution. The axial force F used for correction is obtained by integrating σ_z (obtained from the plane strain solution) over the cross-sectional area of the tube. Similarly, M_o is obtained by integrating σ_z over the cross-sectional area.

Figure V-15 is a computer plot of the axial stress contours (with only axial force corrections). Note that in the operating mode the axial stress is compressive on the front side (heated side) of the tube and tensile on the back.

The effects of the outer diameter on metal temperatures and maximum stress intensity are plotted in Figures V-16 and V-17. These effects are linear. This study was done for 304 stainless steel. The thickness used is that corresponding to the design temperature of 894 K (1150°F) as shown earlier. The gauge thickness was increased by 10 percent (only in thermal stress analysis) to allow for tolerance. The effect of heat flux on metal temperature is shown in the curves plotted in Figure V-18. Plots are made for three different diameters and the corresponding thicknesses. The effects are again linear.

The film coefficient h_i has a nonlinear effect on the stress intensity and metal temperature. In Figure V-19 the secondary stress intensity (Q) and secondary plus peak stress intensity ($Q + F$)

Table V-5 Thickness Calculations Using Section 1 Of The ASME Code

O.D. = 31.75 mm (1.25 in) Design Pressure = 10 MPa (1450 psi)

MATERIAL	DESIGN TEMPERATURE, K (°F)	ALLOWABLE STRESS, S_o MPa (ksi)	CALCULATED THICKNESS, mm (in)	GAGE THICKNESS mm (in)
304 SS	866 (1100)	66.8 (9.7)	2.36 (0.093)	2.41 (0.095)
	894 (1150)	53.1 (7.7)	2.90 (0.114)	3.05 (0.120)
316 SS	866 (1100)	85.5 (12.4)	1.91 (0.075)	2.11 (0.083)
	894 (1150)	67.6 (9.8)	2.34 (0.092)	2.41 (0.095)
Incoloy 800	866 (1100)	93.1 (13.5)	1.78 (0.070)	1.83 (0.072)
	894 (1150)	77.2 (11.2)	2.08 (0.082)	2.11 (0.083)

Table V-6 Boundary Conditions For Superheater Tube Analysis

BOUNDARY	THERMAL	STRESS
$r = r_o$ $0 < \theta < \pi/2$	$K \partial T / \partial r = -q_o \cos \theta$	$\sigma_{rr} = 0$
$r = r_o$ $\pi/2 < \theta < \pi$	$\partial T / \partial r = 0$	$\tau_{r\theta} = 0$
$r = r_i$ $0 < \theta < \pi$	$K \partial T / \partial r = h_i (T - T_f)$	$\sigma_{rr} = -p$ $\tau_{r\theta} = 0$
$r_i \leq r \leq r_o$ $\theta = 0$	$\partial T / \partial \theta = 0$	$v = 0$
$r_i \leq r \leq r_o$ $\theta = \pi$		$\tau_{r\theta} = 0$
$z = 0$	$\partial T / \partial z = 0$	$w = \text{constant}$
$z = 1$		$\int_A \sigma_{zz} dA = 0$

Material: 304 SS

OD = 25.4 mm (1 in.), $t = 2.65$ mm (0.1045 in.)

$q = 126.2$ kW/m² (40,000 Btu/hr ft²)

$h_i = 2.612$ kW/m² K (460 Btu/hr ft² °F)

$T_f = 786$ K (955°F)

Figure V-13 Temperature Distribution In Superheater Tube

Figure V-14 Moment And Axial Force Corrections (Stringer and Sleeve Scheme)

Figure V-15 Axial Stress Distribution In Superheater Tube

Figure V-16 Effect Of Outer Diameter On Tube Wall Temperature

Figure V-17 Effect Of Outer Diameter On Stress Intensity

Figure V-18 Effect of Flux On Tube Wall Temperature

are plotted against film coefficient. The peak stress is in element 5 [shown in Figure V-6(a)]. Figure V-20 shows the effect of the film coefficient on $T_{\max}$ and T_{av} . $T_{\max}$ is the absolute maximum temperature in the tube wall, and T_{av} is the wall-averaged temperature on the heated side of the tube. The material in both the above cases is 304 stainless steel. It is clear that a decrease in the film coefficient has a considerably adverse effect on the metal temperature and stresses.

To study the effect of material properties, the thermal stresses were evaluated for two different materials at two different temperatures. The results are shown in Table V-7. As can be seen from the table, the stresses and the metal temperatures are fairly close in all four cases. The allowable stresses and strains vary considerably for different materials and temperatures.

Table V-8 shows the $P_L + P_R + Q$ calculations for the range of flux varying from 221 to 379 kW/m^2 (10,000 to 120,000 Btu/h-ft²). These quantities are compared with the allowable limit, viz., $3 S_m$. This limit is intended to ensure that the structure shakes down. (A structure is said to shake down if, after a few cycles of operation, the response is elastic.) The S_m value used is the average of S_m values at the highest and lowest temperature in a cycle. Q is the secondary stress intensity which is the equivalent linear part of the thermal stress.

The general primary membrane stresses and their allowable values (S_{mt}) are shown in Table V-9 for all three materials and two different thicknesses. S_{mt} values are given for 100,000 hours of actual operating time.

- Fatigue - The results of elevated temperature fatigue calculations for various heat fluxes are shown in Table V-10. The fatigue curves in Code Case 1592 are used to estimate the fatigue life. Figures T-1430-1A and 1B of Ref. [10] are the curves to be used for elastic analysis. Figures T-1420-1A and 1B of Ref. [10] are to be used for inelastic analysis. The numbers in the last column are given only to indicate that an inelastic analysis might yield a much higher value for fatigue life.

- Creep - The results of creep life calculations based on elastic analysis are given in Table V-11. The creep rupture curves of Code Case 1592 are used in these calculations. The first column under each material is based on the Code Case requirement of 11 percent margin on stresses. The second column is based on the minimum creep rupture properties. It should be emphasized that these calculations assume identical creep rupture properties under tension and compression, which may yield very conservative results in this case. The creep life values shown are in hours.

Figure V-19 Effect Of Film Coefficient On Stress Intensity

Figure V-20 Effect of Film Coefficient On Tube Wall Temperature

Table V-7 Effect Of Material Properties

$q = 126.2 \text{ kW/m}^2$ (40,000 Btu/h- ft^2) $t = 2.65 \text{ mm}$ (0.1045 in)
O.D. = 25.4 mm (1 in) $h_i = 2.612 \text{ kW/m}^2\text{-K}$ (460 Btu/h- $\text{ft}^2\text{-}^\circ\text{F}$)
 $T_F = 786 \text{ K}$ (955 $^\circ\text{F}$)

MATERIAL PROPERTIES	T_{av} K ($^\circ\text{F}$)	T_{av} K ($^\circ\text{F}$)	Q MPa (psi)	Q + F MPa (psi)
304 SS at 1050 $^\circ\text{F}$	844 (1060)	852 (1074)	126.0 (18,269)	164.0 (23,790)
304 SS at 900 $^\circ\text{F}$	845 (1062)	854 (1077)	130.7 (18,953)	171.4 (24,862)
Incoloy 800 at 1050 $^\circ\text{F}$	844 (1059)	851 (1072)	118.1 (17,133)	153.5 (22,267)
Incoloy 800 at 900 $^\circ\text{F}$	845 (1061)	853 (1075)	117.4 (17,032)	154.1 (22,345)

Table V-8 Section VIII Code Compliance

O.D. = 25.4 mm (1.0 in) $t = 2.65 \text{ mm}$ (0.1045 in) $T_F = 680 \text{ K}$ (765 $^\circ\text{F}$)
 $h_i = 3.35 \text{ kW/m}^2\text{-K}$ (590 Btu/h- $\text{ft}^2\text{-}^\circ\text{F}$) Material: 304 SS
Pressure: 10.0 MPa (1450 psi)

q kW/m^2 (Btu/hr- ft^2)	T_{max} K ($^\circ\text{F}$)	$P_L + P_b + Q$ MPa (psi)	ALLOWABLE STRESS $\frac{3}{4} S_m^*$ MPa (psi)
379 (120,000)	850 (1071)	299.0 (43,369)	318.5 (46,200)
347 (110,000)	836 (1046)	276.2 (40,055)	320.6 (46,500)
316 (100,000)	822 (1020)	252.6 (36,631)	322.7 (46,800)
284 (90,000)	808 (995)	229.6 (33,300)	324.7 (47,100)
252 (80,000)	794 (969)	206.4 (29,948)	326.3 (47,325)
221 (70,000)	780 (944)	183.3 (26,587)	327.8 (47,550)
* $S_m = 1/2 [S_m(T_{max}) + S_m(519 \text{ K})]$			

Table V-9 Code Case 1592 Compliance: Limit On General Primary Membrane Stress Intensity ($P_m < S_{mt}$)

O.D. = 25.4 (1 in) Pressure = 10.0 MPa (1450 psi), $T_{av} = 866$ K (1100°F)

MATERIAL	t_d mm (in)	P_m MPa (psi)	S_{mt} (10^5 h = 34.2 yr) MPa (psi)
304 SS	1.98 (0.078)	59.1 (8,570)	46.9 (6,800)
	2.41 (0.095)	47.6 (6,905)	
316 SS	1.65 (0.065)	71.9 (10,429)	65.5 (9,500)
	1.98 (0.078)	59.1 (8,570)	
Incoloy 800	1.47 (0.058)	81.2 (11,885)	80.7 (11,700)
	1.83 (0.072)	64.4 (9,344)	

Table V-10 Fatigue Life (Elastic Analysis)

Material: 304 SS, O.D. = 25.4 mm (1 in), $T = 2.65$ mm (.1045 in)

Operating Pressure = 10.0 MPa (1450 psi), Hot Standby Pressure = 2.41 MPa (350 psi)

$h_i = 3.35$ kW/m²-K (590 Btu/h-ft²-°F), $T_{max} = 811 - 894$ K (1000 - 1150°F)

q kW/m ² (Btu/h-ft ²)	$\Delta \epsilon_R$ 10^{-3} mm/mm	N_D (From Figure T-1340-1A,1B, Ref. 10)	N_D (From Figure T-1420-1A,1B Ref. 10)
379 (120,000)	2.39	80	8,000
316 (100,000)	1.99	150	20,000
252 (80,000)	1.59	500	50,000
189 (60,000)	1.19	4,000	10^6

c. Superheater Analysis The model and the computer program used in the thermal stress analysis of the superheater are the same as those used in the parametric study (Section A.b). The axial force and moment corrections are, however, different because of the support conditions. The new corrections are shown in Figure V-21. The unknown forces and moments F_1 , F_2 , M_1 , M_2 are obtained from the following compatibility equations:

$$\delta_1 = \delta_2, \theta_1 = \theta_2$$

$$F_1 + F_2 = F_0, M_1 + M_2 = M_0$$

where

$$F_0 = \int_A \sigma_z dA, M_0 = \int_A \sigma_z y dA$$

δ_1, θ_1 = Horizontal displacement and slope of point (1) on the beam

δ_2, θ_2 = Horizontal displacement and slope of point (2) on the cantilever

The maximum heat fluxes, metal temperatures, and peak stresses in all passes are tabulated in Table V-12. Based on this table, passes 2, 3A, and 5A were identified as the critical passes. Passes 3A and 5A are made of 316 stainless steel, and they have the highest metal temperatures and stresses. Even though the metal temperatures in pass 2 are less, it was also selected as a critical pass because it is made of a different material, viz., 304 stainless steel. The detailed analysis of these three passes is reported in this section.

Figure V-22 is a computer plot of the temperature contours (isotherms) in pass 5A. The peak temperature during steady-state operation in pass 5A is 901 K (1162°F), and the peak temperatures in all the other passes are lower. Figure V-23 is a plot of the axial stress contours in pass 5A. The temperature and axial stress plots for pass 1A are given in Figures V-24 and V-25. The stress and temperature contours of the other passes are similar and hence not shown in this report. Tables V-13, V-14, and V-15 show the Section VIII Code calculations for passes 2, 3A, and 5A. It may be noted that the limits on P_m , P_L , $P_L + P_B$ and $P_L + P_B + Q$ are satisfied in all the three cases. The value of k is 1.

The S_{mt} limits on primary stresses using Code Case 1592 are shown in Table V-16. These S_{mt} values correspond to 100,000 hours of operating time. The elastic fatigue calculations of passes 2, 3A, and 5A are shown in Table V-17. The elastic calculations show that the number of allowable cycles is rather low. However, if the elastic strain range and the "inelastic fatigue curves" [Figure T-1420-1A and 1B of Ref. (10)] are used, it can be seen that the number of allowable cycles is two orders of magnitude higher. This indicates that inelastic analysis is needed to estimate the fatigue life more realistically.

Table V-11 Creep Life (Elastic Analysis)

TEMP. K (°F)	304 SS		316 SS		INCOLOY 800	
	With Extra 11% Margin	Based on Min Prop.	With Extra 11% Margin	Based on Min Prop.	With Extra 11% Margin	Based on Min Prop.
783 (950)	>300,000	>300,000	>300,000	>300,000	>300,000	>300,000
811 (1000)	73,600	148,700	>300,000	>300,000	>300,000	>300,000
825 (1025)	35,700	75,400	>200,000	>250,000	>300,000	>300,000
839 (1050)	16,100	32,800	104,500	214,300	>300,000	>300,000
853 (1075)	8,300	16,900	45,900	93,200	>200,000	>300,000
866 (1100)	5,300	8,100	20,200	37,800	>200,000	>300,000
880 (1125)	2,100	4,300	10,800	19,700	70,600	178,000
894 (1150)	1,100	2,200	4,800	10,000	28,400	68,000
908 (1175)	600	1,300	2,500	5,000	12,000	30,000
922 (1200)	320	670	1,300	2,500	5,000	10,600

Table V-12 Critical Locations In Various Passes

Materials: 316 SS (Passes 3 to 6), 304 SS (Passes 1 & 2)

$$h_1 = 4.713 \text{ kW/m}^2 \text{ K (830 Btu/h-ft}^2\text{-}^\circ\text{F)} - \text{Passes 1 \& 5}$$

$$3.975 \text{ kW/m}^2 \text{ K (700 Btu/h-ft}^2\text{-}^\circ\text{F)} - \text{Pass 6}$$

PASS NUMBER	MAX. HEAT FLUX (Q_o) kW/m ² (Btu/hr ft ²)	BULK TEMPERATURE	T_{ay} K (°F)	T_{max} K (°F)	Q + F MPa (psi)
1A	505 (160,000)	611 (640)	769 (924)	805 (990)	363.4 (52,700)
1B	505 (160,000)	611 (640)	769 (924)	805 (990)	363.4 (52,700)
2*	505 (160,000)	644 (700)	802 (984)	839(1050)	363.4 (52,700)
3A*	505 (160,000)	672 (750)	830(1034)	866(1100)	363.4 (52,700)
3B	505 (160,000)	672 (750)	830(1034)	866(1100)	363.4 (52,700)
4A	410 (130,000)	733 (850)	861(1091)	891(1144)	295.1 (42,800)
4B	410 (130,000)	722 (840)	850(1071)	880(1124)	295.1 (42,800)
5A*	410 (130,000)	743 (878)	871(1109)	901(1162)	295.1 (42,800)
5B	410 (130,000)	739 (870)	867(1101)	896(1154)	295.1 (42,800)
6	227 (72,000)	789 (960)	869(1104)	885(1133)	175.1 (25,400)

Figure V-21 Moment And Axial Force Corrections (Floating Tube Scheme)

Figure V-22 Temperature Distribution In Superheater Tube (Pass 5A)

Figure V-23 Axial Stress Distribution In Superheater Tube (Pass 5A)

Figure V-24 Temperature Distribution In Superheater Tube (Pass 1A)

Figure V-25 Axial Stress Distribution In Superheater Tube (Page 1A)

Table V-13 Superheater Analysis (Pass 2)

Material: 304 SS Pressure = 11.72 MPa (1700 psi)
 O.D. = 25.4 mm (1 in), $t = 3.05$ mm (.12 in), $T_{\max} = 839$ K (1050°F)
 $T_{\text{av}} = 802$ K (934°F), $q = 505$ kW/m² (160,000 Btu/h-ft²)
 $h_i = 4.713$ kW/m²-K (830 Btu/h-ft²-°F), $T_F = 644$ K (700°F)

CATEGORY	STRESS MPa (psi)	ALLOWABLE MPa (psi)
P_M	43.0 (6,233)	93.8 (13,600)
P_L	43.0 (6,233)	140.0 (20,400)
$P_L + P_B$	43.0 (6,233)	140.7 (20,400)
$P_L + P_B + Q$	213.0 (30,900)	281.3 (40,800)

Table V-14 Superheater Analysis (Pass 3A)

MATERIAL: 316 SS PRESSURE = 11.72 MPa (1700 psi)
 $t = 3.05$ mm (.12 in), $T_F = 672$ K (750°F) $T_{\max} = 866$ K (1100°F)
 $T_{\text{av}} = 830$ K (1034°F) $q = 505$ kW/m² (160,000 Btu/h-ft²)
 $h_i = 4.713$ kW/m²-K (830 Btu/h-ft²-°F), O.D. = 25.4 mm (1 in)

CATEGORY	STRESS MPa (psi)	ALLOWABLE MPa (psi)
P_M	43.0 (6,233)	102.0 (14,800)
P_L	43.0 (6,233)	153.1 (22,200)
$P_L + P_B$	43.0 (6,233)	153.1 (22,200)
$P_L + P_B + Q$	213.0 (30,900)	306.1 (44,400)

Table V-15 Superheater Analysis - Pass 5A

MATERIAL: 316 SS Pressure = 11.72 MPa (1700 psi)
 O.D. = 25.4 mm (1 in), $t = 3.05$ mm (.12 in), $T_F = 743$ K (878°F)
 $T_{\max} = 901$ K (1162°F), $T_{\text{av}} = 871$ K (1109°F)
 $q = 410$ kW/m² (130,000 Btu/h- ft²), $h_i = 4.713$ kW/m³
 (830 Btu/h- ft²-°F)

CATEGORY	STRESS MPa (psi)	ALLOWABLE MPa (psi)
P_M	43.0 (6,233)	101.2 (14,680)
P_L	43.0 (6,233)	151.8 (22,020)
$P_L + P_B$	43.0 (6,233)	151.8 (22,020)
$P_L + P_B + Q$	173.1 (25,100)	303.6 (44,040)

Table V-16 Primary Stress Limits Using Code Case 1592

MATERIAL: PASS 2 - 304 SS
 Pass 3A, 5A - 316 SS

O.D. = 25.4 mm (1 in), $t = 3.05$ mm (.12 in)

PASS NUMBER	T_{av} K (°F)	P_M MPa (psi)	ALLOWABLE STRESS S_{mt} @ 10^5 h MPa (psi)
2	802 (984)	43.0 (6,233)	83.4 (12,100)
3A	830 (1,034)	43.0 (6,233)	92.4 (13,400)
5A	871 (1,109)	43.0 (6,233)	62.7 (9,100)

Inelastic analyses of passes 2 and 5A were done using the finite element program MARC. ANSYS was not used in this analysis because it does not have the provision to use the type of creep law appropriate for 304 stainless steel and 316 stainless steel. A generalized plane strain model was used in the analysis. The creep and plastic properties were taken from Ref. [11].

Figure V-26 shows a stress strain histogram for the critical point (element 5) in pass 5A for the first five diurnal cycles. This diagram is utilized to evaluate creep, fatigue, and ratcheting characteristics of the superheater tube. The points a, b, c, d, e, and f describe the first cycle starting from and returning to hot standby condition. The gaps in the loading and unloading lines appear owing to residual stresses induced by plastic flow and creep. In the figure the curve 'abc' indicates loading from hot standby to peak operating conditions, 'cd' indicates the 8-hour operation at the peak condition, and 'def' corresponds to the unloading from peak operating mode to hot standby. Inelastic analysis was also done for pass 2 (304 stainless steel).

The strain ranges obtained from the inelastic analysis were used to evaluate the allowable number of cycles (N_d). The results are shown in Table V-18.

The creep life calculations using the inelastic analysis are shown in Tables V-19 and V-20. The creep rupture curves of Code Case 1592 were used in this evaluation. The rupture times correspond to the minimum properties.

The incremental inelastic strain components for element 5 in pass 5A are shown in Table V-21. The last column shows the cumulative inelastic strain ϵ_2 (the circumferential strain).

d. Waterwall (Boiler) Analysis The determination of the boiler tube wall thickness was done according to the Section I formula. The material used is SA-210 A1. The outer diameter is 38.1 mm (1.5 in) and the thickness is 3.76 mm (.148 in). The thickness was increased by 20 percent for thermal analysis to allow for the tolerance. The thermal stress analysis was done using a finite element model shown in Figure V-6(b) and the finite element program ANSYS. The boundary conditions used in the finite element program are given in Table V-23. The stress calculations and the Section VIII Code limits are shown in Table V-22. The temperatures and thermal stresses correspond to the peak flux of 694 kW/m^2 ($220,000 \text{ Btu/h-ft}^2$). The fatigue curves of Section VIII, Division 2, were used to evaluate the allowable number of cycles.

e. Thickness Calculations for Other Pressure Parts The thickness calculations for the downcomer, waterwall headers, feeders and risers, superheater supply tube, superheater outlet pipe, superheater headers, and steam drum are shown in Table V-24. These calculations were done using the appropriate tube and pipe formulae of Section I.

Table V-17 Fatigue Analysis (Elastic)

Material: Pass 2 - 304 SS
Pass 3A, 5A - 316 SS

O. D. = 25.4 mm (1 in) $t = 3.05$ mm (.12 in)
 $h_i = 4.713$ kW/m² K (830 Btu/h-ft²-°F)

PASS NUMBER	kW/q m ² ₂ Btu/hr ft ²	T _{max} K (°F)	$\Delta \epsilon_R$ (10 ⁻³ mm/mm)	N _D (from Fig. T-1420-1A, 1B, C.C. 1592)	N _D (from Fig. T-1420-1A, 1D, C.C. 1592)
2	505 (160,000)	839 (1050)	1.97	150	20,000
3A	505 (160,000)	866 (1100)	1.97	150	20,000
5A	410 (130,000)	901 (1162)	1.60	500	50,000

Table V-18 Fatigue Analysis (Inelastic)

Material: Pass 2 - 304 SS
Pass 3A, 5A - 316 SS

O. D. = 25.4 mm (1 in) $t = 3.05$ mm (.12 in)
 $h_i = 4.713$ kW/m² K (830 Btu/h-ft²-°F)

PASS NUMBER	T _{max} K (°F)	Δε ₁	Δε ₂	Δε ₃	Δε _r ^e	FATIGUE LIFE (N _D)
		(10 ⁻³ mm/mm)				
2	839 (1050)	7.439	7.128	4.202	2.062	≈ 19,000
3A, 5A	866 - 901 (1100-1162)	8.826	8.590	5.513	2.134	≈ 16,000

Figure V-26 Effective Stress Vs. Effective Strain For Element 5
Of Pass 5A

Table V-19 Creep Damage Evaluation (Inelastic)

Material: 304 SS (Pass 2) $T_F = 644 \text{ K (700}^\circ\text{F)}$

$h_i = 4.712 \text{ kW/m}^2\text{-K (830 Btu/h-ft}^2\text{-}^\circ\text{F)}$

O.D. = 25.4 mm (1 in) $t = 3.05 \text{ mm (0.12 in)}$

FRACTION OF PEAK FLUX	$T_{\max}$ K ($^\circ\text{F}$)	AVG. EFF. STRESS IN 8 HOURS OPERATION MPa (ksi)	CREEP LIFE T_D (HOURS)	ESTIMATED OPERATION TIME (HOURS)
1.0	839 (1050)	106.9 (15.5)	30,000	10,900
0.9	819 (1015)	105.5 (15.3)	110,500	27,400
0.8	800 (980)	108.2 (15.7)	3×10^5	23,100
0.7	780 (945)	109.6 (15.9)	$>> 3 \times 10^5$	13,600
0.6	761 (910)	111.7 (16.2)	$>> 3 \times 10^5$	56,300

Table V-20 Creep Damage Evaluation (Inelastic)

Material: 316 SS (Pass 5A) $T_F = 743 \text{ K (878}^\circ\text{F)}$

$h_i = 4.712 \text{ kW/m}^2\text{-K (830 Btu/h-ft}^2\text{-}^\circ\text{F)}$

O.D. = 25.4 mm (1 in), $t = 3.05 \text{ mm (.12 in)}$

FRACTION OF PEAK FLUX	$T_{\max}$ K ($^\circ\text{F}$)	AVG. EFF. STRESS IN 8 HOURS OPERATION MPa (ksi)	CREEP LIFE T_D (HOURS)	ESTIMATED OPERATION TIME (HOURS)
1.0	901 (1162)	106.9 (15.5)	10,300	10,900
0.9	885 (1134)	105.5 (15.3)	23,900	27,400
0.8	869 (1105)	104.8 (15.2)	60,300	23,100
0.7	854 (1077)	106.9 (15.5)	143,000	13,600
0.6	838 (1048)	108.2 (15.7)	$> 3 \times 10^5$	56,300

Table V-21 Ratcheting Analysis (Inelastic) Pass 5A

Material: 316 SS $T_{\max} = 901 \text{ K (1162}^{\circ}\text{F)}$
 $T_F = 743 \text{ K (878}^{\circ}\text{F)}$ $h_i = 4.712 \text{ kW/m}^2 \text{ K (830 Btu/h-ft}^2\text{-}^{\circ}\text{F)}$
 O. D. = 25.4 mm (1 in) $t = 3.05 \text{ mm (.12 in)}$

CYCLE NUMBER	10^{-3} mm/mm			MAX CUMULATIVE PRINCIPAL STRAIN ϵ_2 (10^{-3} mm/mm)
	$\Delta\epsilon_1$	$\Delta\epsilon_2$	$\Delta\epsilon_3$	
1	.4042	.6974	-1.1017	0.6974
2	-.1307	.1813	-0.0504	0.8787
3	-.0810	.1159	-.0347	1.0995
4	-.0777	.1049	-.0266	1.0995
5	-.0708	.0677	-.0218	1.1672
6	-.0508	.0671	-.0162	1.2343
7	-.0490	.0614	-.0126	1.2957
8	-.0430	.0532	-.0115	1.3489

Table V-22 Boiler Tube Analysis

O. D. = 38.1 mm (1.5 in) $t = 4.52 \text{ mm (0.178 in)}$ Material: SA-210 A1
 Pressure = 11.72 MPa (1700 psi)
 $h_i = 17.03 \text{ kW/m}^2 \text{ K (3000 Btu/h-ft}^2\text{-}^{\circ}\text{F)}$ $T_{\max} = 713 \text{ K (824}^{\circ}\text{F)}$
 $q = 694 \text{ kW/m}^2$ (220,000 Btu/h-ft²) $T_F = 583 \text{ K (590}^{\circ}\text{F)}$

CATEGORY	STRESS MPa (psi)	ALLOWABLE MPa (psi)	ALLOWABLE CYCLES
P_M	52.4 (7,650)	75.5 (10,800)	-
P_L	52.4 (7,650)	111.7 (16,200)	-
$P_L + P_B$	52.4 (7,650)	111.7 (16,200)	-
$P_L + P_B + Q$	91.3 (27,740)	223.4 (32,400)	-
$P_L + P_B + Q + F$	243.3 (35,290)	-	100,000

Table V-23 Boundary Conditions for Waterwall Analysis

BOUNDARY	THERMAL	STRESS
$r = r_o$ $0 \leq \theta \leq \pi/2 - \phi$	$k \partial T / \partial r = -q''_o \cos \theta$	$\sigma_{rr} = 0$
$r = r_o$ $\pi/2 + \phi \leq \theta < \pi$	$\partial T / \partial r = 0$	$\tau_{r\theta} = 0$
$r = r_i$ $0 \leq \theta \leq \pi$	$k \partial T / \partial r = h_i (T - T_f)$	$\sigma_{rr} = -p$ $\tau_{r\theta} = 0$
$r_i \leq r \leq r_o$ $\theta = 0$	$\partial T / \partial \theta = 0$	$V = 0$
$r_i \leq r \leq r_o$ $\theta = \pi$		$\tau_{r\theta} = 0$
$z = 0$	$\partial T / \partial z = 0$	$W = \text{constant}$
$z = 1$		$\int_A \sigma_{zz} da = 0$
$x = w/2$ $c \leq y \leq d$	$k \partial T / \partial x = -q''_o$	$\sigma_{xx} = 0$
$x = -w/2$ $c \leq y \leq d$	$\partial T / \partial x = 0$	$\tau_{xy} = 0$
$-w/2 \leq x \leq w/2$ $y = d$	$\partial T / \partial y = 0$	$V = \text{constant}$ $\tau_{xy} = 0$

f. Transient Analysis The thermal transient limitations during startup and shutdown were determined by using Chart 16 of the Schneider Charts.[12] For a prescribed thickness of the metal and heat-up rate of its surface, these charts yield a corresponding 'through-the-thickness temperature gradient' (ΔT). The allowable stress for 10^5 cycles was obtained from the Section VIII fatigue curves and the ΔT corresponding to this stress was calculated. Allowable heatup or cool-down rates were then computed by using the Schneider charts. The results for various metal thicknesses are shown in Table V-25.

The transient analysis of the superheater tubes is being done using the finite element program ANSYS. The effect of re-radiation is also being taken into account. The flux is assumed to vary linearly, and the variation of the pressure, flow rate, and fluid temperature are shown in Figures V-28 and V-29 of section 4.

4. Operational Characteristics and Control Concept

a. Steady-State Performance During steady-state operation, diurnal and seasonal variations in incident heat flux distribution between superheater and boiler surfaces must be accommodated. Attention was first focused on the steam temperature control concept for steady-state operation and then extended to transient operations. Superheater outlet steam temperature can be controlled internally to the receiver by cooling superheated steam with incoming feedwater or externally to the receiver by controlling the incident heat flux to the receiver cavity. The external control concept requires control of heliostats in the collector subsystem that is beyond the scope of receiver design. Therefore, only the former concept was considered.

Within the receiver two methods of superheat control can be used; spray control in which feedwater is injected directly into superheated steam in an unheated header to reduce the steam temperature, or condenser control in which heat is removed by condensing a portion of the steam in a surface-type condenser cooled by feedwater. Spray control was chosen over condenser control because of its more rapid control response and lower cost. In a typical spray attemperator, a spray nozzle assembly is installed in a header containing two thermowells (one at each end), a disc, and a liner for most of the header length. The spray nozzle is either a fixed-orifice or a spring-loaded variable-orifice type. Feedwater is mechanically atomized by the spray nozzle into a fine, hollow cone-shaped spray, which is mixed with the steam. The liner is installed adjacent to and downstream of the spray nozzle to protect the header walls from thermal shock caused by the temperature difference between the spray water and the superheated steam. The disc, located immediately upstream of the nozzle, promotes rapid mixing of steam and spray water.

Two stages of attemperation were anticipated for superheater temperature control in the pilot plant receiver. The first-stage

Table V-24 Thickness Calculations For The Pressure Components Of The Receiver

COMPONENT(S)	MATERIAL	DESIGN TEMPERATURE K (°F)	OUTER DIAMETER mm (in)	THICKNESS CALCULATED mm (in)	SCHEDULE NUMBER IF PIPE
Downcomer	SA 106-C	644 (700)	219.1 (8.625)	11.2 (0.440)	80
Waterwall Headers	SA 106-C	644 (700)	219.1 (8.625)	14.9 (0.585)	120
Feeders and Risers (tube)	SA 210-C	644 (700)	127.0 (5.0)	6.8 (0.269)	8.0 mm* (.313 in)
Superheater Supply Tube	SA 210-C	644 (700)	88.9 (3.5)	4.8 (0.188)	40
Superheater Outlet Pipe and Headers	SA 335 (P-22)	789 (960)	273.1 (10.75)	22.6 (0.891)	160
Steam Drum	SA 515	644 (700)	ID = 1219.2 (48)	99 (3.9)	101.6 mm (4.0 in)

*Schedule numbers are not applicable. Chosen thicknesses are given.

Table V-25 Thermal Transient Limitations During Startup and Shutdown

THICKNESS mm (in)	ALLOWABLE ΔT FOR 10^5 CYCLES	ALLOWABLE HEATUP OR COOLDOWN RATE
101.6 (4.0)	56 K (100° F)	5 K/min (9° F/min)
25.4 (1.0)	56 K (100° F)	83 K/min (150° F/min)
12.7 (0.5)	56 K (100° F)	3056 K/min (5500° F/min)

attemperator, located between pass 2 and pass 3 of the superheater, is sufficient for steady-state control. The second-stage attemperator, located between pass 4 and pass 5, is used to provide additional temperature control capability during such transient operations as start-up. The control system arrangement of this two-stage attemperation is shown schematically in Figure V-4. Temperature and pressure are sensed at the outlet of each attemperator. Pressure feedback is used by the control regulator to determine the saturation temperature at the attemperator outlet. Spray flow to each attemperator is regulated so that the steam temperature leaving the attemperator is at least 11.1 K (20°F) above saturation. Each attemperator flow control valve is roughly positioned by the final steam flow according to a pre-determined correlation between spray flow and final steam flow. Fine positioning of these valves is controlled by the final steam temperature at the superheater outlet. During control operation the first-stage attemperator is actuated first. The spray flow rate increases with the increasing requirement for de-superheat until the mixing temperature at its outlet reaches 11.1 K (20°F) above saturation. At this point the second-stage attemperator is brought into action to provide additional spray flow at the downstream superheater section for control of the final steam temperature.

b. Start-up and Shutdown Transients A comparison of start-up and shutdown transients shows that the latter is much easier than the former for the receiver system to accommodate. During a shutdown transient, since the incident heat flux to the receiver is cut off, heat exchange is between the boiler, superheater, and cavity interior surfaces and from the receiver exterior surfaces to its surroundings. The closed cavity door and wall insulation prevent any drastic heat loss from the receiver. Heat soaks back into the boiler section at a moderate rate, and any slight over-pressure caused by this heat flow in the initial period of the transient can be vented off either through the superheater or directly from the steam drum. The superheater section, when experiencing gradual decreases in temperature level and front-to-back wall temperature gradients, is subjected to a stress condition no more severe than that encountered during steady-state operation.

A start-up transient, on the other hand, is more difficult for a receiver, particularly a superheater, to handle. One example of this complicated phenomenon is the beginning of a start-up transient, where the boiler generates insufficient steam (or no steam at all) to flow into superheater panels and provide the necessary cooling of the heated tubes. Pressures, temperatures, and flow rates must be thoroughly analyzed for any proposed start-up transient. To obtain this essential information so that transient stress analyses could be performed, a computer program was developed. This 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 calculations for each time interval. Figure V-27 shows the analytical model used for the transient study. The superheater was divided into three

*These represent the portions of the superheater tubing not exposed to incident radiant flux.

Figure V-27 Analytical Model For Thermal/Hydraulic Transient Study

sections with each section representing two superheater passes. Spray attemperation was incorporated between sections 1 and 2 and sections 2 and 3. The method and procedure of calculation of this computer program are outlined below.

- (1) Select time intervals and rates of increase of incident heat energy and drum pressure.
- (2) Proportion input heat energy among boiler and three superheater sections according to proper distribution.
- (3) Calculate energy required to raise enthalpy of water and temperature of metal parts in boiler during time interval according to rate of increase of drum pressure (saturation temperature).
- (4) Calculate the steam flow that can be generated by the total energy into boiler section less that calculated in (3) above.
- (5) Divide superheater into the three sections and sixteen components as shown in Figure V-27.
- (6) Assume a steam temperature leaving superheater section 1. Calculate average metal temperature and heat stored in the metal of each component of section 1. Sum total heat stored in section 1 components and calculate steam temperature leaving section 1. Repeat until assumed and calculated steam temperatures approach each other within a pre-determined tolerance.
- (7) Calculate steam temperatures leaving sections 2 and 3 by repeating procedure described above.
- (8) Check whether final steam outlet temperature exceeds a pre-set value. If it does, repeat spray flow rate calculations required for each attemperator to bring temperature within its assumed limit.
- (9) Continue same procedure for next time interval until complete transient period is covered.

The results of a hot start-up transient analysis are shown in Figures V-28 and V-29. The drum pressure was assumed at 3620 kPa (525 psig) after overnight shutdown and was programmed for rise to full

Figure V-28 Predicted Pressures And Flow Rates For An Assured Hot Start-Up Transient

Figure V-29 Feedwater Temperature And Predicted Steam Temperatures For An Assumed Hot Start-Up Transient

pressure of 9343 kPa (1355 psig) one hour from sunrise. The incident heat energy distributions on boiler and superheater passes were taken from Table 3 of Section IV of this report for winter solstice. Sunrise was assumed at 0712. Energy increase rates between the reported values at 0712, 0800, 1000, and 1200 were assumed to be linear with time. The feedwater temperature vs. time assumed in this transient case are shown in Figure V-29 along with calculated steam temperatures at the inlet and outlet of each superheater section. As indicated in Figure V-28, both attemperators are actuated at approximately 14 minutes from sunrise. As expected, the steam temperature is controlled primarily by the first-stage attemperator. The second-stage attemperator is used only when the steam temperature leaving the first attemperator is 11.1 K (20°F) above saturation and additional spray flow is necessary.

B. RESEARCH EXPERIMENT RECEIVER

1. Objective and Basic Requirements of the Receiver Experiment

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 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.

2. Scaling Criteria

The pilot plant receiver design does not lend itself to being split up into identical modules, one of which could be tested in the receiver experiments. Instead, it must be scaled down as a unit. However, complete geometric similarity is not possible, or even desirable, because it is important to maintain full scale in certain critical areas as pointed out below.

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 projected surface area be proportional to total absorbed heat flux, with the result that linear dimensions are scaled down in proportion to the square root of total absorbed heat flux. Hence the internal dimensions of the experimental receiver cavity were 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. The experimental receiver cavity dimensions thus obtained are shown in Figure V-30.

Drum diameter and thickness are made the same to obtain the same stress intensities under equivalent transient temperature and pressure conditions and to provide for identical arrangement and positioning of drum internals (steam separators and moisture eliminators, or driers) for steam purity control.

Identical diameters and thicknesses of tubes are used together with a tube support scheme and spans between supports as nearly identical

Figure V-30 Experimental Receiver Cavity Dimensions

as scaled-down dimensions will allow so that similar stresses and strains will be developed in the tubes under similar pressure and temperature conditions, both steady-state and transient.

Identical loading on steam drum internals is maintained to provide an equivalent test of steam purification equipment. Exact duplication of loading may not be possible because drum internals are made up of an integral number of fixed-size modules.

Steam/water circulation (velocity and steam fraction) cannot be made identical because the heat input per tube is lower in the experimental receiver in direct proportion to the reduction in tube length; and hence for equal velocities of water entering the tubes, the fractional steam content at corresponding points will be proportionately less in the experimental receiver. A possible compromise can be reached by reducing velocities entering the tubes of the experimental receiver to obtain equal steam fractions at corresponding points along the tubes. This would be a more severe test in the experimental receiver in terms of possible tube burnout. An alternative is to bias the heat flux density pattern to provide equal heat input per tube to a selected number of tubes in a critical zone. This would not necessarily provide exactly the same velocities in the experimental receiver unless the elevation of its drum were made out of proportion to provide the requisite net natural-circulation head.

In a similar vein, it is not possible to duplicate steam temperatures and mass velocities in the superheater. To obtain desired mass velocities, fewer tubes in parallel, and hence a greater number of passes, must be used in the superheater of the experimental receiver. Thus the steam temperature rise per pass is much less in the experimental receiver.

3. Design Description

a. Mechanical Arrangement - Figures V-31, V-32, and V-33 show that the experimental receiver is geometrically and structurally similar to the pilot plant receiver in most respects. Some exceptions, as explained in the previous discussion of scaling criteria are:

- The length of the drum has been shortened in proportion to steaming capacity, while the drum diameter is the same.
- The superheater and boiler tubes are the same diameter as in the pilot plant receiver.
- There are 16 passes, rather than 6, in the superheater.

Other exceptions, 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.

Figure V-31 Side Elevation View - Experimental Receiver

Figure V-32 Plan View - Experimental Receiver

Figure V-33 Arrangement Of Superheater Passes - Experimental Receiver

The arrangement of superheater passes is shown in Figure V-33. There are a total of 16 passes located on six zones of the rear and side walls. These zones resemble the 6 superheater panels of the pilot plant receiver. Superheater tubes are of the same size as those of the pilot receiver, i.e., 25.4 mm (1 in) O.D. by 2.77 mm (0.109 in) minimum wall. They are all on 28.6-mm (1.125-in) centerline except those in passes 3A, 3B and 3C, where tubes are on 31.8-mm (1.25-in) centerline. SA-213 TP304H stainless steel tubes are used in passes 1A through 2C, and SA-213 TP316H tubes are used in the remaining passes. Two 76.2-mm (3-in) O.D. feeder tubes are required to carry saturated steam from the drum to the superheater inlet. All connecting headers and transfer pipes are 114.3 mm (4.5 in) O.D. Two spray attemperators, not shown in the figure, are used in the superheater, one between passes 2C and 3A and the other between passes 4B and 5A.

The superheater tubes in the experimental receiver span the complete width of the heated walls which makes the spans nearly equal to those in the pilot plant receiver, where there are two intermediate supports, and hence three spans, across each wall. The three passes (1A, 1B, and 1C) across the lower right side wall of the experimental receiver will 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, some tubes will also be installed on 31.8-mm (1-1/4-in) spacing 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 2.67-m (8-ft 9-in) square aperture.

The experimental receiver schematic is the same as the one for the pilot plant shown in Figure V-4 except that the superheater has 16 passes instead of 6 and the attemperators are located between the 6th (2C) and 7th (3A) and 11th (4B) and 12th (5A) passes. Further details of the design are given in subsequent sections.

b. Thermal/Hydraulic Design - The procedures used in the thermal/hydraulic design of the experimental receiver are the same as described previously for the pilot plant receiver. The rated heat flux condition, which is shown in Figure V-38 in a later section of this report, was used as the basis for calculating the steady-state tube wall and steam bulk temperatures. The results are shown in Figure V-34. The superheater pressure drop, not including attemperators, is 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

Figure V-34 Superheater Metal And Steam Temperature Profiles - Experimental Receiver

circuitry arrangement depicted in Figure V-35 resembles, again, many features of the pilot receiver boiler circuits. Table V-26 is a material list of this arrangement. Circulation analyses were also performed at the rated heat flux condition, and the results are summarized in Table V-27.

c. Structural Analysis - The structural analysis and design of the experimental unit are similar to those of the pilot plant in many respects. The outer diameters and thicknesses of the superheater and boiler tubes are the same as in the pilot plant. However, their spans are somewhat different. The structural analysis of the different components of the experimental unit are given in the following pages.

• Superheater Analysis - The outer diameter and thickness of the superheater tubes are 25.4 mm (1 in) and 2.8 mm (0.109 in). The finite element model used in the thermal stress analysis is the same as in the pilot plant. The maximum metal temperatures and stresses in various passes are shown in Table V-28. Passes 1A and 5A were selected as the critical passes. It may be noted that in pass 5A the maximum metal temperature is 899 K (1158°F), which is comparable to the maximum metal temperature of 901 K (1162°F) in the pilot plant. The maximum thermal stresses are also comparable.

Tables V-29 and V-30 give the primary and secondary stress calculations and their limits using Section VIII, Division 2. The primary stresses and the S_{mt} limits (Code Case 1592) are shown in Table V-31. Creep and fatigue calculations were not done separately because the stresses and metal temperatures in the experimental unit are similar to those of the pilot plant.

Table V-31 Primary Stress Limits Using Code Case 1592

Material: Pass 1A - 304 SS

Pass 5A - 316 SS

O. D. = 25.4 mm (1 in) t = 3.05 mm (.12 in)

PASS NUMBER	T_{av} K (°F)	P_M MPa (psi)	ALLOWABLE STRESS S_{mt} at 10^5 h MPa (psi)
1A	782 (948)	43.0 (6,233)	97.9 (14,200)
5A	847 (1,113)	43.0 (6,233)	62.7 (9,100)

LEGEND

1. Feedwater Pipe	6. Side Boiler Tubes
2. Steam Drum	7. Rear Boiler Tubes
3. Downcomer	8. Side Screen Tubes
4. Lower Header	9. Corner Screen Tubes
5. Feeder	10. Upper Header
11. Riser	

Figure V-35 Boiler Circuitry Of The Experimental Receiver

Table V-26 Material List Of The Experimental Receiver
Boiler Circuitry

Item No.	Name	No. Req'd	Material	Description
1	Feedwater Inlet Pipe	1	SA-106 C	60.3 mm (2.375") O. D.
2	Steam Drum	1	SA-516 GR 70	1220 mm (48") O. D.
3	Downcomer	6	SA-106 C	114.3 mm (4.5") O. D.
4	Lower Header	3	SA-106 C	168.3 mm (6.625") O. D.
5	Feeder	27	SA-210 A1	63.5 mm (2.5") O. D.
6	Side Wall Boiler Tubes	74	SA-210 A1	38.1 mm (1.5") O. D. by 3.76 mm (0.148") Min. Wall on 44.45 mm (1.75") centerline
7	Rear Wall Boiler Tubes	47	SA-210 A1	"
8	Side Wall Screen Tubes	4	SA-210 A1	"
9	Corner Screen Tubes	6	SA-210 A1	"
10	Upper Header	3	SA-106 C	168.3 mm (6.625") O. D.
11	Riser	31	SA-106 A1	63.5 mm (2.5") O. D.

Table V-27 Boiler Circulation Summary - Experimental Receiver

CIRCUIT I.D.	FLOW RATE kg/h (lb/h)	VELOCITY ENTERING m/s (ft/s)	STEAM EXIT QUAL. % Wt/Vol	STEAM GENERATED kg/h (lb/h)
Rear Wall Panel	63,392 (139,754)	0.75 (2.47)	5.27/39.90	3,341 (7,365)
Rear Wall Downcomer	63,392 (139,754)	1.70 (5.59)	-	-
Side Wall Panel 2 (left)	24,284 (53,537)	0.55 (1.82)	3.83/32.20	930 (2,050)
Side Wall Downcomer	24,284 (53,537)	1.30 (4.28)	-	-
Side Wall Panel 2 (right)	24,644 (54,330)	0.61 (2.01)	3.95/33.82	973 (2,146)
Side Wall Downcomer	24,644 (54,330)	1.33 (4.35)	-	-
Side Wall Panel 1 (left)	9,948 (21,932)	0.41 (1.33)	3.13/27.83	311 (686)
Corner Screen (left)	4,950 (10,912)	0.94 (3.09)	4.61/36.58	288 (503)
Side Wall Downcomer	14,898 (32,844)	0.80 (2.63)	-	-
Side Wall Panel 1 (right)	9,706 (21,398)	0.46 (1.52)	3.30/28.94	320 (706)
Corner Screen (right)	4,895 (10,792)	0.93 (3.06)	4.68/36.95	229 (505)
Side Wall Screen (right)	4,774 (10,525)	0.68 (2.24)	3.96/32.98	189 (417)
Side Wall Downcomer	19,376 (42,715)	1.04 (3.42)	-	-

Total Circulation Rate: 146,594 kg/h (323,180 lb/h)
 Steam Generation Rate: 6,522 kg/h (14,378 lb/h)
 Overall Quality - % by Wt: 4.45
 Overall Circulation Ratio: 22.48

Table V-28 Critical Locations In Various Passes

Material: 316 SS (Passes 3 through 6), 304 SS (Passes 1 & 2)

$h_i = 3.237 \text{ kW/m}^2 \text{ K}$ (570 Btu/h-ft²-°F) - Passes 1, 3 & 6

$h_i = 3.634 \text{ kW/m}^2 \text{ K}$ (640 Btu/h-ft²-°F) - Passes 2, 4 & 5

PASS NUMBER	MAX. HEAT FLUX (q_o)		BULK TEMPERATURE (T_F)		T_{av}		T_{max}		Q + F	
	kW/m ² (Btu/h-ft ²)		K (°F)		K (°F)		K (°F)		MPa	(psi)
1A*	442	(140,000)	603	(625)	782	(948)	814	(1,005)	372.0	(53,950)
1B	426	(135,000)	603	(625)	775	(936)	806	(991)	358.5	(52,000)
1C	410	(130,000)	611	(640)	778	(940)	807	(993)	345.4	(50,100)
1D	347	(110,000)	630	(675)	771	(929)	796	(974)	292.3	(42,400)
2A	378	(120,000)	644	(700)	786	(955)	813	(1,004)	303.4	(44,000)
2B	315	(100,000)	658	(725)	776	(937)	799	(942)	253.0	(36,700)
2C	252	(80,000)	666	(740)	761	(910)	779	(942)	202.0	(29,300)
3A	363	(115,000)	678	(760)	825	(1,025)	851	(1,072)	305.4	(44,300)
3B	347	(110,000)	678	(760)	819	(1,014)	844	(1,059)	292.3	(42,400)
3C	347	(110,000)	689	(780)	830	(1,034)	855	(1,079)	292.3	(42,400)
3D	335	(90,000)	708	(815)	824	(1,023)	844	(1,059)	239.2	(34,700)
4A	354	(95,000)	716	(830)	829	(1,032)	850	(1,070)	239.9	(34,800)
4B	224	(60,000)	739	(870)	809	(997)	824	(1,023)	151.7	(22,000)
5A*	347	(110,000)	744	(880)	874	(1,113)	899	(1,158)	277.9	(40,300)
5B	261	(70,000)	764	(915)	846	(1,064)	881	(1,127)	177.2	(25,700)
6A	198	(53,000)	778	(940)	845	(1,062)	858	(1,084)	140.8	(20,420)
6B	112	(30,000)	783	(950)	821	(1,019)	828	(1,031)	80.0	(11,600)
6C	93	(25,000)	789	(960)	821	(1,018)	826	(1,028)	66.2	(9,600)

*Used for detailed analysis

Table V-29 Superheater Analysis - Pass 1A

Material: 304 SS Pressure = 11.72 MPa (1700 psi) O.D. = 25.4 mm (1 in)
 $t = 3.05 \text{ mm } (.12 \text{ in})$ $q = 442 \text{ kW/m}^2$ (140,000 Btu/h-ft²)
 $T_F = 603 \text{ K } (625^\circ\text{F})$ $T_{\text{max}} = 814 \text{ K } (1005^\circ\text{F})$ $T_{\text{av}} = 782 \text{ K } (948^\circ\text{F})$
 $h_i = 3.237 \text{ kW/m}^2 \text{ K } (570 \text{ Btu/h-ft}^2\text{-}^\circ\text{F})$

CATEGORY	STRESS		ALLOWABLE	
	MPa	(psi)	MPa	(psi)
P_M	43.0	(6,233)	96.5	(14,000)
P_L	43.0	(6,233)	144.8	(21,000)
$P_L + P_B$	43.0	(6,233)	144.8	(21,000)
$P_L + P_B + Q$	238.5	(34,600)	289.6	(42,000)

Table V-30 Superheater Analysis - Pass 5A

MATERIAL: 316 SS, PRESSURE = 11.72 MPa (1700 psi)
OD = 25.4 mm (1 in), $t = 3.05 \text{ mm } (.12 \text{ in})$, $T_F = 744 \text{ K } (800^\circ\text{F})$
 $q = 347 \text{ kW/m}^2$ (110,000 Btu/h-ft²) $T_{\text{max}} = 899 \text{ K } (1158^\circ\text{F})$
 $T_{\text{av}} = 874 \text{ K } (1113^\circ\text{F})$, $h_i = 3.634 \text{ kW/m}^2 \text{ K } (640 \text{ Btu/h-ft}^2\text{-}^\circ\text{F})$

CATEGORY	STRESS		ALLOWABLE	
	MPa	(psi)	MPa	(psi)
P_M	43.0	(6,233)	101.2	(14,680)
P_L	43.0	(6,233)	151.8	(22,020)
$P_L + P_B$	43.0	(6,233)	151.8	(22,020)
$P_L + P_B + Q$	173.8	(25,200)	303.6	(44,040)

Table V-32 gives the maximum horizontal deflections in passes 1A and 5A. The maximum deflections in passes 1A and 5A are 24.2 mm (0.95 in) and 17.6 mm (0.69 in). The corresponding deflections in the pilot plant are 16.0 mm (0.63 in) and 13.2 mm (0.52 in). The larger deflections in the experimental unit are because of the increased span between supports.

Table V-32 Horizontal Deflections Of Superheater Tubes
In Passes 1A & 5A

PASS NUMBER	MAX HEAT FLUX (Q_o) kW/m ² (Btu/h-ft ²)	MOMENT (M_o) N-m (lb-in)	MAX DEFL. mm (in)
1A	422 (140,000)	334.5 (2,961)	24.2 (0.95 in)
5A	347 (110,000)	243.5 (2,155)	17.6 (0.69 in)

• Waterwall (Boiler) Analysis - The outer diameter and thickness of the boiler tubes are identical to those of the pilot plant. Hence the same finite element model was used in the thermal stress analysis. The pressure stresses are identical to those of the pilot plant, and the thermal stresses are lower in proportion to the peak fluxes. Typical temperature and stress distributions are shown in Figures V-36 and V-37.

d. Comparison of Pilot Plant and Experimental Receivers - The important physical data of the two receivers are compared in Table V-33. The heating surface distributions between the boiler and the superheater are almost identical for the two receivers: The weight of the pilot plant receiver is a rough estimate at this time. To compare the responses of the receivers to start-up transients in terms of their respective heat capacities, the dry weight of all pressure parts and the volume of water contained in the boiler are also listed in this table. It can be seen that the weight and water volume of the experimental receiver are greater than would be obtained by direct scale-down in proportion to heat input (or heating surface, which is directly proportional to heat input). This indicates that the experimental receiver will take more time than

Figure V-36 Temperature Distribution In Boiler Tube

Figure V-37 Axial Stress Distribution In Boiler Tube

Table V-33 Comparison Of Physical Data Of Pilot Plant
And Experimental Receivers

	Pilot Plant	Experimental
Total heating surface (proj.), $m^2(ft^2)$	231.3 (2490)	29.3 (315)
Boiler surface (proj.), $m^2(ft^2)$	179.9 (1936)	22.6 (243)
Superheater surface (proj.), $m^2(ft^2)$	51.4 (554)	6.7 (72)
Superheater surface/total surface, %	22.22	22.86
Linear dimension scale factor	1	0.3553
Estimated total receiver weight, 10^3 kg (10^3 lb)	272.2 (600)	61.7 (136)
Estimated dry weight of pressure parts, 10^3 kg (10^3 lb)	77.7 (171.2)	21.0 (46.3)
Estimated volume of water contained m^3 (ft^3)	13.6 (480)	2.8 (99)

the pilot receiver to heat up under the same heat input condition. This effect has to be investigated further so that proper simulation of pilot receiver start-up transients can be obtained in experimental receiver testing.

A comparison of performance data of the two receivers is shown in Table V-34. Design point and maximum load refer to the rated and maximum heat flux conditions.

The design parameters of the two superheaters are compared in Table V-35. As shown in the table, steam mass velocities and heat transfer coefficients of the pilot plant superheater are closely matched by those of the experimental superheater.

A comparison of boiler circulation parameters of the two receivers is given in Table V-36. As indicated, the circulation characteristics of the two boilers are quite different. The cause of these differences is twofold. First, the heat input per tube is lower in the experimental boiler because the same size of tube is used as in the pilot plant boiler and the total number of tubes in the experimental boiler is not scaled down in proportion to the energy input but rather to its square root. Second, the elevation of the drum relative to the entrance of the upflowing circuits is less in the experimental boiler because of the scaling down of linear dimensions. This reduces the available head in the downcomers to motivate natural circulation. The lower entrance velocities in the tubes of the experimental boiler indicate that its design is more critical than that of the pilot plant boiler. Experimental verification that the entrance velocity is adequate in the experimental receiver will automatically prove the same or better for the pilot plant receiver. The lower exit qualities predicted for the experimental boiler, on the other hand, suggest the opposite. 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 in the experimental receiver.

4. Preliminary Test Plan

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
- steady-state operation at the pilot plant receiver peak load
- pilot plant receiver startup and other transients

Table V-34
Comparison of Performance Data of Pilot Plant and
Experimental Receivers

	Pilot Plant		Experimental	
	Design Point	Maximum Load	Design Point	Maximum Load
Thermal duty, MW (10^6 Btu/h)	36.7 (125.26)	49.05 (167.41)	4.63 (15.80)	6.19 (21.13)
Steam flow, kg/h (lb/h)	51,963 (114,556)	69,400 (153,000)	6,559 (14,460)	8,760 (19,312)
Superheater outlet pressure, kPa (psig)	9,136 (1,325)	9,136 (1,325)	9,136 (1,325)	9,136 (1,325)
Superheater outlet temperature, K (°F)	789 (960)	789 (960)	789 (960)	789 (960)
Avg. heat flux densities:				
Boiler surface, 10^3 W/m ² (10^3 Btu/h-ft ²)	146.35 (46.58)	196.40 (62.26)	147.68 (46.82)	197.50 (62.61)
Superheater surface, 10^3 W/m ² (10^3 Btu/h-ft ²)	199.71 (63.31)	266.91 (84.61)	193.56 (61.36)	258.86 (82.06)
Peak heat flux densities:				
Boiler surface, 10^3 W/m ² (10^3 Btu/h-ft ²)	540.03 (171.19)	699.44 (211.72)	540.03 (171.19)	699.44 (211.72)
Superheater surface, 10^3 W/m ² (10^3 Btu/h-ft ²)	451.83 (143.23)	536.28 (170.0)	451.83 (143.23)	536.28 (170.0)

Table V-35 Comparison Of Superheater Design Parameters
Of Pilot Plant And Experimental Receivers

	Pilot Plant	Experimental
Number of passes	6	16
Steam mass velocity, 10^6 kg/h-m^2 (10^6 lb/h-ft^2)		
Maximum	3.19 (0.654)	3.11 (0.637)
Minimum	2.61 (0.535)	2.72 (0.557)
Steam film coefficient $\text{W/m}^2\text{-K}$ ($\text{Btu/h-ft}^2\text{-}^\circ\text{F}$)		
Maximum	3697 (651)	3623 (638)
Minimum	3117 (549)	3231 (569)
Unsupported heating tube span, m (ft)		
Rear wall passes	2.03 (6.67)	2.17 (7.11)
Side wall passes	2.23 (7.33)	2.38 (7.82) 1.19 (3.91) and
Pitch between tubes, mm (in)	28.58 (1.125)	28.58 (1.125) and 31.75 (1.25)

Table V-36 Comparison Of Boiler Circulation Parameters
Of Pilot And Experimental Receivers

	Pilot Plant (Maximum Flux)	Experimental (Rated Flux)
Average height between drum and lower headers m (ft)	19.81 (65)	9.14 (30)
Velocities entering boiler circuits, m/s (ft/s)		
Maximum	1.66 (5.45)	0.94 (3.09)
Minimum	0.99 (3.26)	0.41 (1.33)
Steam exit qualities, % by wt./% by vol.		
Maximum	14.82/65.51	5.27/39.90
Minimum	4.29/32.86	3.13/27.83
Overall Circulation Ratio	10.07	22.48

Steady-state operation at the receiver design point will be simulated by operating the radiant heaters in such a manner as to duplicate the design point heat flux distribution over the heating surfaces. Further steady-state tests will be run at fractions of design point load down to the lowest permissible operating load. Insofar as possible, partial loads will be run with heat flux distributions approximating those predicted for various times of day for the pilot plant receiver.

As discussed previously in Section IV-B, full peak load will not be obtainable with radiant heaters at the Sandia Radiant Heat Facility because of the choice of scale factor and the power limitations of the facility. Peak load testing will be carried out by subjecting selected critical zones of the heating surfaces to peak heat flux densities by biasing the heat flux distribution. This will be discussed further in Section B.5, Test System Requirements.

In the transient tests the radiant heaters will be programmed to effect start-up of the receiver as fast as prior analysis has shown to be permissible under the following conditions:

- cold start
- hot re-start after brief (less than 1-hour) shutdown, simulating passing cloud cover
- hot re-start after overnight shutdown

Normal and emergency shutdown will also be simulated.

Because of the serious and costly damage which will occur if the graphite heaters are contacted with steam or water released by a ruptured tube, all tests will be approached cautiously by gradually increasing the severity of the test conditions while monitoring all sensor outputs to check that all components of the receiver are functioning properly and that no excessive temperatures are being developed at any point.

The following information will be obtained while operating in the test modes:

- Drum pressure and saturation temperature
- Overall heat balance (steady-state only)
- Heat input, total and per channel (zone)
- Absorbed heat flux density at selected points
- Structural deflections and stresses at selected points using linear variable differential transformers (LVDT's) and high temperature strain gages

- Temperatures of pressure parts (tubes, headers, drum) at selected locations using thermocouples,
- Superheater performance, including attemperator control, by measuring superheater steam flow, steam temperatures, and pressures at the outlet of each pass, attemperator spray flow and water temperature,
- Boiler circulation performance by using downcomer static taps and specially designed and calibrated flow meters in downcomers and feeders,
- Steam purity by sampling and analyzing boiler water and steam leaving the drum,
- Feedwater control system response by measuring steam flow, feedwater flow, and drum water level,
- Inelastic strains by measurements taken before and after periods of operation using an indium transfer technique developed at Foster Wheeler.

Regrettably, no creep or fatigue information can be obtained because the scheduled test period is too short.

5. Test System Requirements

Testing the experimental receiver at the Sandia Radiant Heat Facility requires a support system with several functions. A description of this system, including a schematic diagram, has been given previously in Section B.3 of this report. The principal functions and requirements of the support system are discussed in this section.

Since there will be no steam turbine cycle associated with the experimental receiver, the support system must contain provisions for

- Controlling the steam pressure leaving the receiver,
- Condensing the steam generated by the receiver,
- Supplying feedwater to the receiver at the required pressure, temperature, and flow rate,
- Containing the volume of water displaced from the receiver as it is brought from cold condition to maximum output.

To simulate steady-state operation in the pilot plant, the receiver outlet steam pressure must be controlled at $9136 \text{ kPa} \pm 69 \text{ kPa}$ ($1325 \text{ psig} \pm 10 \text{ psi}$) by means of a pressure control valve. Outlet pressure

must also be controlled as the receiver steam pressure is ramped up during start-up.

The steam generated by the receiver must be condensed to reject the heat added to the system and to return the steam to its liquid phase to be pumped back into the receiver as feedwater. An air-cooled condenser will be used since cooling water is scarce at the Sandia test site. The condensate is also sub-cooled in a small water-cooled heat exchanger after leaving the condenser.

A feedwater pump is provided to supply feedwater to the receiver at the required high pressure. After leaving the pump, the high pressure feedwater is heated in a feedwater heater by indirect contact with the superheated steam leaving the receiver. A bypass control on this heater is used to obtain the desired feedwater temperature. Feedwater flow is regulated in response to steam flow and drum water level by a conventional 3-element feedwater regulator as proposed for the pilot plant.

As the receiver is brought from cold condition to maximum operating load, a considerable reduction in water inventory occurs because the specific volume of the contained water increases with temperature and water is also displaced by steam bubbles in the tubes and risers. The reverse occurs when load is reduced and the receiver is shut down. Therefore, adequate condensate storage volume must be provided in the condenser hotwell to accommodate this change in water inventory.

The support system must also maintain the desired quality of feedwater entering the receiver. Since the peak heat flux densities on the boiler tubes are high by conventional steam generator standards, it has been decided to use high-quality feedwater in the receiver. The support system will therefore require a bypass demineralizer and filter and hydrazine and ammonia injection systems on the feedwater circuit. The limits on critical impurities in the feedwater are:

Iron	10 ppb or less
Copper	5 ppb or less
Oxygen	0
Hardness	0

Residual hydrazine, used for oxygen scavenging, should be held between 5 and 20 ppm. Ammonia should be injected to maintain pH in the range 9.2 to 9.4.

Demineralized water must also be supplied for acid cleaning and hydrostatic testing of the receiver prior to initial operation.

A very important function of the test facility is to furnish and control the radiant heat input to the receiver. The radiant heaters

used must have the capability to:

- Supply up to 5 MW total input,
- Duplicate the heat flux distribution of the pilot plant receiver,
- Bias the heat flux distribution for special peak flux tests,
- Simulate heat flux transients.

A typical pilot plant receiver heat flux distribution which must be duplicated by the radiant heaters is shown in Figure V-38. A biased heat flux distribution to be used for obtaining peak fluxes on critical areas of the heating surfaces is given in Figure V-39.

The radiant heating elements will be made of graphite, which must be kept in an inert atmosphere when operating. These elements will be placed opposite the heating surfaces in the cavity, and the space between the elements and the heating surfaces must be kept blanketed with nitrogen furnished by the facility.

Instrumentation and data logging is another important requirement for the tests. The information desired during the tests has already been summarized in the preceding section, Preliminary Test Plan. To obtain this information the following data is to be displayed and recorded on suitable equipment furnished by the test facility:

- Flow, pressure, and temperature of steam leaving receiver,
- Flow, pressure, and temperature of feedwater entering receiver,
- Flow, pressure, and temperature of feedwater entering each attemperator spray,
- Pressure and temperature of steam at outlet of each superheater pass,
- Drum pressure,
- Circulation flow sensor pressure differentials in downcomers, feeders, and inlets of selected tubes,
- Temperatures of pressure parts (tubes, headers, drum) at selected locations,
- Structural deflection readings at selected locations,

Insolation Intensity: 0.825 kW/m^2
 Day: 355 (Winter Solstice)
 Time: 1400

Figure V-38 Heat Flux Profiles to be Duplicated on the Experimental Receiver by Radiant Heaters

INCIDENT HEAT FLUX IN: kW/m^2 ($10^3 \text{Btu/h} - \text{ft}^2$)

- Strain gage readings at selected points,
- Drum water level,
- Heat flux density at selected points on heating surfaces,
- Temperature of outer casing at selected locations,
- Ambient air temperature,
- Electrical conductivity of feedwater and boiler water,
- Electrical power input to heaters (total and per channel),
- Cavity nitrogen blanket pressure.

The estimated number and type of sensors to be used on the receiver are given below.

- 250 thermocouples
- 25 heat flux transducers
- 25 high-temperature strain gages
- 20 linear variable differential transformers
- 30 pressure transducers
- 40 pressure-differential transducers
- 2 water-level transducers

Other important items to be furnished by the test facility are instrument and control air, cooling water, and auxiliary power.

6. Detailed Design, Procurement, and Fabrication Plan

a. Detailed Design - The detailed design of the 5 MWt experimental receiver was begun in October 1975 (see schedule in Figure V-40). The work started with development of general arrangement drawings. These drawings were to be issued in mid-December, but because of revisions resulting from thermal/hydraulic start-up analysis, will not be issued until the end of January. These revisions include separate headers for each superheater pass and the splitting of one side waterwall panel into two panels to accommodate different thermal expansions.

Similarly, pressure part calculations must be revised, and their completion has been rescheduled for the end of January.

The cavity door design has been rescheduled because of current uncertainty about design conditions for the door and for the receiver front wall.

LEGEND

ACTIVITY BAR, ORIGINAL PLANNED SCHEDULE

COMPLETION DATE: PLANNED

ACTUAL

SCHEDULE SLIP: START

COMPLETION

Figure V-40 Detailed Design, Procurement And Fabrication Schedule For Research Experiment Receiver

The start of structural steel design was delayed as a result of continuing discussion on supporting column locations on the pilot plant receiver, since the experimental receiver column locations are to reflect the pilot plant receiver column locations.

Outside vendor equipment handling and shop detailing was begun during December.

Foster Wheeler does not anticipate any delay mentioned above to affect the completion of the detailed design by June 1976. Assignment of additional manpower during the first quarter of 1976 will assure the June completion date.

b. Procurement - Procurement of material for the 5 MWt experimental receiver has begun. The line items are defined as follows:

Safety Valves -	1 superheater and 2 drum outlet valves
Steam Drum Material -	drum shell plate, heads, and nozzles
Superheater Material -	all superheater tubing, headers, transfer lines, spray control headers, and saturated steam connections
Waterwall Material -	all waterwall tubing, headers, downcomers, feeders, and risers
Spray Nozzles -	spray nozzles for spray control headers
Miscellaneous Valves and Gages -	vent and drain valving, steam gage, and water gage
Instrumentation and Controls -	steam temperature controls and valves, feedwater regulating controls and valves
Setting and Insulation -	setting and insulation materials
Enclosure Materials -	roof, side walls, floor and front wall plate materials, and drive mechanism for door
Hot Structural Steel -	hangers, buckstays, and other structural steel material that will be inside the insulation and therefore will undergo significant temperature changes during operation

Cold Structural Steel - columns and bracing between columns external to the insulation, which will be designed by Foster Wheeler, bought in the "as fabricated" condition, and shipped directly to the erection site.

According to the procurement schedule, safety valves which were ordered in September are expected to be shipped by mid-September 1976 to meet Foster Wheeler's schedular requirements. Steam drum materials were ordered in December to enable shop fabrication of the drum to begin in April. The remainder of the pressure parts, superheater, and waterwall materials will be ordered in early January 1976.

c. Fabrication, Shipping, and Erection - The line items of the fabrication schedule are the same as those defined above for the procurement schedule. Those items not listed under fabrication will be purchased as "as fabricated" components and shipped directly to the erection site.

Foster Wheeler expects that all procurement and shop fabrication will be completed on a timely basis so that shipping and erection of the receiver can be completed by November 30, 1976, as scheduled.

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CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM (PHASE I)

SECTION VI

THERMAL STORAGE SUBSYSTEM

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1. INTRODUCTION

The Martin Marietta program team is structured so that program responsibilities are divided among the members of the team, under the overall direction of the Martin Marietta Corporation. Georgia Tech's main area of responsibility is the Thermal Storage Subsystem, including the 10 MWe Pilot Plant Thermal Storage Subsystem preliminary design, the Thermal Storage Research Experiment, and various support tasks related to storage.

The sequence of activities during this reporting period has closely followed that envisioned by ERDA in the Request for Proposal (RFP 75-124) which solicited bids for research and development work on a solar thermal electric power generation system. Initially, a preliminary design was developed for a 10 MWe Solar Thermal Power System by the member organizations of the team; the several subsystems were required to be compatible with one another and the complete plant was required to meet specifications defined by ERDA in the RFP and by the Martin Marietta team in its proposal to ERDA. Next, a series of Research Experiment conceptual designs were developed, one for each major subsystem, which would permit demonstration of the feasibility of the Pilot Plant engineering concepts. The Research Experiment conceptual designs were required to be compatible with the Pilot Plant preliminary designs, and were developed somewhat in parallel with the Pilot Plant designs. Design reviews were conducted to obtain technical approval from ERDA and detailed design of the Research Experiment was begun.

In subsequent reporting periods, the Thermal Storage Research Experiment Detailed Design will be completed, the experiment built and tested, and the test results used to accomplish a second iteration on the Pilot Plant Preliminary Design. The final "product" of the program will be a preliminary design of a 10 MWe Solar Thermal Pilot Plant supported by experience and operating data from the Research Experiments. Georgia Tech's activities will continue to be concentrated on the Thermal Storage Subsystem.

A. PROGRAM OBJECTIVES

The objective of the present program 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 is selected for further development, the next step will be detailed design and construction of a 10 MWe Pilot Plant in order to acquire operating experience and demonstrate the feasibility of generating commercial electric power by solar thermal conversion.

One of the outstanding features of the commercial electric power system in the United States is its extremely high level of reliability. If solar electric power is to achieve acceptance among the commercial

producers of electricity, the plants must be able to continue operation during periods when clouds interrupt the direct solar radiation and during the early evening hours when electrical demands are high but the sun has gone down for the day. Therefore, a method must be developed for storing energy while the sun is shining and releasing that energy, in the form of electricity, when the sun is not shining. One could conceivably store energy as electricity in batteries, kinetic energy in flywheels, chemical energy in the form of hydrogen, or by various other methods. However, when the state of technology in all these fields is considered, the quantity of energy to be stored is considered, and the equipment which will already be available in the power plant is considered, the most reasonable and economical storage approach at this time is the use of thermal energy (heat). Thus, ERDA specified in its RFP that thermal storage capacity would be provided to maintain electrical output of the plant at 70 per cent of its nominal value for six hours.

Once storage in the form of thermal energy had been selected, various technical approaches within this category were possible. The Martin Marietta team proposed to design a storage system using sensible heat in liquid media. There were three important reasons for this choice: (1) storage in liquid media has a high probability of technical success because much of the necessary technology already exists, (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 would delay the entire program, and (3) the cost of stored energy appears to be competitive with energy stored by other methods, especially when the costs of accessory equipment are considered.

The ERDA Solar Thermal Conversion Pilot Plant programs are strongly oriented toward the early use of solar energy for generation of commercial electric power. As a consequence of this emphasis, existing technology is to be applied wherever possible and the practices and preferences of the electric utility industry are to be considered at all times. In order to be commercially acceptable, the Pilot Plant designs must be capable of scale-up into the size range of 100 to 1,000 MWe and the risk of plant failure must be reduced to a level comparable with existing fuel-fired plants. Even though they will not be expected to operate 24 hours per day, the solar plants must generate power on a predictable schedule with high reliability. The development of thermal storage is a direct consequence of these requirements.

Likewise, the Thermal Storage Subsystem must be an integral part of the total plant. Specifically, 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 control system 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.

B. PROGRESS DURING FIRST QUARTERLY REPORTING PERIOD

This report describes technical progress during the first six months of the program. It is planned that subsequent quarterly reports will be issued every three months.

When the program proposals were prepared in April 1975, the contractor teams defined the specifications of their respective systems within the limits that were possible at that time. For example, the Martin Marietta team proposed a Pilot Plant design consisting of a cavity-shaped receiver mounted on a 91.5 Meter (300 ft) to 122 Meters (400 ft) tower and facing a field of heliostats to the north. The receiver was to produce superheated steam at 742 K (950° F) and 8619 kPa (1250 psi). The steam was to be delivered to the Electrical Power Generation Subsystem or to the Thermal Storage Subsystem. The thermal storage concept employed four stages operating at various levels of temperature. Upon initiation of the program in late June 1975, the process of refining the system specifications began; a large number of technical alternatives were considered within the broad framework established by the proposal. For example, a reduction in steam output pressure from storage would reduce the cost of storage by reducing the volume of storage media; however this change also impacted the turbine efficiency adversely and was ultimately abandoned. The thermal storage concept was changed to employ only two stages with a resulting simplification and reduction in cost.

Details of the technical accomplishments during this reporting period are given in the following sections of this report. However, a broad description of the progress is appropriate at this point to illustrate major milestones that have been reached:

1) Pilot Plant Preliminary Design

- Subsystem analyses have been conducted to define Thermal Storage Subsystem requirements and assure compatibility with other subsystems.

- An analytical performance model based on Martin Interactive Thermal Analysis System (MITAS) heat transfer computer program has been developed.
- Several approaches toward system simplification have been considered in detail and the thermal storage concept has evolved from a four-stage to a three-stage and finally a two-stage system.
- Heat exchangers, piping, tanks and other process equipment have been sized and preliminary detailed schematics and ground layout drawings have been prepared.
- Users of the candidate storage fluids have been visited, HITEC has been selected as the higher-temperature fluid and a hydro-carbon oil has been selected as the lower-temperature storage fluid.
- Subsystem specifications have been updated and preliminary reliability allocations established.
- A Preliminary Design Baseline Review has been conducted and ERDA approval received.

2) Research Experiment Conceptual Design

- A Research Experiment has been defined, based on the Pilot Plant preliminary design; the charging and discharging rates were initially established at 5 MWth and the capacity at 4 MW-hr, but these were subsequently reduced to 2 MWth and 1.6 MW-hr respectively.
- A search for a boiler suitable for charging the Research Experiment using superheated steam was conducted; the most advantageous solution was found to be erection of the Research Experiment at a plant owned by the Georgia Power Company.
- Arrangements have been made with the Georgia Power Company to install the Research Experiment at Plant Yates, near Newnan, Georgia.
- An analytical performance model has been developed for the Research Experiment, based on the MITAS computer program.
- Detailed materials lists, schematics and preliminary ground layout drawings have been prepared for the Research Experiment, components sized and vendor price estimates obtained.
- Qualified erection contractors for the Research Experiment have been identified and cost estimates obtained.

---Long lead procurement items have been identified and a procurement schedule established.

---A Conceptual Design Review has been conducted and ERDA approval received; subsequently a second review was conducted and approval received for several proposed changes in the Research Experiment.

---Detailed design of the approved Research Experiment has begun.

---Procurement specifications have been released for heat exchangers and valves and responses received; further negotiations with potential suppliers are underway.

3) Other Program Activities

---A latent heat thermal storage concept has been considered but abandoned for this program because of development time and cost considerations.

---A series of elevated temperature aging tests on candidate oil storage media have been conducted.

---Two alternate thermal storage concepts involving the use of sensible heat in solids have been considered but abandoned for this program because of technical difficulties and cost considerations.

2. THE PILOT PLANT THERMAL STORAGE SUBSYSTEM

The Thermal Storage Subsystem concept calls for storage of energy in the form of sensible heat in liquids. Two basic operating modes are 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 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 known as HITEC will be circulated through the heat exchanger countercurrently to the steam. This process raises the temperature of the HITEC to approximately 728K (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 567K (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. From a conceptual viewpoint, the operations are straightforward; details will be described later in this report.

A. REQUIREMENTS FOR PILOT PLANT THERMAL STORAGE SUBSYSTEM

The Thermal Storage Subsystem has three basic requirements:

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

The system can operate in either a charging or discharging mode, but not in both modes simultaneously. Simultaneous charging and discharging would require two separate steam flow paths and would substantially increase the system cost and complexity. The Thermal Storage Subsystem can effectively perform its function as an energy buffer between the Receiver and Electrical Power Generation Subsystems with one steam flow path in which the direction of flow is reversed when the operating mode is reversed.

In the charging mode, the plant operational requirement is to store all the energy which the Receiver Subsystem is capable of generating

and which is not required for use by the Electrical Power Generation Subsystem. The Thermal Storage Subsystem can accept up to 61,236 kg/hr (135,000 lb/hr) of steam from the receiver at the nominal receiver output conditions (8825 kPa (1280 psig), 784 K (952° F) at tower base). Intermediate steam flow rates can be accepted down to approximately 2268 kg/hr (5000 lb/hr) with small variations in pressure and temperature which result from variations in piping losses. Steam will be vented to the condenser only during a brief interval at the beginning of the day when the receiver flow rate is being brought up to the threshold level required by the thermal storage unit.

In the discharging mode, the plant operational requirement is to supply steam for operation of the turbine from storage only or in combination with steam from the receiver. The Thermal Storage Subsystem can supply up to 57,380 kg/hr (126,500 lb/hr) of steam to the turbine at nominal turbine conditions (4137 kPa (600 psig), 672 K (750° F) at turbine throttle). Intermediate steam flow rates can be supplied down to approximately 2268 kg/hr (5000 lb/hr) for turbine operation and sealing steam can be supplied to the turbine in the saturated condition at 2413 kPa (350 psig) during shutdown periods. Certain lines within the thermal storage unit will be steam traced to prevent freezing of storage liquids, and these will be supplied with steam during shutdown periods up to a few days.

B. PILOT PLANT THERMAL STORAGE SUBSYSTEM BASELINE DESIGN

A system schematic of the Thermal Storage Subsystem baseline design given in the original proposal is shown in Figure VI-1. This schematic illustrates the initial thermal storage concept, which has subsequently been changed in detail but not in principle. The diagrams emphasize the charging mode in the upper part of the figure and the discharging mode in the lower part.

The storage medium in each stage (a molten salt mixture or a hydrocarbon oil) is pumped from a reservoir, through a heat exchanger, to a second reservoir; in this process heat is either charged into the medium or extracted from the medium, depending on whether the system is operating in the charging or discharging mode. The quantity of fluid in each reservoir changes as fluid is moved from one to another. The respective design temperatures are shown in the diagram. This system requires a tank capacity equal to twice the volume of fluid in each stage since the system must be capable of holding all the fluid of a stage in either the cooler tank or the warmer tank, depending on the condition of charge. Warmer and cooler fluids in a particular stage must not be allowed to mix because this reduces the temperature at which the stored energy is available for producing output steam.

HITEC is an eutectic salt mixture in the system $\text{NaNO}_2\text{-NaNO}_3\text{-KNO}_3$ which has a melting point of 288° F. It is used in industry as a heat transport medium and is capable of operation at temperatures up to 755 K (900° F). It is the most feasible material yet identified for

Figure VI-1. System Schematic, Proposal Baseline Thermal Storage Subsystem.

the high temperature stage(s) of a sensible heat thermal storage system. Hydrocarbon oils appear to be suitable for use as storage media at temperatures up to about 588 K (600° F) and possess technical advantages compared with HITEC when the proposed use temperature is within the above limit. Several candidate hydrocarbon oils have been considered for use as storage media, as have the Therminol products manufactured by Monsanto.

Two second-stage heat exchangers were planned in the baseline thermal storage system design; one of these served as a desuperheater during charging and the other served as an evaporator during discharging. This entire stage was subsequently eliminated because its function can be incorporated into the condensing-evaporating stage. Also, the storage medium flow path in the fourth stage was radically changed in order to simplify the system.

A temperature-enthalpy diagram representing the proposal baseline Thermal Storage Subsystem concept is shown in Figure VI-2. The uppermost line represents the temperature potential in 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 temperature-enthalpy line and supply energy to steam represented by the lower temperature-enthalpy line. Thus, a method must be found to extract energy when required and feed it back when required at the proper temperature levels.

The proposal baseline Thermal Storage Subsystem envisioned the use of four heat exchanger stages in which liquid thermal storage media would be circulated countercurrently against the steam and/or water.

During charging, the first two stages of the proposal baseline Thermal Storage Subsystem perform as desuperheaters and raise the temperature of a molten salt storage medium known as HITEC. The third stage condenses steam to heat a hydrocarbon storage medium and the fourth stage subcools water to heat a hydrocarbon medium. The storage media were chosen on the basis of minimum cost of stored heat in their respective ranges of operating temperature.

During discharging, the direction of water-steam flow is reversed and the fourth stage preheats the entering water, the third and second stages evaporate the water to produce saturated steam, and the first stage superheats the steam to the required outlet conditions. The diagonal dashed lines on the diagram represent operating temperatures of the storage media.

The detailed schematic of the proposal baseline Thermal Storage Subsystem, shown in Figure VI-3, illustrates the scope of the equipment

Figure VI-2. Temperature-Enthalpy Diagram for Proposal Baseline Thermal Storage Subsystem...

required more accurately than the conceptual schematic given earlier. Each of the four stages incorporated multiple tanks, with one empty tank. The fluid in a full tank was pumped through a heat exchanger into the empty tank. The tank that had just been emptied then served as the 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 was incorporated in each stage in order to extend the life of the thermal storage fluids.

During charging, the pressure in the thermal storage subsystem was controlled by a back pressure valve located downstream of the subcooling heat exchanger (HE4). The pressure sensor for this control function was also located downstream from the subcooling heat exchanger and upstream from the control valve. The exit temperature from the TSS back into the central receiver feedwater line was controlled by heat exchanger HE4. The temperature sensor for this control was located downstream from HE4. The flow rate into the TSS was regulated by a throttling valve located upstream from the desuperheating heat exchanger (HE1). The flow rate into the system was adjusted to maintain a constant receiver supply pressure.

During discharging, the flow rate into the TSS was controlled by the thermal storage feedwater pump. The flow rate was adjusted to maintain a constant turbine throttle pressure. The liquid level in the evaporating heat exchanger (HE3) was held constant by adjusting the flow rate of hot hydrocarbon oil through the heat exchanger.

C. REDEFINITION OF THERMAL STORAGE SUBSYSTEM

The proposal baseline design of the Thermal Storage Subsystem is a functionally correct approach to the storage of heat as sensible heat in liquids. The design is however, relatively complicated. A number of thermal storage subsystems have been studied in an effort to simplify the proposal baseline design and develop a more cost effective system. The studies have successfully achieved this goal.

The first alternate design that was studied used the same heat exchanger arrangement that was used in the proposal baseline design. In this alternate design, a single flow of heat transfer salt is circulated through the stage one and stage two heat exchangers in series. The quantity of heat transfer salt required to operate both stages in series is approximately equal to the quantity required by the proposal baseline design stage two alone. The separate oil inventory for the subcooling/preheating fourth stage of the proposal baseline design has also been eliminated. In this alternate design, the low temperature reservoir of the original third stage serves as the high temperature reservoir for the subcooling/preheating stage. The schematics for this modified system are shown in Figures VI-4 and VI-5. Figure VI-4 shows the system configuration when heat is being stored in the thermal storage subsystem and Figure VI-5 shows the system configuration when steam is being produced by extracting heat from the thermal storage subsystem.

LEGEND

LLS	LIQUID LEVEL SENSOR
TS	TEMPERATURE SENSOR
P	PUMP
PR	PRESSURE REGULATOR
TV	THROTTLE VALVE

The controls for the modified four stage system are simpler than the controls for the proposal baseline system. During charging, the heat transfer salt flow through the desuperheating heat exchangers is controlled by a single temperature sensor downstream of the second heat exchanger. During discharging, the salt flow is controlled by a temperature sensor located in the thermal storage outlet line.

The principle advantage of this system over the proposal baseline design is that both the salt and the oil inventories have been reduced. The controls of the heat transfer salt stages have also been simplified. But this modified system still employs five heat exchangers and a complicated high pressure valving system is still required to control the flow of steam and feedwater through the low temperature heat transfer salt heat exchangers.

The second stage of the proposal baseline design was eliminated in the next system that was considered. The schematics for this system, a three stage system, are found in Figures VI-6 and VI-7. Figure VI-6 shows the system configuration when heat is being stored in the Thermal Storage Subsystem and Figure VI-7 shows the system configuration when steam is being produced by extracting heat from storage. In this system, the final desuperheating of the incoming steam is accomplished in the condensing stage during charging. The desuperheating allows sufficient heat to be stored in the condensing stage to accomplish the complete evaporation of the feedwater during discharge. The high pressure flow control circuit has been eliminated. In addition, the overall heat storage media inventory has been reduced. The heat transfer salt inventory of the second stage of the proposal baseline design has been replaced by the larger temperature difference of the oil in the condensing/evaporating stage. The high temperature of the oil reservoir can now be raised to 572 K (579⁰ F) by the superheated steam entering the condensing stage.

The controls for the three stages that are retained in this new design are the same controls that were used in the corresponding stages of the proposal baseline design. The flow of heat transfer salt is regulated to maintain a constant steam temperature out of this stage during both charging and discharging. The flow of oil in the subcooling/preheating stage is also regulated to maintain a constant water temperature out of this stage during both charging and discharging. The oil flow in the condensing/evaporating stage is regulated to maintain a constant temperature of the condensate leaving during charging and a constant liquid level during discharge.

A problem area in this three stage system is the oil film temperature in the condensor during charging. Film temperatures are a particular concern during periods of low flow when the film thickness increases. The 636 K (686⁰ F) superheated steam entering the condensing heat exchanger will accelerate the degradation of the oil and can produce coking of the oil in the heat exchanger during periods of low flow. This problem has been overcome in the current Thermal Storage Subsystem design.

Figure VI-6. Three Stage Thermal Storage Subsystem-
System Configuration During Heat Storage.

Figure VI-7. Three Stage Thermal Storage Subsystem-
System Configuration During Heat Extraction.

D. TWO STAGE THERMAL STORAGE SUBSYSTEM

The current Thermal Storage Subsystem two-stage design is similar to the three stage system just described. The system schematics for the current design are found in Figures VI-8 and VI-9. Figure VI-8 shows the system configuration when heat is being stored in the thermal storage subsystem and Figure VI-9 shows the system configuration when steam is being generated by extracting heat from storage. There are two differences between the flow circuit in the current design and the flow circuit in the three stage design. During charging, the partially desuperheated steam exiting the heat transfer salt stage is further desuperheated in a spray desuperheater before entering the condensing/evaporating heat exchanger. The second change is in the oil flow circuit. In the current two stage design, all of the oil flows through the condensing/evaporating heat exchanger and the subcooling/preheating heat exchanger in series.

The controls of the current two stage design differ from the controls of the three stage design in a number of instances. During charging, superheated steam from the receiver enters the heat transfer salt heat exchanger where some of the superheat energy is transferred to the salt storage medium. The flow in this exchanger must be counter-current to accommodate the large temperature cross in this stage. The salt flow rate is again controlled by a temperature sensor operating on the outlet steam from the exchanger. Next, the steam enters a spray desuperheater where most of the remaining superheat is removed by dilution with subcooled water. This process eliminates excessively high oil film temperatures and prevents oil coking in the condensing heat exchanger. The water flow rate into the desuperheater is controlled by a temperature sensor operating in the desuperheater outlet steam. The desuperheating operation does not change the overall heat balance of the condensing stage because the decrease in steam temperature is accompanied by an increase in flow rate.

The charging steam is condensed by transferring its latent heat to a hydrocarbon oil storage medium. The oil flow rate is regulated by a pressure sensor that senses the condensing pressure. As such the condensing rate is made to match the incoming steam rate. The water level in the condensing exchanger is regulated by a level sensor controlled water outflow valve. The liquid level in the condenser will be maintained such that the outflowing water will be slightly subcooled. The condensed water passes finally to the subcooling exchanger where further heat is removed. The water exiting from the subcooling heat exchanger will be subcooled sufficiently to prevent flashing when it enters the central receiver feedwater system. The subcooling heat exchanger will be sized so that the maximum water temperature leaving the thermal storage subsystem will be 539 K (510° F).

During discharge (Figure VI-9), the feedwater from the electrical power generation subsystem enters the preheating heat exchanger and

Note: Table Condensate Temperature vs Flowrate

Flowrate		Temperature	
Kg/Hr	Lb/Hr	K	OF
57,380	(126,500)	488	(420)
45,360	(100,000)	480	(405)
34,020	(75,000)	466	(380)
22,680	(50,000)	444	(340)

LEGEND	
DSH	DESUPERHEATER
FRS	FLOW RATE SENSOR
LLS	LIQUID LEVEL SENSOR
TS	TEMPERATURE SENSOR
P	PUMP
PR	PRESSURE REGULATOR
PS	PRESSURE SENSOR
TV	THROTTLE VALVE

Figure VI-8. Two Stage Thermal Storage Subsystem-
System Configuration During Heat Storage.

Figure VI-9. Two Stage Thermal Storage Subsystem-
System Configuration During Heat Extraction.

is heated to a temperature just below the saturation level before being introduced into the evaporator. In this mode of operation, the water level in the evaporator shell must be maintained at a position above the tubes. A three-element feedwater regulator is used for the purpose in a manner analogous to that employed in conventional fossil fuel fired boiler. The oil flow rate in the evaporating stage is controlled by a pressure sensor near the turbine throttle. This is also analogous to regulating in a fossil fuel boiler to control the turbine throttle pressure.

Steam leaves the evaporator in the saturated condition at a pressure sufficient to deliver 4137 kPa (600 psig) steam at the turbine throttle and enters the superheating heat exchanger. The spray desuperheater is not used during discharge of storage and the steam merely passes through it. In the heat transfer salt stage, heat is extracted from the molten salt to raise the steam temperature to approximately 672 K (750° F). The salt flow rate is controlled by a temperature sensor in the output steam line.

Figure VI-10 shows a heat balance for the two stage thermal storage subsystem. This diagram resembles a temperature vs enthalpy diagram. In fact, the graph for the discharging steam and the oil and heat transfer salt lines do represent true 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 pound 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. This ratio is, however, very close to 1.0, and it is safe to neglect this difference unless a very accurate heat balance is required.

In the two stage system, all of the oil flows through the subcooling heat exchanger. This means that the slope of the oil line on the heat balance diagram is the same in both the subcooling/preheating section and the evaporating/condensing section. The charging steam can be cooled to just under 533 K (500° F) in the two stage system. Since less heat is extracted from each pound of charging steam, the flow rate of charging steam must be increased slightly to store heat at the same rate. In this system, the heat from 1.034 Kg of charging steam has to be stored for each Kg of discharge steam. In the proposal baseline design, the mass ratio of charging to discharging steam was exactly 1.000. The heat balance diagram for that case was also a true temperature vs enthalpy diagram. It should be observed that the heat which is not extracted from the charging steam is returned to the central receiver in the feedwater. No increase in heat input into the boiler is required by the two stage system. Only the mass flow rate of charging steam/water must be increased slightly to transport heat at the same rate.

Figure VI-10. Heat Balance for Thermal Storage Subsystem

The changes that have been made in the proposal baseline design to produce the current two stage design can be summarized as follows:

- 1) Stage two of the baseline system has been eliminated by combining it with stage three and adding a desuperheater,
- 2) The high temperature reservoir of the subcooling stage has been eliminated.

Together, these changes result in the following advantages:

- 1) A heat transfer salt stage and associated hardware have been eliminated.
- 2) The possible temperature difference of the oil stage has been increased.
- 3) A high pressure flow control circuit has been eliminated.
- 4) The overall heat storage media inventory has been reduced.

All of these systems simplifications have been achieved with no sacrifice in system performance.

3. PILOT PLANT PERFORMANCE STUDIES

A. MITAS Computer Studies

General Description of MITAS Operation

The partial differential equations describing the transient flow of heat and fluids in the thermal storage subsystem are solved using finite difference routines contained in MITAS. MITAS (Martin Marietta Interactive Thermal Analysis System) is a multipurpose computing program installed on the CDC Cyber 74 computer at the Georgia Institute of Technology.

A physical description of heat flow in the thermal storage subsystem is entered into MITAS by establishing the boundary conditions and by constructing a thermal analog network of the system. Individual capacitances in this network are sized according to the thermal storage capabilities of quantized, elemental sections of the heat exchangers and the storage tanks. Conductances in the network define the flow of thermal energy by conduction, convection, and radiation between the elemental sections and the ambient environment. Differential fluid flow expressions are introduced into MITAS via logic portions of the program that are an integral part of the solution of the thermal analog network. These logic sections are also utilized in simulating the action of the operating controls which stabilize the overall storage subsystem during transient charge or discharge conditions.

Description of the Thermal Network Analog of the Storage System

Each major component of the heat exchangers and storage tanks in the storage subsystem is partitioned into elemental volumes that are mathematically identified as nodes. Nodes associated with a thermal mass and capable of exchanging thermal energy with adjacent volumes are connected to lumped parameter capacitances and conductances, respectively. Boundary conditions and control points are entered in the network as nodes which have no capacity for storing thermal energy. Figure VI-11 shows a nodal network analogous to the first heat exchanger in the thermal storage subsystem. This stage operates as a desuperheater in the charge condition. The capacitances for diffusion nodes in the network are computed using the mass and specific heat for the elemental volume of structure or fluid.

Conventional conduction, convection, or radiation equations are employed to compute conductances in the network. Local heat transfer coefficients are used to calculate the effective conductance between fluid and heat exchanger nodes. These coefficients are recomputed at each time increment in the solution to incorporate current Stanton numbers and fluid properties. Conductors connecting fluid nodes are unilateral in the sense that heat can only be transferred in the downstream direction. Physical properties such as the specific heat, density, thermal conductivity, and viscosity of the thermal salt and charging steam are tabulated in arrays as a function of temperature. The program interpolates in these arrays to arrive at values corresponding to the volume temperature at each incremental time step.

Figure VI-11. Analog Thermal Model for HX1 (HITEC Stage-Charging).

The network model for the last stage in the storage subsystem closely resembles the model shown in Figure VI-11. Material properties in this stage reflect the different physical characteristics of the lower temperature thermal fluid and water flowing in the subcooler.

Successful modeling of steam condensation and water evaporation requires a finer subdivision of the nodal network for the second stage condenser/evaporator. The number of nodes and lumped parameter elements for this stage are increased by approximately 30 percent over those shown in Figure VI-11. Fluid nodes experiencing a change of phase are treated as boundary nodes. Properties of elements attached to these nodes are determined by the condensation or evaporation occurring in the fluid control volumes during the differential computation interval.

The individual heat exchanger models are linked together to form a total operating system. Fluid flow corrections are derived from the temperature, pressure and liquid level at designated sensor nodes. These data form the input to the program logic which models the operating controls of the thermal storage subsystem.

During charging the storage fluid flow rates are controlled by the steam or water temperatures at the heat exchanger outlets. Outlet steam temperature is also sensed to determine the saturated water flow rate entering the mixing valves preceeding the desuperheater and the condenser. Water flowing from the condenser is controlled as a function of the condenser pressure and liquid water level.

In the discharge mode of operation, the evaporator steam pressure controls the flow of thermal fluid through the evaporator. Outlet steam temperatures are monitored to establish the flow rate of the thermal salt in the superheater.

Results of MITAS Analysis of the Pilot Plant Storage Subsystem

Transient operation of the thermal storage subsystem is of particular interest. Realistic field operating conditions and schedules are being simulated on the MITAS computer model to study the performance of the heat exchangers and the effectiveness of the control system in maintaining system stability. This analysis includes initial charging response as well as operation following a changeover from charge to discharge, charge to standby, discharge to standby, and discharge to charge.

Examples of MITAS simulations of the pilot plant model during charging from a standby condition are presented in Figures VI-12, VI-13, and VI-14. A step input of charging steam flows at the nominal rate of 51,963 Kg/hr (114,556 lb/hr). The steam enters the subsystem at 783 K (950°F) and is condensed in the second stage at a pressure of 8619 kPa (1250 psig) and a temperature of 574 K (574°F). Initial heat exchanger temperatures are set in the range of 422 K (300°F) to 478 K (400°F).

Figure VI-12. Charge Cycle Water Outlet Temperature vs Time for Nominal Steam Flow.

Figure VI-13. Charge Cycle Thermal Fluid Flow Rates vs Time for Nominal Steam Flow.

Figure VI-14. Charge Cycle Thermal Fluid Outlet Temperature vs Time for Nominal Steam Flow.

The outlet steam and water conditions from the heat exchangers are controlled by a feedback system using proportional control which continuously corrects the flow rates of the thermal fluids.

Figure VI-12 indicates that the major transients in the outlet water temperatures for the above initial conditions have decayed within five to six minutes. The control temperatures are 634 K (683°F), 588 K (600°F) and 525 K (486°F) for the desuperheater, desuperheater mixer, and subcooler, respectively. The oscillatory behavior at the condenser outlet reflects the model's sensitivity to changes in the liquid level at the bottom of the condenser tubes.

Flow rates and outlet temperatures for the thermal fluid are shown in Figures VI-13 and VI-14, respectively. In this particular model, only a portion of the thermal fluid flowing through the condenser flows through the subcooler. Figure VI-13 indicates that constants for the control system of the subcooler have been selected to produce a slightly underdamped response in the thermal fluid flowing through it. The dip in the thermal fluid flow rate for the condenser reveals the propagation of variations in steam temperature at the outlet of the desuperheater. More recent results presented on page VI-62 indicate that these oscillations in the desuperheater valve can be reduced by using proportional control plus reset. In each exchanger, a steady state flow of thermal fluid is obtained within approximately 10 minutes.

Outlet thermal fluid temperatures plotted in Figure VI-14 also reach steady state values within approximately 10 minutes. Again the initial oscillations in the thermal salt flow rates for the desuperheater during the first 10 minutes perturb the outlet temperature of the salt as it increases to a steady state value of 727 K (850°F). Inlet, thermal fluid node temperatures for the desuperheater, condenser, and subcooler are 583 K (590°F), 538 K (510°F) and 518 K (473°F), respectively.

Other computer simulations are evaluating the ability of the system to handle inputs such as a premature valving failure, receipt of variable steam flow rates from the receiver, and following turbine loads in a demand mode. Two examples of charging operations using steam flow rates which deviate from nominal rates are plotted in Figures VI-15 and VI-16. Thermal fluid flow rates for a step input of steam at the maximum receiver output flow rate of 61,236 Kg/hr (135,000 lb/hr) are given in Figure VI-15. Other steam conditions, heat exchanger temperatures, and control system constants remain unchanged from those discussed above. The initial fluid flows increase faster than the flow rates shown in VI-13. Final steady state rates are higher with the increased quantity of steam flowing in the storage subsystem. Otherwise the response curves are similar to the nominal flow curves. Since thermal fluid flow oscillations and exit steam temperature variations damp out more quickly in the desuperheater at the higher steam flow rate, the flow pattern for the condenser thermal fluid becomes smoother.

Figure VI-15. Charge Cycle Thermal Fluid Flow Rates vs Time for Maximum Steam Flow.

Figure VI-16. Charge Cycle Thermal Fluid Flow Rates vs Time for Low Steam Flow.

Charging the pilot plant at a low steam flow rate of 17,350 lb/hr (38,250 lb/hr) is simulated in Figure VI-16. Other operating conditions for the storage subsystem were unchanged. With respect to Figure VI-13, the lower steam flow rate lengthens the rise times for the thermal fluid response curves to 10 to 14 minutes. Rise times of approximately five minutes are observed for nominal steam flow rates. The model also predicts an increased tendency toward oscillation about the lower steady state thermal fluid flow rates as the charging steam flow is reduced.

4. CODE CONSIDERATIONS

The applicable design codes and regulations and their respective areas of application as currently defined for the design of the pilot plant are presented in Table VI-1.

There are three areas which are notable and perhaps require additional comment. These are:

- ** The first area concerns the design philosophy of the overall system with regard to the physical layout that will be required if the rules set forth in NFPA No. 30-73, the Uniform Building Code, and the Occupational Safety and Health Standards, are rigidly interpreted to apply to all geographic locations. The land space requirements for the Pilot Plant Thermal Storage System will increase over the preliminary layout as will the costs for the system. The principal determining factor which dictates component and system proximities is the application of the provisions whereby a Class IIIB fluid (flash point above 200°F) is stored or handled above its flash point. The interpretation that a Class IIIB liquid above its flash point must be treated as a Class II liquid is the detail consideration involved. This area will require further study to determine the appropriateness of this requirement for a plant located in a desert area far removed from the "normal" construction context and for a system utilizing nitrogen inerting as a cover over the heated fluid.
- ** The second area relates to the design of all components operating at temperatures up to 800°F. The rules of ASME Section VIII, Division 2, are considered to be applicable to the components here with possibly the rules of B31.3 covering the interconnecting piping for the oil and salt. However, there are areas where the design rules for the pilot plant should encompass more stringent analytical treatment for appropriate comparison to the results from these existing more simplistic codes. Appropriate to this, it is believed, are the analytical rules of Section III, Division 1, Subsection UB of the ASME Code. Accordingly, the design of the pilot plant anticipates its use for these components for comparison to the results obtained using Section VIII, Division 2.
- ** The third area encompasses the design of components operating above 800°F. The only existing choice where explicit criteria exist, is to utilize the rules of the special Code Case 1592 of Section III of the ASME Code, so that proper consideration of fatigue and creep is included.

The above observations are not intended to rigidly endorse the total application of Section III of the ASME Code, but instead, recognizes

TABLE VI-1

SUMMARY OF DOCUMENTS AND DESIGN COVERAGE AREAS APPLICABLE TO THE
DESIGN OF THE PILOT PLANT THERMAL STORAGE SUBSYSTEM

Document	Coverage
(1) American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (1974)	
SECTION II Material Specifications	All material specifications for materials used in the construction of the pressure equipment and heat storage vessels and all materials used as welding rods, electrodes and filler materials.
SECTION III Nuclear Power Plant Components, Division 1	All components operating at temperatures above 427° C (800° F).
SECTION V Nondestructive Examination	All nondestructive examination methods specified and utilized on all equipment and components.
SECTION VIII Pressure Vessels-Division 2	All components operating at temperatures up to 427° C (800° F).
SECTION IX Welding & Brazing Qualifications	Shall be specified for all welding and brazing.
(2) American National Standards Institute	
ANSI A58.1-1972 Building Code Requirements for Minimum Design Loads in Buildings and other Structures	All wind loading and earthquake loading design.

(Continued)

TABLE VI-1 (Continued)

SUMMARY OF DOCUMENTS AND DESIGN COVERAGE AREAS APPLICABLE TO THE
DESIGN OF THE PILOT PLANT THERMAL STORAGE SUBSYSTEM

Document	Coverage
(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 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 the American Institute of Steel Construction and American Concrete Institute	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.

the need to selectively apply certain of the rules for the design of the pilot plant. The apparent conclusion at this time is that either additions to the existing codes are necessary for the design coverage for solar plants or, there needs to be generated as a separate code section, a specific code for the design of solar plants. This should be addressed by the appropriate ASME committees.

5. ECONOMIC AND FEASIBILITY TRADE STUDIES

During the early stages of the program, a number of methods of storing heat at high temperatures were investigated. These methods can be grouped into two categories. The first category includes systems that store heat in the form of sensible heat in either solids or liquids. The second category includes systems that store heat as the latent heat of fusion of solids. A liquid sensible heat system was chosen for the baseline design and the operation of this system has already been described. The principles of operation of a liquid sensible heat system will again be outlined in this section for comparison with other systems that have been and are being studied. The cost and developmental effort required to produce an operating system will also be assessed and compared with the liquid sensible heat system.

The principle of operation of a liquid sensible heat system is outlined in Figure VI-17. When heat is being stored in the system (charging operation) liquid is pumped from a low temperature reservoir into a heat exchanger. The fluid is heated in the heat exchanger by cooling or condensing a gas or liquid 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 different heat exchanger where it heats or evaporates a cool liquid or gas. The fluid 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 are used in the proposal baseline design 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.

The cost of the major components of the hydrocarbon oil stage of the current two stage liquid sensible heat system for a 10 Mw electrical facility have been estimated and are found in Table VI-2. This facility is capable of a 10 Mw electrical output for six hours. The oil stores a total of 166.0×10^6 Watt hours. Ten tanks 9.14 meters in diameter and 12.19 meters tall are required to hold the oil (this number includes one extra tank to accept flow). The oil cost is based on a price of 264 dollars per cubic meter. These cost figures will be used for comparison with estimated cost of the other heat storage concepts that have been studied.

A. SOLID SENSIBLE HEAT STORAGE SYSTEMS

The use of an inexpensive solid storage medium such as rocks or scrap metal has been studied. The concept of operation of these solid sensible heat systems will first be described and then the details of the studies will be presented. The principle of operation of the solid sensible heat system is the same for both rocks and scrap metal. In either case, the solid storage medium would be contained in a tank. A liquid heat transport medium, such as a hydrocarbon oil would be cir-

Figure VI-17. Operation of a Sensible Heat Fluid Storage Subsystem.

TABLE VI-2

COST COMPARISON OF THREE HEAT STORAGE CONCEPTS 1/

Storage Method	Sensible Heat		Heat of Fusion	
	Oil System	Rock System	92.2 J/g Salt <u>2/</u>	372 J/g Salt <u>2/</u>
Tanks	$\$0.6 \times 10^6$	$\$0.90 \times 10^6$	$\$0.30 \times 10^6$	$\$0.12 \times 10^6$
Oil	$\$1.5 \times 10^6$	$\$0.29 \times 10^6$	$\$0.10 \times 10^6$	$\$0.10 \times 10^6$
Rocks		$\$0.21 \times 10^6$		
Tubes			$\$1.90 \times 10^6$	$\$0.58 \times 10^6$
Total	$\$2.1 \times 10^6$	$\$1.4 \times 10^6$		
Subtotal			$\$2.3 \times 10^6$	$\$0.80 \times 10^6$
Less cost of salt				
Price of Salt to be competitive with oil system				$\$1.06/\text{Kg}$

1/ Total heat stored in each system equal to 166.0×10^6 watt-hrs.

2/ See text for explanation.

culated in a loop through the tank and through a heat exchanger. Figure VI-18 contains a conceptual schematic of the process with rocks in the tank. When heat is being stored in the media (charging operation), 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 this 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. A thermal gradient with this characteristic is shown in Figure VI-18.

For a storage system of this kind to be practical, the thermal gradient must be relatively sharp, almost all of the solid media undergoes a 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 large portion of the solid media will undergo only a small temperature change, and the volumetric efficiency will be low. Since the tanks represent an important part of the system cost, high volumetric efficiency is necessary for a solid sensible heat system to be economically competitive with a liquid sensible heat system.

The character of the thermal gradient depends on the thermal parameters of the heat transfer fluid, the thermal parameters of the solid media and the size, shape and packing of the solid media. MITAS, the thermal analysis program described on pages VI-24 through VI-26 of 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. The details of this model will be discussed in the section on low density solid storage systems.

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 media 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.

B. HIGH DENSITY MEDIA STUDY

A metal, or metal alloy could be used as the solid in the heat storage system just described. Metals and their alloys are relatively dense, so if a cheap alloy with a high specific heat were available, it could be used to produce a compact, economical heat storage system. But the specific heats of the common metals are not large compared with the specific heats of liquids and the product of the specific

Figure VI-18. Operation of a Sensible Heat Solid Storage System.

heat and density usually has about the same value for both metals and liquids. Table VI-3 list the pertinent thermal and cost data for the most likely metal candidates along with the same data for a hydrocarbon oil and for heat transfer salt. This table illustrates that even the least expensive scrap metals at current prices cannot compete economically with oils in a thermal storage application. The table also shows that iron alloys may be competitive with molten salt in the same storage application. It should be noted that the pumping costs and piping costs of the solid storage system will probably be greater than the corresponding costs for a liquid system. This cost increase may eliminate the apparent advantage of scrap metals over molten salts in a heat storage application. Because a sensible heat oil system can be built more economically than the cheapest solid metal system. Metals are no longer active candidates for the heat storage media for this project.

C. LOW DENSITY MEDIA STUDY

The more common minerals have a lower density than metals, but some of them are much less expensive. For instance, crushed granite that has been sized and washed sells for around \$.0033 per Kg (\$3.00 per ton) in many areas of the country. This low price makes rocks attractive candidates for solid heat storage media. To see if rocks could be used as the storage media in a system that was compatible with the requirements of an electricity generating plant, a computer model of a rock bed was written. The model was general in form and permitted the evaluation of rock beds with different physical and thermal parameters. A densely packed bed of spheres of a single diameter was assumed.

The results of the computer study indicate that small rocks with diameters on the order of one centimeter can be used in a power plant heat storage system. The thermal parameters of granite were used for this model, but the thermal parameters of most common minerals used in construction are similar to those of granite.

The tanks were assumed to be 9.14 meter in diameter and the total heat stored was 166.0×10^6 Watt hours. These numbers correspond to the numbers used in oil system cost study summarized in Table VI-2. With these assumptions, it was found that 15 tanks 12.19 meters high connected in series were required to hold the rocks. The thermal gradient was approximately 9 meters long when the fluid velocity through the tank was 50 meters per hour. The thermal gradient length for this case was a small fraction of the total effective tank length.

Table VI-2 summarizes the estimated cost of a heat storage system using rocks. The rock cost is based on \$.0110 per Kg (\$10 per ton) instead of the \$.0033 per Kg (\$3 a ton) quoted earlier to allow for additional rock handling and cleaning before they are used in the storage system. The rock system is less expensive than the oil system. These costs do not however, include the cost of pumping

TABLE VI-3

ECONOMIC ANALYSIS OF HIGH DENSITY STORAGE MEDIA

Material	Density		Heat Capacity	Heat Conductance		Cost	Cost per [*] 10 ⁵ Joules	Volume of Storage ^{**} Required per 10 ⁶ Joules
	(g/cm ³)	(lb/ft ³)	(J/g K)	(J/m sec K) (Btu/ft hr °F)				(m ³)
Iron Alloys	7.8	(487)	0.279	2.556	(26.)	0.11	\$6.58	0.027
Lead Alloys	10.5	(655)	0.070	1.887	(19.0)	0.33	7.90	0.080
Copper Alloys	8.5	(530.)	0.209	5.923	(60.)	1.32	105.20	0.033
Nickel Alloys	8.5	(530)	0.240	3.344	(34)	4.40	306.13	0.029
Zinc Alloys	7.1	(443)	0.212	6.329	(64)	0.55	43.40	0.039
HITEC	1.9	(117)	0.868	7.881	(0.33)	0.46	8.89	0.025
Thermonol 55	0.72	(45)	1.628	1.576	(0.07)	0.46	4.74	0.035
Therina 33	0.72	(45)	1.512	1.576	(0.07)	0.37	4.33	0.038

* Calculation assumes a temperature change of 288.6 K.

** Calculation assumes a temperature change of 288.6 K and a 70 percent packing fraction for the solids.

and filtration equipment and these costs will be higher for the rock system.

There are a number of technical problems that must be solved before a rock storage system can be successfully operated. The rock system requires that the oil be pumped through the tank. The tanks are large, and to be economical they must be unpressurized. Thus, the oil must be pumped into and out of the tanks with sufficient control to ensure that the tank remains unpressurized. The oil flow in the tanks may have a tendency to channel. If this happens, baffles will be required inside the tank adding to the tank cost and complexity. The rocks will have a tendency to settle in the tank as they are cycled thermally. This settling will both load the tank walls and abrade the rocks as they try to move relative to each other. The first effect will result in additional tank costs and the effect will necessitate the use of special pumps or a filtration system to remove the rock particles from the oil. And finally, the long term chemical compatibility of oil and rocks at high temperatures is unknown. This problem is complicated by the fact that the chemical composition of the rocks will vary depending on their source.

These problems associated with the rock system have eliminated it from further consideration on this program.

D. CAPTIVE SALT HEAT OF FUSION SYSTEM

A latent heat thermal storage concept has also been investigated on this program. The system would use an inorganic salt encapsulated in thin wall tubes. These tubes would be placed in a tank in bundles. A storage heat transfer fluid would be circulated 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. Figure VI-19 shows a schematic of the operation of this heat of fusion concept.

The operation of the system is similar in many ways to the operation of the solid sensible heat system described earlier. 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 tube wall would be thin and would therefore have little thermal resistance. The tube diameter would be small and the heat conduction path 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 continues. During discharge, the oil flow through the tank would be reversed. The direction of the thermal/melting gradient would also reverse. It would now move from the bottom of the tank towards the top.

Figure VI-19. Encapsulated Salt Latent Heat Storage Concept.

Table VI-4 contains a list of inorganic salt mixtures that are suitable for use in the condensing/evaporating stage of the thermal storage subsystem. All of these salts melt in the range from 540 K to 555 K. Not all of the thermal parameters that are needed to define the salt performance as a heat storage media were available in the literature. For instance, the latent heat of fusion, specific heat, and thermal conductivity of these eutectic mixtures have not all been published. Since all the pertinent information was not available, a cost study was carried out for two hypothetical salts. The latent heat of fusion of one salt was assumed to be 92.2 J/gr and for the second salt a latent heat of fusion of 372 J/gr was assumed. These numbers represent what we believe to be the bounds of the latent heat values. The tank size used in these estimates was the same size used in the oil and rock estimates and the price of oil was also assumed to be the same. The cost of encapsulation was included in the tube price of \$0.55/kilogram (\$.25/pound) of tube.

The results of these two price estimates are summarized in Table VI-2. The total heat stored was the same for all four systems whose costs are estimated in this table. The cost for each salt system less the cost of the salt is listed in the table. This subtotal for the 92.2 J/gr salt is greater than the total cost of the sensible heat oil system, and a salt with a latent heat of fusion in this range would not be competitive. The subtotal for the 372 J/gr salt is $\$1.30 \times 10^6$ less than the total for the oil system and $\$0.06 \times 10^6$ less than the total for the rock system. A salt cost of \$1.06/Kg (\$.48 lb.) or less would make the salt system competitive with the oil system.

There are problem areas in the captive salt concept that must be investigated thoroughly before a long lifetime system utilizing this approach can be built with confidence. The first problem that must be considered is the long term chemical stability of the eutectic salt mixture. The chemical degradation of the salt could result in:

- 1) A pressure buildup in the tubes through gas evolution,
- 2) A drift in the melting temperature,
- 3) Or a decrease in the heat of fusion of the salt.

The chemical compatibility of the salt and tubes would have to be established. The thermal and pressure stresses in the tubing are cyclic. Any embrittlement of the tube walls through interaction with the salt would lead to cracking and tube failure. The price of carbon steel tubing was used in estimating the salt system cost. If stainless steel tubes are required to increase system lifetime, the tube cost will be much greater. A manufacturing process for encapsulating large volumes of salt economically would also have to be developed.

None of the salt mixtures listed in Table VI-4 have been used on a large scale in industrial applications for a long period of time. A thorough investigation of the problem areas just discussed would

TABLE VI-4
CANDIDATE SALT MIXTURES FOR THE CONDENSING STAGE OF THE TSS

Salt Mixtures	Composition (percent)	MP		Cost/lb (\$/lb)	Cost/kg (\$/kg)
		(K)	(°F)		
1. BaCl ₂ -ZnCl ₂	3-97	545	(521.6)	0.166	0.366
2. Ba(NO ₃) ₂ -NaCl-NaNO ₃	5.8-7.9-86.2	553	(536)	0.15	0.33
3. CuCl-PbCl ₂	5.91-40.9	553	(536)	0.75	1.65
4. CuCl-FeCl ₃	50-50	548	(527)	0.422	0.93
5. CuCl-PbCl ₂	10-90	547.5	(526.1)	0.76	1.68
6. CuCN-KCN	48-52	548	(527)	0.83	1.83
7. InCl ₃ -ZnCl ₂	4-96	549	(528.8)	3.68	8.11
8. KCl-ZnCl ₂	66-34	547	(525.2)	0.17	0.374
9. KCl-LiCl-LiOH	11.5-45.0-43.5	553	(536)	1.05	2.31
10. KCl-PbCl ₂ -PbI ₂	31.2-27.8-40.9	543	(518)	1.09	2.40
11. KNO ₃ -Ba(NO ₃) ₂	15-85	548	(527)	0.17	0.375
12. KNO ₃ -Sr(NO ₃) ₂	85-15	548	(527)	0.12	0.265
13. LiBr-LiOH	55-45	548	(527)	1.51	3.33
14. LiCl-LiOH	34.5-65.5	547	(525.2)	1.16	2.56
15. MgCl ₂ -ZnCl ₂	1-99	544	(519.8)	1.17	2.58
16. NaCl-NaNO ₃ -Na ₂ SO ₄	8.4-86.3-5.3	551	(532.4)	0.09	0.198
17. NaCl-ZnCl ₂	42-58	543	(518)	0.19	0.419
18. NaNO ₃ -Pb(NC ₃) ₂	84.5-15.5	547.2	(525.6)	0.24	0.529
19. NaOH-Na ₂ S	95.7-4.3	553	(536)	0.26	0.573

therefore be required before any one of these mixtures could be a viable candidate as a long lifetime heat storage medium. It has been judged that this investigation could not be completed in time to allow a thermal storage subsystem to be built and operated in the time available for the program. The captive salt approach is therefore no longer being actively investigated on this program.

E. OIL TESTS

In addition to the heat storage technique studies, some preliminary hydrocarbon oil tests are being conducted. A number of hydrocarbon oils suitable for use in the condensing/evaporating stage of the thermal storage subsystem have been identified. These oils along with their approximate bulk quantity prices are listed in Table VI-5. The object of the tests is to determine the effect of elevated temperatures on certain thermal and physical properties: viscosity at room temperature, weight loss or gain, accumulation of solid residues, and other evidence of deterioration as evidenced by visual inspection.

The tests that are normally conducted by the suppliers of heat transfer oils often do not simulate the thermal storage subsystem operating conditions. For instance, oil lifetime tests are usually conducted with the oil samples sealed in tubes. The pressure in the tubes during the tests is neither controlled or measured. Frequently, the existing test results do not include measurements of all of the properties that are of interest. Independent oil test are therefore being conducted to achieve a reasonably close simulation of thermal storage conditions.

Figure VI-20 shows a diagram of the apparatus that has been used in the oil evaluation tests. The oils were poured into individual glass beakers and covered with watch glasses. The beakers were placed on a slab of insulation inside a sealed copper box. The copper box was in turn placed inside a furnace. Copper is used for the box material to minimize thermal gradients. At the beginning of the tests, the properties listed above are observed for all of the samples. The samples are then soaked at 572 K (570° F) for periods up to 500 hours. During the tests, a low pressure nitrogen blanket is maintained over the oil samples. At 100 or 200 hour intervals, the samples are cooled to room temperature, and the four properties again measured. The results of these tests will be presented in a later report.

F. COST ESTIMATES FOR A 10 MWe PLANT

Figure VI-21 presents a preliminary estimate of the relative costs of the thermal storage system capital equipment. These cost relationships were obtained by taking detailed estimates for the Thermal Storage Research Experiment and adjusting them to Pilot Plant size, capacity, and steam rate using size-cost relationships from published literature.

TABLE VI-5

PRICE FOR BULK QUANTITIES OF HYDROCARBON OIL

<u>Oil Trade Name</u>	<u>Cost</u>	
	<u>Dollars/m³</u>	<u>Dollars/Gallon</u>
Amoco Oil 4199	266.81	1.01
Gulf Security 53	250.96	.95
Shell Thermia 33	219.26	.83
Exxon Caloria HT43	258.89	.98
Therminol 55	396.26	1.50
Therminol 66	1505.77	5.70

Figure VI-20. Hydrocarbon Oil Test Apparatus.

Figure VI-21. Relative Cost Relationships for the Preliminary Design Pilot Plant Thermal Storage System.

These relative costs were derived for the Pilot Plant preliminary design configuration and are expected to change relatively after the Pilot Plant design has been optimized. This optimization of the Pilot Plant design is scheduled to be completed later in the program.

6. RESEARCH EXPERIMENT DESCRIPTION

A. RESEARCH EXPERIMENT REQUIREMENTS

The Research Experiment is designed to accomplish the following -

- Accept superheated steam at 8619 kPa (1250 psig) and 783 K (950°F) in the CHARGE MODE and produce superheated steam at 4137 kPa (600 psig) at 672 K (750°F) in the DISCHARGE MODE.
- Function in the CHARGE & DISCHARGE MODES at
 - .. A maximum rate of 2 MWth
 - .. A minimum rate of 0.4 MWth
 - .. Rates between the maximum and minimum rates
 - .. Rates which vary throughout the maximum-minimum range and simulate the anticipated characteristics of the Pilot Plant Central Receiver and Turbine.
- Demonstrate the suitability of commercially available control systems for the control of the Thermal Storage System.
- Accept in the charged condition sufficient energy such that 1.6 MWhth can be recovered into the DISCHARGE MODE steam at 4137 kPa (600 psig) and 672 K (750°F).
- Provide performance data suitable for correlation to design predictions.
- Establish design criteria and performance expectancies for the Pilot Plant Thermal Storage Subsystem.

B. DESCRIPTION OF RESEARCH EXPERIMENT

The Thermal Storage Subsystem Research Experiment temperature-enthalpy diagram is shown in Figure VI-22. In the Research Experiment during charging, steam enters at 783 K (950°F) and 8619 kPa (1250 psig) and is cooled to 626 K (667°F) in the first stage section (heat transfer salt stage). The remaining superheat in the steam is then removed, the steam is condensed at 574 K (574°F) and the condensate is sub-cooled to 539 K (510°F) in the hydrocarbon oil stage.

During discharging, feedwater enters at 505 K (450°F) and 4137 kPa (600 psig) and is preheated and evaporated in the hydrocarbon oil stage at 530 K (495°F). The saturated steam is then superheated to 672 K (750°F) in the HITEC stage.

The simplified Research Experiment is illustrated in Figure VI-23 in the charging mode and in Figure VI-24 in the discharging mode. These are highly simplified functional schematics. In Figure VI-23, the charging steam from the Georgia Power Company Plant at 783 K (950°F), 8619 kPa (1250 psi) and 3254 kg/hr (7159 lb/hr) is shown entering the

Figure VI-22. Temperature-Enthalpy Diagram, Thermal Storage Subsystem.

Figure VI-23. Schematic of Research Experiment Operation from Receiver (Charging).

system from the right side. The steam is cooled to 772 K (930°F) using an injector-type desuperheater. Simulation of the characteristics of a solar boiler will be accomplished using a throttling valve which will be programmed to vary the steam flow. The superheated steam at 772 K (930°F) enters the desuperheater/superheater heat exchanger (HE 1) and in counterflow, gives up energy to the HITEC which is pumped from the low-temperature storage tank at 577 K (580°F), through the heat exchanger, where it increases in temperature to 727 K (850°F) and is stored in the high temperature storage tank. The steam temperature at the exit of the desuperheater/superheater heat exchanger is expected to be 626 K (667°F). At this point, the temperature of the steam is considered to be excessively high for entry into the oil circuit, and therefore, undergoes further temperature reduction using a second injector-type desuperheater, down to 600 K (620°F) where it then is admitted to the condenser/evaporator heat exchanger (HE 2).

The steam entering the condenser/evaporator at 600 K (620°F) is cooled to saturation temperature at 574 K (574°F) and condensed. The condensate is then further cooled in this unit to 566 K (560°F). This condensate then passes through the third heat exchanger (HE 3) where it is subcooled to less than 539 K (510°F) and discharged. The desuperheating/condensing/subcooling releases energy to the hydrocarbon oil which is pumped through the system from the low temperature storage tank at 531 K (497°F), to the high temperature storage tank at 570 K (567°F).

In the charge mode, the charge rate is 2 MWth and the stored energy in the HITEC and hydrocarbon oil storage tanks provide for discharging at a rate of 2 MWth at a steam rate of 3267 kg/hr (7187 lbs/hr) at exit conditions of 672 K (750°F) and 4135 kPa (600 psig). The discharge mode reverses the fluid flow direction and utilizes treated feedwater from the Georgia Power Company Power Plant. The feedwater at 505 K (450°F) and 4135 kPa (600 psig) enters HE 3 which now functions as a preheater to raise the feedwater temperature to 525 K (485°F) where it passes into HE 2. Evaporation takes place at 530 K (494°F) in HE 2. Separation occurs in HE 2 and the saturated steam passes into HE 3 where it is superheated to 672 K (750°F) and discharged. A throttling valve is used to simulate the turbine characteristics.

The detailed schematic is shown in Figure VI-25. Operation is planned to virtually duplicate the operation of the Pilot Plant; the only exception will be the use of feedwater at 505 K (450°F) rather than the Pilot Plant condition of 477 K (400°F). All control functions and devices will be incorporated analogous to the Pilot Plant requirements. Testing will be designed to provide operational data over the range of 0.4 MWth to 2.0 MWth rates with the storage capacity of 1.6 MWhth.

Figure VI-25. Detailed Schematic, Two-Stage Research Experiment at 2.0 MWth Capacity.

C. DISCUSSION OF SCALING

The scaling of the Research Experiment Thermal Storage Subsystem to the Pilot Plant is shown schematically in Figure VI-26. The maximum flow rate into the Research Experiment during charging will be 3214 Kg/hr (7085 lb/hr) and the maximum flow rate out of the system during discharge will be 3267 Kg/hr (7187 lb/hr). Both of these mass flow rates correspond to a heat flow rate of 2 MWth because 2241 J/gm (963.5 Btu/lb) is extracted from the steam during charging, and 2207 J/gm (948.9 Btu/lb) is absorbed by the feedwater to make superheated steam during discharge. The maximum steam flow rates for the Pilot Plant will be 61,236 Kg/hr (135,000 lb/hr) during charging and 57,380 Kg/hr (126,500 lb/hr) during discharge. The steam conditions for the two systems are the same with two exceptions. The condensing pressure in the Research Experiment will be slightly lower than the corresponding pressure in the Pilot Plant. The pressure difference will be reported when the Research Experiment design is complete enough to allow its determination. The feedwater will enter the Research Experiment at a temperature of 505 K (450° F). The Pilot Plant feedwater temperature is expected to be 477 K (400° F) at the nominal discharge rate. This lower temperature requires that 2335 J/gm (1004 Btu/lb) be supplied to the Pilot Plant feedwater to produce steam at the desired conditions during discharge at the nominal steam flow rate. Both the condensing pressure difference and the feedwater temperature difference are small. Neither of the differences affects any fundamental aspect of system operation. If these differences are ignored, then the scaling ratio of 20 to 1 is obtained by comparing the maximum discharge rate of the Research Experiment with the maximum discharge rate of the Pilot Plant. Approximately the same scaling relation holds for the charging rates.

The storage capacity of the Research Experiment is presently set at 1.6 MWHth. The storage capacity of the Pilot Plant is based on the nominal discharge rate of 47,174 Kg/hr (104,000 lb/hr). The Thermal Storage Subsystem of the Pilot Plant will supply steam at the nominal rate for six hours. This requires a storage capacity of 192 MWHth for the specified steam conditions. In general, the size of the tanks will be based on nominal flow conditions and the size of the heat exchangers, drums, and fluid lines will be based on maximum flow conditions.

RESEARCH EXPERIMENT

120 to 1

20 to 1

PILOT PLANT

Figure VI-26. Scaling of Research Experiment.

7. RESEARCH EXPERIMENT PERFORMANCE STUDIES

A. MITAS COMPUTER STUDIES

MITAS simulations of the thermal storage subsystem for the Research Experiment are based on the Pilot Plant model. Aside from scaling differences in the physical sizes of heat exchangers, storage tanks, and fluid flow rates, the basic analog network and control system are essentially identical in the two models.

The curves plotted in Figures VI-27, VI-28, and VI-29 represent the anticipated charging behavior of a Research Experiment carrying a thermal power rating that is 1/8 of the rating for the Pilot Plant. Note: (The Research Experiment computer studies reported here represent a 5 MWth experiment. Since this analysis was completed the size of the experiment has been reduced to 2 MWth.) A step input of charging steam flowing at 7258 Kg/hr (16,000 lb/hr) is applied in this startup simulation. Initial heat exchanger and thermal fluid temperatures fall in the range of 394 K (250° F) to 589 K (600° F).

Temperatures of steam and water at the outlets of the three heat exchangers are plotted in Figure VI-27. These temperature responses are similar to the results obtained for the Pilot Plant. With the exception of the condenser, steady state temperatures are reached within 10 minutes following the sudden startup. The periodic oscillation in the water temperature at the outlet of the condenser has been traced to a deficiency in the basic analytical models. This oscillation is not observed in later iterative models which satisfy pressure and energy balances in the condenser/evaporator stage at each time interval in the finite difference thermal calculations.

Flow rates of thermal fluids and recycle water are shown in Figures VI-28 and VI-29 for a charging steam flow of 7258 Kg/hr (16,000 lb/hr). The flow of these fluids through the system generally experiences rapid oscillations during the first minutes after startup. The small periodic oscillation (period equals 0.0875 hours) present in Figure VI-28 is again attributed to the approximations being employed in modeling the condenser in this particular simulation. Several initial oscillations are observed in the thermal fluid flow rates for the desuperheater and subcooler.

The controllers for the above MITAS simulations have proportional action. While the steady state error of this type controller creates few problems in liquid level control loops, proportional action in fluid loops often leads to a "bang-bang" valve operation during transient conditions where the valve oscillates between the fully open and completely shut states. Such behavior is graphically demonstrated in the initial flow of thermal salt for the desuperheater shown in Figure VI-28. Conventional flow controllers typically employ

Figure VI-27. Research Experiment Predicted Heat Exchanger Outlet Temperatures vs Time During Charging.

Figure VI-28. Research Experiment Predicted Storage Media Flow Rates vs Time for a Step Input of Charging Steam.

Figure VI-29. Research Experiment Predicted Storage Medium and Recycle Water Temperature Flow Rates vs Time.

proportional action plus reset to provide tight, fast control action. Controllers with this action have recently been simulated in the fluid control loops of a Research Experiment model rated at 2.0 MWth. Preliminary results indicate a considerable improvement in fluid control response characteristics. Using a startup input of steam flowing at 2903 Kg/hr (6400 lb/hr) with the improved control system, the salt flow response curve for the desuperheater reveals a complete elimination of the initial "bang-bang" oscillation observed for the proportional action controller. Further adjustments in the width of the proportional band and reset time constant are expected to reduce the small low frequency oscillation (period equals 0.125 hours) in this stage. The low temperature thermal fluid flows through both the condenser and the subcooler in this model of the Research Experiment. The thermal fluid flow rates predicted for these stages behaves in a smooth manner that is nearly optimum.

B. STORAGE FLUID TANK HEAT LOSS

The fluid storage tanks are required to store enough hot fluid to deliver 1.6 million watt-hours of thermal energy (MWHth) into steam when the thermal fluids are pumped through the heat exchange equipment. The oil tank system (cold and hot tanks) must be large enough to deliver 1.30 MWHth and the salt tank system must be capable of delivering 0.3 MWHth.

These tank systems are designed such that they have a ten percent excess capacity built in for tank heat loss. As such, when the storage system has just been charged the oil system will be capable of supplying 1.43 MWHth and the salt system will be capable of supplying 0.33 MWHth. A schematic illustration of the oil and salt tanks is presented in Figure VI-30. The heat loss from these tanks as a function of insulation thickness has been generated (Figure VI-31 for the oil tank and Figure VI-32 for the salt tank).

During a Research Experiment test, the time between a charge sequence and a discharge sequence can be kept to less than 8 hours. For the storage system to be able to deliver 1.6 MWHth with an eight hour hold between charge and discharge, the heat loss from the oil tank must be equal to or less than 16.25 KWth and the heat loss from the salt tank must be equal to or less than 3.75 KWth. If the span time between the charge and discharge is less than eight hours, more than 1.6 MWHth can be delivered by the storage system. Conversely if the time span is more than eight hours, less than 1.6 MWHth will be delivered by the storage system.

An insulation thickness of 0.127 meters (five inches) is satisfactory for both tanks (see Figure VI-31 and Figure VI-32) to meet these heat loss requirements.

The temperature profile through a 0.127 meter thickness of insulation on both the salt tank and oil tank is shown on Figure VI-33. Fiber-

Figure VI-30. Tank Schematic.

Figure VI-31. Oil Tank Heat Loss vs Insulation Thickness.

Figure VI-32. Salt Tank Heat Loss vs Insulation Thickness.

Figure VI-33. Tank Insulation Temperature Profile.

glass insulation can be used at temperatures below 533 K (500°F). A mineral wool insulation will be required at temperatures above 533 K. The oil system will require 0.0254 meters (one inch) of mineral wool next to the tank with 0.1016 meters (4 inches) of fiberglass for the rest of the insulation. A covering on the external surface will be necessary to protect the insulation from rain and to provide wind protection.

The insulation on the salt system will have the same general construction except that the mineral wool will be 0.0635 meters (2.5 inches) thick and the fiberglass salt insulation will be 0.0635 meters (2.5 inches) thick.

The temperature decay history of a fully charged oil tank (initial temperature 566 K (560°F)) is shown on Figure VI-34. Also shown in Figure VI-34, is the temperature decay history of an empty oil tank. The empty tank was initially at the cold storage temperature, 533 K (500°F).

Figure VI-35 shows the temperature history of the salt tank system. The salt temperature will not be allowed to go below 477 K (400°F) to protect it from freezing. This will be accomplished by the addition of heat to the tank by a steam coil in the salt tank.

The full tank takes in excess of 20 days before it would require the addition of heat. The empty tank (still contains some salt) would require heat after 24 hours.

The handling of the salt tanks during long down times has not yet been defined. Two approaches are under consideration:

(1) Increase the insulation on the salt tank and (2) During overnight or longer non-operating times, store fifty percent of the salt in one tank and fifty percent of the salt in the other tank.

The discussion and analysis presented herein are preliminary. The design of the Research Experiment tanks has just begun. As such, the heat loss analysis presented herein will be updated to the actual tank design and will include the effects of all penetrations when the geometry of these penetrations is known. It is also possible that the long time non-operating temperature decay of the salt tanks will require additional thickness of insulation on the salt tanks.

Figure VI-34. Oil Tank Temperature History.

Figure VI-35. Salt Tank Temperature History.

8. RESEARCH EXPERIMENT LOCATION

A. DESCRIPTION OF PLANT YATES

Arrangements are being made with the Production Department and Engineering Department of the Georgia Power Company to build and operate the Thermal Storage Subsystem Research Experiment at Plant Yates, near Newnan, Georgia, about 80 kilometers southwest of Atlanta. Plant Yates is a coal-fired steam turbogeneration plant which has three units whose steam conditions match those required for the Research Experiment. A total of seven units are in operation at the site with a total electrical output capability of 1,300 MW. The three units with compatible steam conditions were completed in 1948. Each of these units has an electrical output of 100 Mw and turbine throttle inlet steam conditions of 8,619 kPa (1250 psig) and 783 K (950°F). Two of these units are scheduled for annual maintenance in early 1976, and Georgia Power Company anticipates that the necessary steam piping could be installed on either unit at this time.

The aerial photograph in Figure VI-36 shows the old power plant which houses five of the seven turbogeneration units at Plant Yates. The site selected for the Research Experiment is a triangular plot of land, located near the coal conveyor entering the building. Steam and water lines would be run from the building to the experiment site along an existing pipe trestle which is parallel to and approximately underneath the coal conveyor. Waste water from the old plant is discharged through underground conduits into the Chattahoochee River which is adjacent to the plant but not shown in the photograph.

B. DISCUSSION OF INTERFACES

Figure VI-37 is a view of the planned Research Experiment site. The pipe trestle which will carry the Research Experiment steam and water supplies from the power plant to the experiment site appears in the upper left portion of the figure. The corner of the plant is visible in this photograph. The heat exchangers for the Research Experiment will be located close to the road in the middle of the photograph. This arrangement will allow access to the heat exchangers and also place these units as close to the steam and feedwater sources as possible.

Figure VI-38 contains schematics for the interfaces with the power plant at Plant Yates. Three of the steam generating units deliver 8,619 kPa (1250 psig) and 783 K (950°F) steam to their turbogenerators and one of these units will supply charging steam to the Research Experiment. The pressure delivered to the Research Experiment will be slightly lower due to the pressure drop in the steam line leading to the Research Experiment heat exchangers. This pressure drop will be relatively small and will not have a significant effect on the operation of the experiment. The condensate will leave the subcooling

Figure VI-36. Aerial View of the Old Power Plant At Georgia Power Company Plant Yates.

Figure VI-37. Photograph of Planned Research Experiment Site.

CHARGING CYCLE

DISCHARGING CYCLE

Figure VI-38. Schematic of Interfaces with Plant Yates.

heat exchanger of the Research Experiment during charging at approximately 539 K (510°F). This water will not be returned to Plant Yates. Instead, it will either be further subcooled to 333 K (140°F) and discharged to a holding tank, or it will be flashed to a steam/water mixture through a pressure reducing valve and discharged to the atmosphere through a silencer. The first approach using the subcooler is depicted in Figure VI-38.

The temperature of the feedwater that the Research Experiment will receive from Plant Yates will be 505 K (450°F) when the supplying turbogenerating unit is operating at its nominal output capacity of 100 MW. This feedwater is slightly hotter than the 477 K (400°F) feedwater the Pilot Plant will produce when it is operating from storage at its normal output rate of 7 MW. A cooler could be used during the discharge operation to lower the feedwater temperature to 477 K, but this temperature does not affect any fundamental aspect of system operation. It does require a shift in the balance of heat stored in the oil and heat transfer salt stages, but this balancing shift can be accomplished by simply adjusting the exit temperature from the heat transfer salt stage. It has therefore been decided to operate the Research Experiment with 505 K (450°F) feedwater. The steam that is produced during system discharge will be discharged through a silencer to the atmosphere as indicated in Figure VI-38. The small pressure and temperature differences discussed above are the only differences that have been identified between Pilot Plant and Research Experiment operation.

C. TOP DRAWING OF RESEARCH EXPERIMENT AT PLANT YATES

Figure VI-39 shows the preliminary layout of the Research Experiment at the Georgia Power Company, Plant Yates, site. The major components are shown and are named whereas minor components reflect designated code "tags" which are described in the nomenclature listing.

The number of components in selected classifications are presented in Table VI-6. Principal items, reflecting the larger sizes and weights, are presented in Table VI-7.

The physical arrangements that are shown in the layout (Figure VI-39) do not reflect the conservative spacing interpretation of NFPA 30-73 which will result in adjustments to the physical layout as will further engineering studies to minimize costs and improve function. Layout designs will comply with code and regulatory requirements.

The plant will occupy approximately one-third acre to the west of the power building. Steam and feedwater will be extracted from the Unit No. 1 generation system and will be provided to the experiment site through piping supported by an existing ash line trestle. Discarded steam will be released through a discharge silencer and discarded condensate will be discharged under controlled conditions (to assure absence of contamination from oil and/or salt and temperature control) into the plant storm drain system.

TABLE VI-6
SUMMARY OF COMPONENTS IN THE RESEARCH EXPERIMENT

<u>Quantity</u>	<u>Description</u>
9	Flow control valves
12	Safety relief valves or rupture discs
7	Automatic on-off valves
17	Manual valves
3	Fluid pumps; 2 HITEC, 1 oil
7	Tanks; 4 storage, 2 safety receiver 1 surge
3	Heat exchangers; 1 HITEC, 2 oil
38	Primary sensors. pressure, temperature, flow
2	Injector desuperheaters
3	Line filters

TABLE VI-7

PRELIMINARY ESTIMATES OF SIZES AND WEIGHTS OF MAJOR COMPONENTS

<u>Item</u>	<u>Size, Less Insulation</u>		<u>Filled Weight</u> kg (lbs)
	<u>Height</u> cm (in)	<u>Diameter</u> cm (in)	
Heat exchanger 1 (2 units)	701 (276)	36 (14)	4,000 (8,800)
Heat exchanger 2	853 (336)	61 (24)	12,000 (26,400)
Heat exchanger 3	244 (96)	25 (10)	600 (1,300)
Oil tanks (2 units)	945 (372)	366 (144)	68,000 (149,600)
HITEC tanks (2 units)	274 (108)	244 (96)	6,800 (15,000)

CENTRAL RECEIVER SOLAR THERMAL PILOT PLANT

SECTION VII—ELECTRICAL POWER GENERATION SUBSYSTEM

SECTION VIII—PILOT PLANT ARCHITECTURE-ENGINEERING AND SUPPORT

PREPARED FOR

MARTIN MARIETTA CORPORATION

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Section VII

ELECTRICAL POWER GENERATION SUBSYSTEM

7.1 INTRODUCTION

The Electrical Power Generation Subsystem (EPGS) conceptual design was based on the pilot plant requirements as 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.
- Condition the working fluid (i.e., pressurize, heat, deaerate, treat chemically, etc., the condensate) and return it to the other subsystems.
- Respond to transient conditions rapidly enough to avoid adverse effects to the operation of all subsystems.

The design point, as specified by ERDA, is at winter solstice at 2 p.m. with an ambient dry bulb temperature of 302 K (85°F).

The EPGS conceptual design effort was divided up into three basic phases. The turbine selection, cycle studies and heat rejection system selection were completed during the initial phase ending in September 1975. That phase was concerned with basic equipment selection and performance considerations. Some instrumentation and control design activity was included to ensure the operational aspects of the EPGS. The next phase included the development of the mechanical system descriptions, system diagram, line sizing, electrical system and balance of the instrumentation and control. This phase consolidated and elaborated on the basic plant developed in the initial phase. Finally, layout drawings were produced for the EPGS structure and the control and administration building.

In addition to these basic design activities, a reliability and failure modes and effects analysis (FMEA) was conducted for the EPGS to ensure a reliable design with adequate safeguards and redundancy.

Optimization studies of the completed baseline conceptual design have been started and will continue through initiation of the preliminary design next fall.

The turbine inlet steam conditions were fixed for the conceptual design at 8.72×10^6 Pa (1,250 psig), 783 K (950°F) for the main steam from the receiver and 4.24×10^6 Pa (600 psig), 672 K (750°F) for the admission steam from storage.

The main steam inlet conditions were based on the recommendation of General Electric made initially to Martin Marietta and subsequently to Bechtel. The temperature limit of 783 K (950°F) is to allow the use of a more ductile steel. This is recommended for turbines used in cyclic operation. The pressure limit of 8.72×10^6 Pa (1,250 psig) is to reduce the thickness of the turbine casing, thereby reducing its thermal stresses caused by cycling.

In an additional meeting with General Electric held late in the conceptual design effort, GE revised their initial recommendation to a maximum main steam pressure limit of 10.1×10^6 Pa (1,450 psig). An increase in throttle pressure to this level may result in a heat rate improvement and lower storage system costs. However, it would also result in higher turbine, piping and valving costs, a slightly thicker casing and a higher windage loss in the high pressure (HP) stage when operating from storage alone due to the greater number of HP stages. A tradeoff study will be performed during an optimization period between the EPGS conceptual and preliminary designs to investigate the 10.1×10^6 Pa (1,450 psig) option.

All heat balances, calculations, performance curves, etc., are based on the initial GE recommendation of 8.72×10^6 Pa (1,250 psig). Martin Marietta has requested this to be increased for the conceptual design to 8.89×10^6 Pa (1,275 psig) based on Thermal Storage Subsystem considerations. No significant changes in the present EPGS conceptual design would result from this slight increase in throttle pressure.

The admission steam conditions of 4.24×10^6 Pa (600 psig), 672 K (750°F) were chosen by Martin Marietta based on a tradeoff study between 672 K (750°F) and 505 K (450°F) storage steam.

The turbine exhaust conditions are determined by the heat rejection system type and size and the prevailing ambient conditions. Air cooling was chosen as the baseline approach in view of water shortage problems in the southwestern states. This results 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. Key temperatures used in the plant overall and design point performance determinations are as follows:

323 K (122°F)	record maximum
315 K (108°F)	1% dry bulb
307 K (93°F)	average summer day
302 K (85°F)	ERDA design point
299 K (79°F)	average summer evening
296 K (73°F)	average spring-fall day
288 K (59°F)	average winter day
288 K (59°F)	average spring-fall evening
281 K (46°F)	average winter evening

7.3 TURBINE SELECTION

7.3.1 Commercial Plant Turbine

Since the pilot plant is intended to simulate future commercial plant operation, it is important to first examine the commercial plant turbine possibilities 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, as shown in Figure 7-1, but recommended the admission turbine as the most flexible. Westinghouse recommended a similar induction type turbine. Allis-Chalmers declined to recommend a specific turbine type since they do not manufacture turbines smaller than 200 MWe. All the turbines shown in Figure 7-1 are the HP sections only. Double flow low pressure (LP) sections would be provided with all options.

The GE mixed pressure turbine would operate without throttling when operating from the receiver alone or from storage alone but would require throttling of receiver steam from 8.72×10^6 Pa (1,250 psig) to 4.24×10^6 Pa (600 psig) for combined operation. Maximum size for this turbine is limited by the control valve gear size to 160 MWe (when operating from the receiver). However, storage steam (at 4.24×10^6 Pa (600 psig), 672 K (750°F)) could generate only 40% of this or approximately 65 MWe. This turbine would also be subjected to significant temperature swings in the front end as the steam source is alternated between the receiver and storage.

The GE admission turbine admits steam at a point in the turbine casing where the internal steam temperature closely matches the admission steam temperature. Operation from the receiver alone, from storage alone and combined operation do not require throttling and the consequent losses are avoided. Maximum size for this turbine is limited by the admission valve gear size to 125 MWe (when operating from the receiver). However, storage steam could generate 80% of this capacity, i.e., 100 MWe. The

Westinghouse induction turbine has similar advantages to the admission turbine except that external valve throttling of storage steam is required for operation from storage alone and combined operation.

In summary, the GE admission turbine appears to be the most flexible machine and at the same time minimizes throttling losses when all operating modes are considered.

7.3.2 Pilot Plant Turbine

Several manufacturers (General Electric, Turbodyne and DeLaval) were contacted for information on small turbines in the 10 MWe range. Westinghouse and Allis-Chalmers do not manufacture turbines of this size. Turbodyne recommended a dual inlet turbine. This turbine may encounter thermal cycling problems in the first stage as the steam source is alternated between the receiver and storage. This turbine type (similar to a boiler feed pump turbine) may also have a low efficiency when operating from the high pressure receiver steam alone. GE recommended an admission turbine based on their recommendation for the commercial unit and because they also consider this type of machine to be the best choice in the 10 MWe range. DeLaval recommended an admission turbine similar to the GE unit.

The General Electric turbine was selected for the conceptual design based on its similarity to the commercial unit, its flexibility, performance and thermal cycling characteristics. Also, General Electric was the most responsive of the three manufacturers in providing considerably more design and performance data. General Electric is also the only manufacturer contacted who manufactures turbines over a range covering both the pilot and commercial plant sizes.

The selected turbine has an electrohydraulic control system operating partial arc control valves at both the main and admission steam inlets. A 12.5 MWe standard turbine-generator size is required to supply the specified 10 MWe net plus approximately 10% auxiliary loads. The turbine has 0.20 m (8 in) last stage buckets of the high backpressure type.

GE can ultimately offer both a low backpressure turbine (backpressure not exceeding 1.69×10^4 Pa (5 in-Hg)) and a high backpressure turbine (backpressure ranging up to $3.39\text{--}4.06 \times 10^4$ Pa (10-12 in-Hg) during operation) for this application. A low backpressure turbine would result in higher gross outputs but would require a larger and more costly dry heat rejection system with considerably greater auxiliary power requirements. The high backpressure turbine has a smaller dry heat rejection system with considerably less auxiliary power requirements.

Information was requested from GE for both turbine types in order to compare net outputs and overall plant costs. However, for the conceptual design, GE would supply information for only one type

FIGURE 7-1 PROPOSED COMMERCIAL TURBINE TYPES

FIGURE 7-3 COMMERCIAL PLANT POSSIBLE CYCLE ARRANGEMENT

FIGURE 7-2 TYPICAL TURBINE CROSS SECTION

of turbine. They chose the high backpressure turbine as the most appropriate and cost-effective for this application. Therefore, all following design information is based on this turbine type. During the optimization period, design information will be sought from GE for the low backpressure turbine to verify their initial selection.

Figure 7-2 shows a cross section of a typical GE automatic extraction turbine of approximately the same size as the 12.5 MWe pilot plant turbine. Although the pilot plant turbine is an admission machine, its cross section would be essentially identical to this turbine shown, including the main steam and admission steam valve gear. Four uncontrolled extraction nozzles will also be provided downstream of the admission valve gear in the turbine casing.

7.4 CYCLE ARRANGEMENT

A representative commercial plant cycle using the admission turbine would include the high pressure turbine with an admission point from storage and a double flow low pressure turbine with uncontrolled extractions for five stages of feedwater heating, as shown in Figure 7-3. Feedwater pumping is split between two feedwater pumps in series to feed both the storage system and the receiver to minimize throttling losses which would otherwise occur in the line to storage if a single feedwater pump were used.

The number of feedwater heaters is typical for a 100-150 MWe plant with 8.72×10^6 Pa (1,250 psig), 783 K (950°F) steam conditions. Economic tradeoff studies may eventually lead to more or less heaters.

Reheat has not been included since it is not considered practical in the 100-150 MWe range for a solar thermal power plant where long reheat pipe runs would be necessary. These runs would extend from the turbine to the receiver and back down to the turbine. They would require large diameter, high alloy steel pipe with expansion loops and supports. The considerable additional cost of this piping is not considered justifiable.

The pilot plant cycle shown in Figure 7-4 is intended to simulate the larger commercial unit while maintaining high performance in the smaller 10 MWe plant. Only four feedwater heaters are considered necessary, and all pumps, controls and operational features remain similar. The admission turbine is not split into high and low pressure casings as in the commercial unit but is functionally identical.

FIGURE 7-4 PILOT PLANT CYCLE ARRANGEMENT

NOTE: FOR OPERATION ON ADMISSION STEAM ONLY CLOSE STOP VALVE¹ AND OPEN BYPASS COOLING LINE

FIGURE 7-5 PERFORMANCE CURVE OF 12.5 MWe TURBINE

7.5 PILOT PLANT PERFORMANCE AND TRADEOFFS

7.5.1 Turbine Performance

After the turbine type had been selected, based on preliminary performance information and other factors discussed above, more definitive performance information and heat balances were requested from GE. The performance curve for the selected admission turbine is shown in Figure 7-5. This curve provides generator gross output in megawatts as a function of both 8.72×10^6 Pa (1,250 psig), 783 K (950°F) main steam throttle flow and 4.24×10^6 Pa (600 psig), 672 K (750°F) admission flow at 2.03×10^4 Pa (6 in-Hg) turbine backpressure. The curve can be read directly for operation on main steam alone or main steam and admission steam in combination. For operation on admission steam alone, approximately 0.3 MWe should be subtracted from the generator output for front end cooling steam windage losses and 6,800 kg/h (15,000 lb/h) of admission steam, for front end cooling, must be added to the admission flow curve value. The cooling steam both eliminates overheating of the high pressure section and generates power in the low pressure section. This curve is based on four stages of feedwater heating, and the uncontrolled extraction flows for feedwater heating have been considered in obtaining the generator output.

Figure 7-6 shows the 4.24×10^6 Pa (600 psig) turbine expansion line end point enthalpy and cycle final feedwater temperature versus apparent LP section flow. Steam temperatures at 4.24×10^6 Pa (600 psig) are shown in parentheses adjacent to the corresponding expansion line enthalpies. Since this is the admission point on the turbine, these temperatures give an indication of the degree of mismatch between the expanded high pressure (HP) steam and the 672 K (750°F) admission steam at the admission point.

As the HP steam flow decreases, the mismatch increases from approximately 28 K (50°F) to 56 K (100°F). Further studies of the admission steam pressure during the optimization period may result in a lower admission steam pressure (keeping the 672 K (750°F) constant) due to thermal storage considerations. This would tend to reduce the temperature mismatch, although GE has stated that a 56 K (100°F) mismatch is satisfactory.

The cycle final feedwater temperature versus apparent LP section flow curve assumes four feedwater heaters in operation. If the top feedwater heater fails and is bypassed, the final feedwater temperature will drop approximately 42 K (75°F) at the 59,000 kg/h (130,000 lb/h) flow. Failure and bypass of other individual closed feedwater heaters will not significantly affect the final feedwater temperature.

Figure 7-7 provides a correction in kilowatts to the performance curve generator output in Figure 7-5 for turbine pressures other than the baseline 20,300 Pa (6 in-Hg) value. Backpressures ranging from 6,800 Pa (2 in-Hg) to 34,000 Pa (10 in-Hg) are

FIGURE 7-6 TURBINE 4,240 kPa (600 psig) END POINT AND FINAL FEEDWATER TEMPERATURE CURVES

FIGURE 7-7 APPROXIMATE CORRECTION FOR VARIANT EXHAUST PRESSURE

expected as operation varies from winter to summer, respectively. The actual backpressure experienced depends upon the size of the selected heat rejection system and the ambient conditions.

7.5.2 Heat Balances

The base heat balances developed by GE were for 12.5 MWe and 11.0 MWe gross output operating from the receiver alone and 9.4 MWe gross output operating from storage alone. The base heat balance diagram and values are given in Figure 7-8 and Table 7-1, respectively. Minor flows such as valve stem leakage and end packing leakage are not included in the balance values in Table 7-1.

The 12.5 MWe gross heat balance represents the maximum possible flow and gross output conditions when operating from the receiver alone on 8.72×10^6 Pa (1,250 psig), 783 K (950°F) steam with 2.03×10^4 Pa (6 in-Hg) backpressure. These conditions could be experienced during peak insolation periods when design point values are exceeded. The cycle gross heat rate for this condition is 1.175×10^7 J/kWh (11,136 BTU/kWh). Net output from the plant depends on the auxiliary power requirements but would be approximately 11.25 MWe. This heat balance corresponds to the point on the upper tip of the turbine performance curve in Figure 7-5.

The 11.0 MWe gross heat balance (with 2.03×10^4 Pa (6 in-Hg) backpressure) was made to establish a fixed receiver design flow rate to meet the design condition requirement of 10 MWe net output (auxiliary power requirements were established at approximately 10% by preliminary calculations). As discussed below, further calculations, cooling system selection and auxiliary power requirements definition resulted in an actual output of 10.225 MWe at the design winter condition (302 K (85°F) ambient temperature, 1.25×10^4 Pa (3.7 in-Hg) backpressure and 1.083 MWe auxiliary loads) at the design flow rate of 51,962 kg/h (114,556 lb/h). This heat balance corresponds to 11 MWe generator output on the zero admission flow line of the turbine performance curve in Figure 7-5. The cycle gross heat rate for this condition is 1.190×10^7 J/kWh (11,284 BTU/kWh).

The 9.4 MWe gross heat balance represents the maximum flow and gross power output conditions when operating from storage alone on 4.24×10^6 Pa (600 psig), 672 K (750°F) steam with 2.03×10^4 Pa (6 in-Hg) backpressure. Net plant output, assuming 1.2 MWe auxiliary power, would be 8.2 MWe. The heat balance corresponds to the point on the lower tip of the turbine performance curve in Figure 7-5, i.e., 9.7 MWe minus 0.3 MWe (front end windage losses) equals 9.4 MWe generator output and admission flow equals 50,300 kg/h (111,000 lb/h) plus 6,800 kg/h (15,000 lb/h) front end cooling flow or approximately 57,000 kg/h (126,000 lb/h). The cycle gross heat rate for this condition is 1.396×10^7 J/kWh (13,230 BTU/kWh).

TABLE 7-1
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 MWe GROSS																	
● PRESSURE, kPa	8722				2261	889	279	109						20	104	265	3448
(psia)	1265				327.9	129.3	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	1146	108.7	119.0	233.7	
● FLOW, kg/h	59003	59003	0	0					43776	4972	3354	2731	3142	47946	3142	59003	
(lb/h)	130077	130077	0	0					96508	10962	7394	6020	6926	105701	6926	130077	
RECEIVER STEAM ALONE																	
11.0 MWe GROSS																	
● PRESSURE, kPa	8722				1875	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	6250	5182	5625	93885	5625	114556	
ADMISSION STEAM ALONE																	
9.4 MWe GROSS																	
● PRESSURE, kPa			4238		2199	840	264	105							100	250	3448
(psia)			614.7		318.9	121.9	33.24	15.24						2.947	14.48	36.33	500
● 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

FIGURE 7-8 BASE HEAT BALANCE DIAGRAM

FIGURE 7-9 EPGS MOLLIER DIAGRAM, 12 MWe GROSS

Although no heat balance was provided by General Electric for a case corresponding to 7.0 MWe net when operating from storage, a design flow rate of 104,000 lb/h from storage was estimated from the performance curve in Figure 7-5. Final calculations show this flow rate to provide 7.005 MWe net at the design winter conditions (302 K (85°F) ambient temperature, 1.15×10^4 Pa (3.4 in-Hg) backpressure, 1.123 MWe auxiliary loads).

For illustrative purposes, a Mollier diagram and an enthalpy-entropy diagram are shown in Figures 7-9 and 7-10, respectively, for the maximum 12.5 MWe gross condition. Extraction pressures and temperatures will vary downwards as throttle flow is reduced.

7.5.3 Dry Cooling System Selection

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. The system can be either a direct air cooled condenser type or a Heller type with a direct contact or jet condenser and a dry cooling tower to cool the condensate. No restriction as to type was imposed in the request for information to the manufacturers. However, for plants smaller than 150-200 MWe, the direct air cooled condenser type is generally more economical.

GEA (with a direct air cooled condenser type system) was the only respondent. The GEA system is shown in Figure 7-11 with an inset photograph of a typical dry cooling installation. The fans and heat exchanger surface are located immediately above the turbine-generator area to provide sufficient 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, it is necessary to tradeoff heat rejection system cost differential 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. These are able to maintain turbine backpressures of 2.03×10^4 Pa (6 in-Hg), 3.05×10^4 Pa (9 in-Hg) and 4.06×10^4 Pa (12 in-Hg), when 84.4×10^9 J/h (80×10^6 BTU/h) are rejected at an ambient temperature of 315 K (108°F) (1% summer dry bulb temperature). The performance curves shown in Figure 7-12 are for a nominal 84.4×10^9 J/h (80×10^6 BTU/h) heat rejection. However, curves were also obtained for the same systems with 76×10^9 J/h (72×10^6 BTU/h) and 92.8×10^9 J/h (88×10^6 BTU/h) heat rejection since actual heat rejection for the turbine cases under consideration varies between these values.

Overall plant performance was calculated for operation from storage alone and from the receiver alone based on the previously

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FIGURE 7-11 DRY COOLING SYSTEM AND INSTALLATION

FIGURE 7-10 EPCS T-S DIAGRAM, 12 MWe GROSS

described turbine and all heat rejection system performance curves for winter, summer and spring-fall average ambient conditions with fans at both full and half speed. Average year-round delta kilowatts were calculated for the 2.03×10^4 Pa (6 in-Hg) and 3.05×10^4 Pa (9 in-Hg) systems using the 4.06×10^4 Pa (12 in-Hg) system as a base. The optimum fan speed was identified and used for each case.

Based on a year-round comparison of the additional kilowatts resulting from the larger heat rejection systems against their additional cost, it appears that the 2.03×10^4 Pa (6 in-Hg) and 3.05×10^4 Pa (9 in-Hg) systems are close to the optimum size for the pilot plant with the former system having a slight advantage over the latter. The 4.06×10^4 Pa (12 in-Hg) system results in a significantly lower plant output and is not considered economically feasible. When overall plant costs are better defined, this comparison can be performed more accurately and a more definitive selection can be made.

Table 7-2 summarizes the design features of the selected 2.03×10^4 Pa (6 in-Hg) heat rejection system and includes the latest capital cost estimate from GEA. The log mean temperature difference (LMTD) between the steam and the air is given for the 1% summer dry bulb reference temperature condition. The extended (finned) tubes increase the effective heat exchange surface by a factor of approximately 13.

7.5.4 Performance Summary

Table 7-3 summarizes the plant net power outputs and associated major flows, pressures, etc., for various conditions around the year including the ERDA design point, for the selected 2.03×10^4 Pa (6 in-Hg) heat rejection system case for operation from the receiver alone. Both full and half-speed fan operation were analyzed to enable an optimum operating mode to be selected for the ambient temperature under consideration. The maximum summer temperature of 323 K (122°F) is expected to result in a turbine backpressure of 3.39×10^4 Pa (10 in-Hg), and a condensate temperature of 345 K (162°F). Net power output at the ERDA design point is 10,225 kWe.

The flow rates shown in Table 7-3 are those anticipated at 2 p.m. around the year. However, the flow rates and corresponding net power outputs can be higher if additional thermal energy is available from the receiver.

Table 7-4 summarizes the selected system performance, etc., for operation from storage alone. The temperatures are the average evening temperatures for the hypothetical site in the southwestern states. Net power output at the ERDA design point is 7,005 kWe at the design flow of 47,200 kg/h (104,000 lb/h). This can be increased by the operator by raising the flow rate up to a maximum of approximately 57,200 kg/h (126,000 lb/h).

TABLE 7-2 AIR COOLER DESIGN SUMMARY

Plot Area: 18.3 m x 22.9 m (60 ft x 75 ft)
Cells: 6
Each 8.2 m x 7.6 m (27 ft x 25 ft)
6.4 m (21 ft) Diameter Fans
Each 93 kW (125 hp) 173 rpm
Thermal Data: LMTD = 9.4K (17°F)
U = 35.5 W/m ² K (6.25 Btu/hr °F ft ²)
Surface (Bare) = 5388 m ² (58,000 ft ²)
Surface (Extended) = 70,604 m ² (760,000 ft ²)
Capital Cost: \$960,000

TABLE 7-3 SUMMARY OF OPERATION ON 8,270 kPa (1,250 psig), 783 K (950°F) STEAM

Ambient Air Temperature, K °F	288 (59)	296 (73)	303 (85)	307 (93)	315 (108)	323 (122)
Designated Condition	Winter Average	Spring-Fall Average	ERDA Design PT	Summer Average	1% Summer	Maximum Summer
Condenser Pressure, kPa (Abs)	16.3	22.3	12.5	14.6	22.3	33.9
in. Hg (Abs)	(4.8)	(6.6)	(3.7)	(4.3)	(6.6)	(10.0)
Condenser Duty, 10 ³ J/h	90.0	90.2	89.7	88.5	89.6	91.3
10 ⁶ BTU/h	(85.3)	(85.5)	(85.0)	(83.9)	(84.9)	(86.5)
Throttle Flow, kg/h	51,963	51,638	51,963	51,313	51,313	51,313
lb/h	(114,556)	(113,840)	(114,556)	(113,124)	(113,124)	(113,124)
Fans Speed	Half	Half	Full	Full	Full	Full
Generator Load, kW	11,170	10,850	11,308	11,109	10,777	10,285
Fan Power, kW	92	90	551	543	529	516
Other Auxiliary Power, kW	532	532	532	532	532	532
Net Power, kW	10,546	10,228	10,225	10,034	9,716	9,237
Net Heat Rate 10 ⁷ J/kWh	1.242	1.272	1.281	1.289	1.332	1.400
BTU/kWh	(11,770)	(12,060)	(12,140)	(12,220)	(12,620)	(13,140)

TABLE 7-4 SUMMARY OF OPERATION ON 4,240 kPa (600 psig), 672 K (750° F) STEAM

Ambient Air Temperature, K °F	281 (46)	288 (59)	299 (79)	303 (85)
Designated Condition	Winter Average	Spring-Fall Average	Summer Average	ERDA Design PT
Condenser Pressure, kPa (Abs)	10.2	13.2	10.5	11.6
in. Hg (Abs)	(3.0)	(3.9)	(3.1)	(3.4)
Condenser Duty, 10 ³ J/h	78.0	78.4	78.1	78.2
10 ⁶ BTU/h	(73.9)	(74.3)	(74.0)	(74.1)
Throttle Flow, kg/h	47,174	47,174	47,174	47,174
lb/h	(104,000)	(104,000)	(104,000)	(104,000)
Fans Speed	Half	Half	Full	Full
Generator Load, kW	8,184	8,065	8,168	8,128
Fan Power, kW	95	92	557	551
Other Auxiliary Power, kW	572	572	572	572
Net Power, kW	7517	7401	7039	7005
Net Heat Rate 10 ⁷ J/kWh	1.468	1.491	1.568	1.575
BTU/kWh	(13,910)	(14,130)	(14,860)	(14,930)

TABLE 7-5 AUXILIARY ELECTRICAL POWER REQUIREMENTS, kWe

STEAM SOURCE	RECEIVER	STORAGE
EPGS		
CONDENSATE PUMP	5	5
BOOSTER PUMP	60	55
FEEDWATER PUMPS	200	60
COOLING SYSTEM FANS	90-550	90-550
THERMAL STORAGE SUBSYSTEM	0	215
COLLECTOR SUBSYSTEM	80	80
RECEIVER SUBSYSTEM	12	12
MISCELLANEOUS MECHANICAL EQUIPMENT	75	75
ELECTRICAL LOSSES	100	70
TOTAL	622-1082	662-1122

7.5.5 Auxiliary Loads

Auxiliary power requirements are listed in Table 7-5 for the major components and subsystems for operation from the receiver or operation from storage. Combined operation requirements will vary between the "storage alone" requirements and the "receiver alone" requirements plus 215 kWe. This results in net plant outputs ranging from 7.005 MWe to 10.010 MWe at the ERDA design point as the turbine inlet flow varies from the "storage alone" design flow to the "receiver alone" design flow with the storage system pumps in operation. The spread in total auxiliary power requirements shown in Table 7-5 for both cases results from the variation in fan power requirements depending upon the fan speed selection.

Considerably higher auxiliary power requirements would result if a low backpressure turbine had been selected due to the necessity for increased cooling to maintain the backpressure below the low backpressure turbine 1.69×10^4 Pa (5 in-Hg) maximum limit.

7.5.6 Storage Steam Conditions

In addition to the major EPGS heat rejection system tradeoff study, several other tradeoff studies or inputs to Martin Marietta tradeoffs were produced in the course of the conceptual design.

A comparison of the performance and cost impact on the EPGS was made between 4.24×10^6 Pa (600 psig), 672 K (750°F) and 1.82×10^6 Pa (250 psig), 505 K (450°F) storage steam conditions as requested by Martin Marietta. The results are summarized in Table 7-6. This information was used by Martin Marietta together with other subsystem impact data to arrive at the 4.24×10^6 Pa (600 psig), 672 K (750°F) storage steam condition selection. The \$550,000 additional EPGS cost for the 1.82×10^6 Pa (250 psig), 505 K (450°F) case results from increases in the cycle equipment and piping sizes arising from the higher flow rate and heat rejection. The turbine size for this case is larger since the production of 7 MWe net from storage requires 60,010 kg/h (132,300 lb/h) maximum turbine flow compared to 49,900 kg/h (110,000 lb/h) maximum turbine flow for the turbine with 4.24×10^6 Pa (600 psig), 672 K (750°F) admission. The 1.82×10^6 Pa (250 psig), 505 K (450°F) case cycle has a considerably poorer heat rate when operating from storage (18.072×10^6 J/kWh (17,130 BTU/kWh)) compared to the 4.24×10^6 J/kWh (13,730 BTU/kWh)).

Efficiencies, flow rates and heat rates shown in Table 7-6 were obtained early in the study and do not necessarily reflect the latest values.

TABLE 7-6 STORAGE STEAM CONDITIONS TRADEOFF

Steam Conditions kPa/K psig/°F	8720/783 (1250/950)	4238/672 (600/750)	8720/783 (1250/950)	1825/505 (250/450)
Steam Source	Receiver	Storage	Receiver	Storage
Net Power, MWe	10	7	10	7
Theoretical Steam Rate, kg/ kWh	2.98	3.63	2.98	4.99
lb/kWh	(6.58)	(8.01)	(6.58)	(11.00)
Turbine Efficiency %	76	74	75	73
Throttle Steam Flow, kg/h	49,941	43,636	50,395	60,011
lb/h	(110,100)	(96,200)	(111,100)	(132,300)
Net Heat Rate J/kWh	1.24×10^7	1.45×10^7	1.25×10^7	1.81×10^7
Btu/kWh	(11,790)	(13,730)	(11,900)	(17,130)
EPGS Δ Cost \$1000	Base		+550	

TABLE 7-7 ALTERNATE FEEDWATER HEATING FROM STORAGE

Energy Source	Receiver Steam	Receiver Steam & Feedwater Heating From Storage	Receiver Steam & Storage Steam
Throttle Flow, kg/hr	47,583	44,546	44,546
lb/hr	(104,900)	(98,205)	(98,205)
Gross Output, MWe	10.28	10.28	10.28
Condenser Flow, kg/hr	35,801	38,247	36,773
lb/hr	(78,926)	(84,319)	(81,070)
Condenser Pressure, kPa (Abs)	20.3	21.7	20.8
in. hg (Abs)	(6)	(6.4)	(6.15)
Net Heat Rate J/kW hr	1.26×10^7	1.33×10^7	1.29×10^7
Btu/kW hr	11,969	12,562	12,192

7.5.7 Alternate Feedwater Heating from Storage

The feasibility of using storage steam for feedwater heating was reviewed and compared against using steam from storage directly into the turbine. The results are shown in Table 7-7. A base case was developed using steam from the receiver alone. This case resulted in a gross power output of 10.28 MWe and a net heat rate of 12.627×10^6 J/kWh (11,969 BTU/kWh). The flow from the receiver was then reduced and the two high pressure (HP) heaters were isolated. Feedwater heating from storage was then introduced to bring the gross power output back up to 10.28 MWe. The resulting net heat rate was 13.253×10^6 J/kWh (12,562 BTU/kWh). Finally, holding the flow from the receiver constant, reinstating the two HP heaters, eliminating feedwater heating from storage and introducing steam into the turbine from storage to achieve 10.28 MWe gross output resulted in a net heat rate of 12.863×10^6 J/kWh (12,192 BTU/kWh). Energy from storage is therefore more efficiently used by supplying steam to the turbine rather than using it for feedwater heating. Feedwater heating from storage was therefore abandoned.

Efficiencies, flow rates and heat rates shown in Table 7-7 were obtained early in the study and do not necessarily reflect the latest values.

7.5.8 Other Cooling Options

Air cooling was chosen as a baseline approach in view of water shortage problems in the southwestern states. As more information becomes available regarding the specific pilot plant site and its water availability, this heat rejection system approach will be re-evaluated. Although water cooling has not been studied in detail for the pilot plant, Table 7-8 shows a preliminary comparison between dry and wet cooling at typical backpressures of 2.03×10^4 Pa (6 in-Hg) and 0.85×10^4 Pa (2.5 in-Hg) respectively. The wet cooling system uses separate mechanical draft towers with a circulating water system. It results in 960 kW or approximately 10% additional power output, \$445,000 lower capital costs and approximately 2.52×10^{-2} m³/s (400 gpm) additional water consumption. If water availability is determined to be adequate, the above performance and capital cost benefits can be realized by using the wet cooling system. Total plant water consumption is estimated to be 2.61×10^{-2} m³/s (415 gpm) for a wet cooled plant compared to only 9.5×10^{-4} m³/s (15 gpm) for the baseline dry cooled plant.

Another type of heat rejection system that can be considered when water availability is defined is the wet/dry hybrid cooling system shown in Figure 7-13. This system can reduce the makeup water requirements from those required by a totally wet cooled plant and has some potential capital investment savings, compared to the totally dry system. Performance is expected to be similar to a dry cooled plant (i.e., approximately 10% less net kilowatts than a totally wet cooled plant). However, wet/dry towers are

FIGURE 7-13 WET/DRY COOLING SYSTEM AND INSTALLATION

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TABLE 7-8 COMPARISON OF AIR AND WATER COOLING

	Air Cooling @ 20.3 kPa (6 in. hg)	Water Cooling @ 8.5 kPa (2.5 in. hg)	Δ
Unit Gross Generation, kW	10,283	11,183	+900
Cooling System Pumping Power, kW	0	390	-390
Cooling System Fan Power, kW	550	100	+450
Gross Minus Cooling System Power, kW	9,733	10,693	+960
Cooling System Water Requirement, gpm	0	400	-400
Cooling System Capital Costs, Dollars			
Condenser	960,000	100,000	-860,000
Cooling Tower	0	200,000	+200,000
Cooling System Pumps	0	80,000	+80,000
Piping, Valves, Etc	0	35,000	+35,000
Makeup Water System	0	100,000	+100,000
Total	960,000	515,000	-445,000

FIGURE 7-14 TSR RATIO VS. TURBINE INLET PRESSURE

relatively unproven and some fouling of the dry surface by wet system recirculation can be expected.

7.5.9 Inlet Pressure Variations

The baseline conceptual design has been established, as described in the previous discussions of equipment selection, tradeoffs and tentative cooling system selection. However, future tradeoffs during the optimization period will consider variations to the baseline.

One future tradeoff will consider turbine inlet pressure variations. Figure 7-14 shows the effect on the theoretical steam rate (TSR) of varying the turbine main steam and admission inlet design pressures from the baseline values. Assuming the overall turbine efficiency to remain constant and these pressure variations to be acceptable to the turbine manufacturer, the curves provide an indication of the percentage change in heat rate as inlet pressure is varied. A steeper curve exists for admission pressure variations since any given pressure variation is a greater percentage of the available pressure at the admission point as compared to the main steam inlet. Although steam design pressures for the baseline pilot plant have been set at 8.72×10^6 Pa (1,250 psig) and 4.24×10^6 Pa (600 psig) for the main steam and admission steam inlets, a future tradeoff will consider variations on these choices, examining both heat rate change and storage system cost.

7.6 THERMAL TRANSIENTS

The diurnal thermal cycling to which the turbine will be subjected, together with possible additional thermal transient conditions due to intermittent clouds, will impose significant thermal stresses on the turbine which tend to reduce its life. Certain preventative measures can be taken to minimize the damage caused by these thermal transients. These measures are presently employed on peaking units which are often subjected to diurnal cycling similar to that expected in a solar thermal power plant. They include limiting the inlet temperature and pressure, using more ductile turbine materials, modifying the turbine mechanical design, adding more thermal insulation to limit overnight temperature drop, modifying starting procedures and limiting transients during operation.

The conceptual design throttle condition limits of 8.72×10^6 Pa (1,250 psig), 783 K (950°F) have already been discussed as a major step to reduce thermal damage by permitting the use of more ductile steel and reducing the casing thickness. Large base loaded units, by comparison, have considerably less thermal transients to contend with and operate at up to 24.2×10^6 Pa (3,500 psig), 811 K/811 K (1,000°F/1,000°F) steam conditions. The lower 783 K (950°F) temperature can also permit closer wheel spacing with a reduction in shaft span and diameter. Additional gains in reducing the thermal stress in the rotor can be

accomplished by reducing the stress concentrations in those areas which undergo the greatest temperature change. However, some efficiency loss may result from such turbine design changes. These mechanical design changes may be incorporated into the larger first commercial unit where thermal stresses are a more serious consideration.

Startup procedures will be formulated to minimize the diurnal thermal stresses and the plant will be operated to minimize transients during the day. Heavy thermal insulation on the turbine will minimize overnight temperature drop to reduce the resultant startup time.

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 change anticipated to the life expenditure for that change. An example of this type of curve is shown in Figure 7-15 for a large commercial turbine.

For this turbine, up to a 56 K (100°F) step temperature change could be imposed on the turbine repeatedly without any appreciable loss of turbine life. However, an overnight temperature drop of 167 K (300°F) would require one hour to bring the turbine up to temperature (20 year life is equivalent to approximately 6,000 starts or 0.017 percent per cycle which allows a 167 K/h (300°F/h) rate of temperature change for a 167 K (300°F) change). This does not include time to bring the turbine up to speed and synchronize or consideration of other transients and their life reducing effect. A realistic hot startup time for this turbine considering these other factors would probably be between 90-120 minutes.

A long startup time is typical of large, high pressure and temperature machines. The time can be reduced for the first commercial solar thermal unit by incorporating the preventative measures discussed previously. However, due to the more massive parts in the commercial unit compared to the pilot plant, the required startup times would still be greater than those for the pilot plant discussed below.

Figure 7-16 shows the corresponding turbine cyclic life expenditure curve for a GE admission turbine of the type and size suitable for the pilot plant. Comparing this with Figure 7-15, it can be seen that this smaller turbine is more amenable to thermal cycling.

Up to a 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 (389 K/h (700°F/h) rate of change for 0.017 percent per cycle). A realistic hot startup time (considering time to bring the turbine up to speed and

FIGURE 7-15 TYPICAL COMMERCIAL TURBINE CYCLE LIFE EXPENDITURE

FIGURE 7-16 RECOMMENDED TEMPERATURE RAMP RATE VS. TEMPERATURE CHANGE USING FIRST STAGE INNER SHELL THERMOCOUPLE

synchronize and 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.

The turbine life expenditure curves together with information on expected turbine temperature drop versus time after shutdown, and consideration of other transients to be imposed on the turbine throughout its life, are used by the manufacturer to derive recommended startup rates and curves for each turbine.

Hot startup rates are shown in Figure 7-17 based on standard turbine data sheets from General Electric applicable to morning startup of the pilot plant turbine after an overnight shutdown. Allowable rates of increase include 500 rpm per minute from turning gear speed to 1,000 rpm, a five minute hold at 1,000 rpm, 500 rpm per minute from 1,000 rpm to 3,600 rpm and loading to full load (after synchronizing) at three percent per minute.

7.7 MODES OF OPERATION

The EPGS can be operated in the following modes:

- 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.

Overnight shutdown is essentially a hot standby mode with the turbine on turning gear and with the mechanical vacuum pump maintaining vacuum in the condenser. This vacuum is maintained to minimize corrosion and the possibility of excessive corrosion product buildup in the receiver after startup. Turbine sealing steam is required since the vacuum extends into the turbine, the extraction lines and the closed feedwater heater shells. Approximately 680 kg/h (1,500 lb/h) of sealing steam and 680 kg/h (1,500 lb/h) of deaerating steam will be provided from the

FIGURE 7-17 PREDICTED TURBINE STARTING AND LOADING CURVES-HOT START

storage, and deaerated condensate will be returned to it by a booster pump.

Startup of the EPGS in the morning will be accomplished after the receiver has established a significant flow to the TSS at rated pressure and temperature. Receiver steam will be diverted to the EPGS at a controlled rate to bring the turbine up to speed, synchronize and apply load.

The operational modes all include power generation using steam from the receiver and/or storage. Steam flow is controlled by the turbine-generator master control system to maintain constant main steam upstream pressure and/or constant power output depending on the mode of operation. For example, for the "operating from the receiver alone" mode, the main steam control valve is on upstream pressure control and the admission steam control valve is wide open with the admission steam stop valve closed; for the "operating from the receiver while charging storage" mode, the main steam control valve is on power output level (or load) control with no change in the admission steam valve positions; for the "operating from the receiver and storage" mode, the main steam control valve is on upstream pressure control, the admission steam stop valve is open and the admission steam control valve is on load control; for the "operating from storage alone" mode, the main steam stop valve is closed and the admission steam control valve is on load control.

Prior to sunset, the EPGS will be placed in the "operating from the receiver and storage" mode. Steam will be automatically drawn from storage to maintain output as the receiver steam flow drops to zero. The plant can be shut down in the evening at a controlled rate in the "operating from storage alone" mode.

The "receiver charging storage alone" mode is basically not an EPGS mode of operation. However, the EPGS feedwater pumps will be provided to return TSS condensate to the receiver and to provide deaerated, demineralized and pressurized makeup water.

7.8 FLOW DIAGRAM

Figure 7-18 shows the main cycle equipment, piping and controls. In order to simplify the diagram, not all valves, piping, instrumentation and other components are shown. Descriptions of the main steam, condensate and feedwater, and other main and auxiliary systems are given in Section 7.9.

Steam flow is controlled by the turbine-generator master control system to maintain constant power output and/or main steam upstream pressure depending on the mode of operation as discussed in Section 7.7.

The cycle pumps and feedwater heaters pressurize and heat the condensate to levels acceptable to the Receiver and Thermal Storage Subsystems. A deaerating heater and full flow condensate

demineralizers are provided to further condition the feedwater to prevent buildup of corrosion products in the boilers.

During the overnight shutdown, a condensate pump, the deaerator and a booster pump are operated at approximately 20% recirculation flow and 1,360 kg/h (3,000 lb/h) of condensate is supplied to the Thermal Storage Subsystem (TSS) seal steam generator where sealing and deaeration steam are generated and returned to the EPGs.

Feedwater is supplied to the TSS by the low pressure (LP) boiler feedwater pumps when the TSS is generating steam. The same line returns condensate from the TSS to the suction of the HP boiler feedwater pump when the TSS is being charged. A small high pressure feedwater flow is supplied to the TSS desuperheater when the TSS is being charged.

Hotwell level control is accomplished using condensate reject and makeup lines to and from the condensate storage tank. Deaerator level control is accomplished using a control valve in the main condensate line. The two high pressure heaters drain via drain traps and the low pressure heater uses a loop seal. All pumps are provided with minimum flow recirculation protection.

7.9 MECHANICAL SYSTEMS

The major and auxiliary systems of the EPGs are:

- Main Steam
- Feedwater Heater Extraction, Drains and Vents
- Condensate and Feedwater
- Demineralized Water Supply
- Turbine Steam Seals
- Turbine-Generator Lube Oil
- Condenser Air Removal
- Turbine Control Oil
- Closed Cooling Water
- Compressor Air
- Fire Protection
- Domestic Water
- Cranes and Hoists
- Sanitary Drainage
- Acid Waste
- Oily Waste

System descriptions have been developed for these systems, the more significant of which are summarized below. These systems will be further defined during the preliminary design stage.

7.9.1 Main Steam

The main steam system shown in Figure 7-19 receives steam from the Receiver and/or the Thermal Storage Subsystem, controls the flow rate of that steam to the turbine, the turbine bypass lines,

and various auxiliaries and converts the thermal energy of the steam to mechanical energy in the turbine.

Both the Receiver and the Thermal Storage Subsystems are connected to the main steam system by eight inch diameter lines located above grade outside the turbine-generator mat. [Pipe sizes referred to here and elsewhere in this report are nominal sizes for specification purposes and do not refer to exact dimensions. Metric equivalents are therefore not given.] Stop and control valves are included in both steam lines to control steam flow to the turbine. Six inch diameter branch lines connect each of these steam lines to turbine bypass valves adjacent to the main condenser. During startups, shutdowns and other transients, the turbine bypass system will be used as required to stabilize steam conditions at the turbine inlets. Four inch diameter branch lines connect each of these steam lines to the pressure reducing station. The reducing station provides steam for turbine sealing and also to the deaerator when extraction steam pressure is inadequate.

Above-seat drains for each of the stop and control valves, and below-seat drains for each of the control valves are connected to the main condenser hotwell through separate one and one half inch lines with automatic valves at the downstream end.

The turbine is provided with four extraction steam connections as follows: a four inch connection for the first high pressure heater, a five inch connection for the second high pressure heater, an eight inch connection for the deaerator and a twelve inch connection for the low pressure heater.

The turbine exhaust is connected to the main condenser by two exhaust ducts. The low points of the exhaust ducts have one and one half inch drain lines routed to a drain sump.

7.9.2 Feedwater Heater Extraction, Drains and Vents

The feedwater heater extraction, drains and vents system shown in Figure 7-20 extracts steam from four stages of the main turbine, heats and deaerates the feedwater with this steam, provides venting of non-condensibles to the condenser and returns the resulting condensate drains to the power cycle.

The closed feedwater heaters are connected to the turbine extraction points through free swing extraction non-return (check) valves and isolation gate valves to prevent wet steam backflow into the turbine and to permit isolation of the feedwater heaters for repair. The two high pressure heater shells drain to their next lower pressure heater via two inch diameter drain lines equipped with steam traps. Two inch diameter branch lines from upstream of the traps connect to the main condenser hotwell through automatic high level dump valves. The deaerator extraction line has a check valve and a power operated non-return valve but no isolation valve since the plant will not be operating if the deaerator is out of service.

FIGURE 7-19 MAIN STEAM

FIGURE 7-20 FEEDWATER HEATER EXTRACTION DRAINS AND VENTS

The shell side drains outlet of the low pressure heater is connected to the main condenser by a four inch diameter loop seal equipped with an isolation gate valve.

All three shell and tube heaters are provided with shell relief valves designed for ten percent of the feedwater flow. The deaerator is provided with a relief valve designed for the maximum possible auxiliary steam flow.

Feedwater heaters 1, 2 and 4 are connected to the main condenser with separate one inch diameter vent lines equipped with globe valves. The deaerating heater is vented to atmosphere through a two inch diameter line and valve.

Auxiliary steam from the receiver or thermal storage is used as required to provide adequate condensate deaeration during startup and at low unit loads.

7.9.3 Condensate and Feedwater

The condensate and feedwater system shown in Figure 7-21 delivers condensate from the main condenser to the steam generator(s) as properly heated, deaerated, chemically treated and pressurized feedwater.

The condensate drains from the air cooled main condenser to the hotwell which is vented back to the air removal sections of the condenser. The hotwell is connected to two parallel 100% capacity condensate pumps by six inch diameter piping through suction strainers. The discharge of the condensate pumps is routed through the steam packing exhaustor and the low pressure heater to the deaerator through a three inch diameter pipe.

A two inch diameter condensate pump recirculation line is routed to the hotwell from a point downstream of the steam packing exhaustor. Another two inch diameter branch line from this point is connected to the condensate storage tank (reject line), the chemical injection tank and the attemperators of the turbine bypass valves and the steam pressure reducing valves. A deaerator level control valve is located in the condensate line just upstream of the deaerator.

The outlet of the deaerator is connected to the suction of the two 100% capacity booster pumps by a six inch line. The discharge of the booster pumps is routed through two 100% capacity powdex cation demineralizers in parallel on three inch lines. A demineralizer backwash line is routed through a spent resin collection strainer to the makeup demineralizer system. Hydrazine and ammonia are injected into the booster pump suction as required for oxygen scavenging and pH adjustment.

From the discharge of the demineralizers, the condensate passes through two high pressure heaters and one of two 100% low pressure boiler feedwater pumps. These pumps deliver feedwater

FIGURE 7-21 CONDENSATE AND FEEDWATER

FIGURE 7-22 DEMINERALIZED WATER SUPPLY

to the Thermal Storage Subsystem and/or the receiver/steam generator via one of the two 100% capacity high pressure feedwater pumps. A line from the discharge of the high pressure feedwater pumps to the Thermal Storage Subsystem provides desuperheating water as required.

All closed feedwater heaters and the steam packing exhausters are provided with bypass lines to be used in the event of tube leakage and other similar problems.

7.9.4 Demineralized Water Supply

The demineralized water system shown in Figure 7-22 supplies the condensate deaerator with sufficient water of acceptable quality to meet the makeup requirements of the power cycle.

Water is assumed available from a deep well located near the plant and is connected to the demineralizer system by a two inch diameter buried steel pipe and a well water filter. The outlet of the filter is connected in series to the cation demineralizer, the anion demineralizer and the mixed bed demineralizer by a two inch diameter rubber lined steel pipe.

In normal operation, the well water pump delivers approximately $8.8 \times 10^{-4} \text{ m}^3/\text{s}$ (14 gpm) through the system to the condensate deaerator. The water is introduced into the power cycle as deaerated condensate and passes through the steam generator and turbine and is rejected to condensate storage by the hotwell level control system. Starting and stopping of the well pump is remote manual as required by the condensate storage tank level indications.

If the demineralizer beds become depleted, the beds are backflushed and automatically regenerated by the running of the acid and caustic pumps and the automatic sequencing of valves. The regenerating chemicals are automatically diluted by a preset condensate flow. Regeneration effluent is dumped to the neutralizing tank and drained to waste after being sampled and checked neutral.

A redundant well and well water pump will be provided. Although not shown in Figure 7-22, a redundant demineralizer train will also be included to ensure continuous service during equipment and control valve outages.

7.9.5 Closed Cooling Water

The closed cooling water system shown in Figure 7-23 provides cooling for various components of the main power cycle and its auxiliary systems.

During normal operation only one of the closed cooling water pumps is running. Water is being delivered to both of the air compressors with one of the compressors running and the other on standby. Water is supplied to one turbine lube oil cooler, one

FIGURE 7-23 CLOSED COOLING WATER

FIGURE 7-24 ELECTRICAL SYSTEM FUNCTIONAL DIAGRAM

turbine control oil cooler, one feedwater pump lube oil cooler and both main generator air coolers. Sample coolers are in service as required by samples being taken.

Both halves of the closed cooling water section of the air cooled main condenser are normally in service for maximum cooling. Each half, however, is served by a separate fan and can carry the full cooling load if it becomes necessary to remove one half from service.

The surge tank provides for temperature swells in the system volume and maintains suction pressure on the pumps. Water is made up to the surge tank through a manual valve from condensate storage when required as indicated by a low level alarm. The second cooling water pump is started automatically by a low pressure switch on the cooling water supply header. The minimum recirculation line protects the pump if started against a closed valve.

Other minor cooling loads will be provided with closed cooling water but are not shown on the simplified schematic.

7.10 ELECTRICAL SYSTEM

The functions of the plant electrical system, as shown schematically in Figure 7-24, are:

- To provide power for the plant auxiliary loads during morning startup and overnight shutdown.
- To generate electrical energy from the plant heat sources, supply the auxiliary loads and transmit plant net generation to the utility during normal operations.
- To supply essential equipment loads when connection to the utility grid is lost.
- To provide emergency power for personnel safety and for protection of vital equipment when all other power sources are not available.

On plant startup, 1,000 kWe of power is drawn from the utility through the main transformer and the transformer circuit breaker. The main generator is then brought up to rated voltage and speed. After it is synchronized with the 13.8 kV main bus, the generator breaker is closed thereby paralleling the unit with the utility power sources. The generator may then be controlled to supply the auxiliary loads and transmit net power to the utility. The reverse of the above process is done for plant shutdown.

The turbine-generator will be specified to remain in operation in the event of a load rejection. In the case of an electrical fault which disconnects the 13.8 kV bus from the utility, the generator will continue to deliver power to the auxiliaries until

it is manually shut down. An automatic starting of the diesel generator will occur whenever power to the 13.8 kV bus is interrupted. During the diesel generator startup period of about ten seconds, the battery supplies emergency lighting, control and instrumentation loads. 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.

A single line diagram of the EPGS electrical system is shown in Figure 7-25. It consists of a 12.5 MWe generator, 13.8 kV switchgear, a 12 MVA, 13.8-34.5 kV step-up transformer and a distribution system for the plant auxiliary loads. The auxiliary distribution system designed for a normal load of 1,250 kVA, is composed mainly of two 750 kVA, 480 V double-ended load center unit substations and a number of 480 V motor control centers. In addition, a 250 kW, 480 V standby generator connected to a load center bus and a 125 V dc battery system are provided to supply the standby and emergency power loads.

Startup power is fed to the auxiliary distribution system through the 13.8 kV bus with the main generator breaker open. As soon as the main generator has been brought up to rated speed and voltage and synchronized to the bus, the generator breaker may be closed. The generator may now be controlled to supply power to the off-site system and feed power to the auxiliary loads.

From the 13.8 kV bus, voltage is stepped down to and distributed at 480 V by the two load center unit substations. Loads for above 75 kW (100 hp) to 187 kW (250 hp) are connected directly to the load center feeder breakers. The majority of the loads, however, are of 75 kW (100 hp) or less and are connected to motor control centers which, in turn, are supplied from the load centers.

Control power, lighting and other miscellaneous small loads are supplied at 120 V through distribution transformers fed from the motor control centers. Emergency dc power for controls, including switchgear control, diesel generator field flashing, emergency lighting and the emergency lube oil motor driven pump is taken from the battery. In addition, power for the dc to ac inverter serving the plant computer and the public address system is provided by the battery. Normal dc power, however, is supplied by a battery charger, with a battery takeover on loss of ac supply.

The plant electrical system switchyard/substation shown in Figure 7-26 interfaces between the 34.5 kV transmission system of the utility and the plant main generator and auxiliary distribution system. It consists of a 600 A incoming load break switch, a 34.5 GND Y/19.9-13.8 kV substation transformer, a 13.8 kV, 500 MVA switchgear, a steel structure and a grounding system.

The incoming 34.5 kV line is connected to the load break switch which is tower mounted at 6.1 m (20 ft) above ground level. The load side of each switch pole connects to a lightning arrester

FIGURE 7-25 ELECTRICAL SYSTEM-SINGLE LINE DIAGRAM

FIGURE 7-26 ELECTRICAL SYSTEM-SWITCHYARD/SUBSTATION

and a fuse which protect the transformer against voltage surges and internal faults.

The transformer is rated at 12.0 MVA and is oil immersed self-cooled. It is provided with a bushing for each line terminal and the primary neutral where the unit is solidly grounded. The unit is provided with cooling fins for natural air cooling, a no-load tap changer, fault pressure relays and other auxiliaries and components.

The 13.8 kV switchgear is of the outdoor, aisle type and contains one incoming, one generator, and two feeder breaker sections. Connection from the transformer secondary terminals to the switchgear incoming section is by overhead exposed buses. Connections from the switchgear to the plant electrical equipment are by means of insulated cables through an underground duct bank. Relaying and metering equipment are mounted on the front panel of each switchgear section.

A ground grid is provided for the switchyard/substation consisting of uninsulated copper cables and rods buried beneath the area. All system and equipment grounding at the switchyard are connected to the grounding system.

7.11 LAYOUT

7.11.1 EPGS Structure and Layout

Figure 7-27 shows an artist's conceptual view of the EPGS structure. Plant main and admission steam lines are shown terminated at their respective stop valves. In reality, these lines will connect to elbows and will run east to the tower. The electrical switchyard is not shown but will be located to the north of the EPGS structure. Six 6.4 m (21 ft) diameter fans draw cooling air through the air cooled condenser unit located above the turbine-generator area.

The EPGS plant arrangement is shown in Figure 7-28. 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 and pumps, and other miscellaneous turbine related equipment. An elevated turbine-generator deck at elevation 3.35 m (11 ft 0 in) extends around to the north side of the plant covering an enclosed area for plant electrical switchgear, motor control centers and the diesel generator installation. Stairs are provided to the air cooled condenser deck at elevation 11.6 m

FIGURE 7-27 EPGS STRUCTURE

(38 ft 0 in) to allow access to the condenser and deaerator for inspection and maintenance.

The major mechanical equipment, including cycle pumps and feedwater heaters, is located on the lower deck at elevation 0 m (0 ft 0 in). Other smaller mechanical equipment such as the air compressors, vacuum pumps, and chemical injection pumps are not shown in the layout, but will be located on the lower deck or the turbine-generator deck along the north side.

A 45,000 kg (50 ton) crane is provided for dissassembly of the turbine-generator unit including removal of the generator rotor.

7.11.2 Control and Administration Building Layout

The control and administration building provides a central location for the master control system and also provides office space for the plant personnel.

As shown in Figure 7-29, the control room is located in the north-east upper corner of the building and provides an excellent view of the EPGS structure, the tower and the receiver, with the mirror field extending beyond. The control room operators can observe the plant as they perform manual control operations. Access is provided to the roof where cloud observations and insolation measurements can be made. The master computer, a card reader, a card keyboard and a line printer are also located in the control room.

The battery room is located in the control and administration building adjacent to the cable spreading room. Offices, laboratory and washroom space is provided for the plant personnel and an assembly/meeting room is provided for visitors.

7.12 RELIABILITY AND FAILURE MODES AND EFFECTS ANALYSIS

A reliability and failure modes and effects analysis (FMEA) was performed for the critical systems and components of the EPGS. Based on this analysis, redundancy was included in critical areas where outages would significantly affect the overall plant availability. Sources of reliability information included various published articles, information from manufacturers, and estimates based on Bechtel in-house reports.

Availabilities were obtained for the critical EPGS systems and components and are summarized in Table 7-9. A factor of 0.005 was applied for miscellaneous EPGS outages not covered in the list. The availability of the EPGS will be almost identical regardless of the steam supply source and the data was prepared only for the case when the receiver is supplying steam. Availability values reflect forced outages only and not scheduled outages intended for maintenance. Overall availability is almost identical for both a water cooled plant and the baseline

FIGURE 7-29

[illegible]

TABLE 7-9 EPGS AVAILABILITY SUMMARY

EPGS SYSTEM	AVAILABILITY
TURBINE AND GENERATOR	.985
CLOSED COOLING WATER SYSTEM	.9969
COORDINATED CONTROL SYSTEM	.9963
DEMINERALIZED WATER SUPPLY SYSTEM	.9989
AIR COOLED MAIN CONDENSER	.997
CONDENSATE AND FEEDWATER	.9986
EXTRACTION STEAM AND	
FEEDWATER HEATER DRAINS	.9988
POWER GENERATION AND TRANSMISSION	.9978
MAIN STEAM AND TURBINE BYPASS	.9982
CONDENSER AIR REMOVAL SYSTEM	1.0
COMPRESSED AIR SYSTEM	.999
TOTAL	.9669
CORRECTION FOR OMISSIONS	.995
OVERALL EPGS AVAILABILITY	.9620

SUBSYSTEM MODEL INPUT DATA:

REF	UNIT NOMENCLATURE	DUTY CYCLE	LIFE EST	MTBF	MTTR	AVAILABILITY
A	CONDENSATE PUMP	CONT.	30 YRS			.999975 (1)
D	FEEDWATER HEATER + BYPASS	CONT.	30 YRS			.9999 (2)
C	DEAERATOR	CONT.	30 YRS			.999
D	BOOSTER PUMP	CONT.	30 YRS			.999975 (1)
E	CONDENSATE DEMINERALIZER	CONT.	30 YRS	4000 HRS	24 HRS	.999964
F	FEEDWATER HEATER + BYPASS	CONT.	30 YRS			.9999 (2)
G	FEEDWATER HEATER + BYPASS	CONT.	30 YRS			.9999 (2)
H	LOW-PRESSURE FEEDWATER PUMP	CONT.	30 YRS			.999975 (1)
I	HIGH-PRESSURE FEEDWATER PUMP	CONT.	30 YRS			.999975 (1)

NOTES: (1) TWO PUMPS IN PARALLEL, EACH FULL CAPACITY, PERMIT FULL LOAD TO BE CARRIED WITH ONE PUMP OUT OF SERVICE.

(2) IF THE FEEDWATER HEATER IS BYPASSED, THERE IS SOME REDUCTION IN THERMAL EFFICIENCY.

H
π
A
P₁ = .9986

FIGURE 7-30 CONDENSATE AND FEEDWATER SYSTEM AVAILABILITY

air cooled plant. A wet/dry cooled plant would have a somewhat lower availability. Overall EPGS availability is 96.2%.

Figure 7-30 shows an example of a typical system availability analysis. All pumps are full capacity and are provided with 100% standby to permit full load to be carried with one pump out of service. All closed feedwater heaters are provided with bypasses to permit operation when a feedwater heater is out of service. Some load reduction will result due to a reduction in thermal efficiency with less feedwater heating. Final feedwater temperature will only be significantly affected if the high pressure feedwater heater is bypassed. The polishing demineralizers are also provided with 100% standby. Overall system availability is 0.9986. Mean time between failure (MTBF) and mean time to repair (MTTR) information is included where available.

7.13 OPTIMIZATION STUDIES

On completion of the EPGS conceptual design, optimization studies have recently been initiated. These studies will take a second look at some of the choices made in the conceptual design and will refine the baseline plant in preparation for the preliminary design effort to be initiated next fall. Information from the subsystem experiments will be factored into the baseline design as will specific site considerations once a pilot plant site is chosen.

Examples of major activities which may be performed during this optimization period are:

- Compare wet, dry and wet/dry cooling options based on a specific site.
- Compare high and low backpressure turbine overall performance and economics.
- Cooperate with Martin Marietta in more detailed studies varying the turbine inlet pressures to optimize the overall plant; e.g., possibly increasing main steam pressure to 10.1×10^6 Pa (1,450 psig) and/or decreasing the admission pressure to 3.55×10^6 Pa (500 psig).
- Optimize the number of feedwater heaters.

Results of these optimization studies will be reported in future progress reports.

Section VIII

PILOT PLANT ARCHITECTURAL-ENGINEERING AND SUPPORT

8.1 INTRODUCTION

In parallel with Bechtel's prime effort in the conceptual design of the Electrical Power Generation Subsystem (EPGS) as described in Chapter VII, other architectural-engineering tasks were also conducted. These include the conceptual development of an integrated plot plan for the entire pilot plant, and conceptual designs of portions of the Collector Subsystem and the Receiver Subsystem. In addition to these efforts, work also commenced on an initial environmental analysis of the pilot plant. Results of these activities are summarized in this Chapter.

8.2 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 - shows the additional facilities required for a functional plant. The conceptual plot plan is shown in Figure 8-1.

8.2.1 The Plot Plan

Since no site has been selected for the plant, it was assumed that the site is level, requiring no major grading.

The collector field versus the receiver tower positioning is 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. Minimizing pipe run lengths was also given serious consideration.

Transmission lines were assumed to leave the plant in a due westerly direction, connecting the plant with a utility network (not shown).

The sewage treatment plant and the thermal storage medium spill retention pond were located south of the plant, assuming a natural site slope toward that direction.

The main plant access road was assumed to enter the plant through the western plant fence line, and continue in a due easterly direction through the plant. This road has 7.3 m (24.0 ft) wide asphalt concrete paving with 1.2 m (4.0 ft) wide asphalt on crushed rock base concrete paved shoulders on either side, accommodating two-lane vehicular traffic. It will be constructed in conformance to Federal Highway Specifications.

Internal plant roads form one loop to encircle the receiver tower, the Electrical Power Generation Subsystem, the control and administration building and the heat exchanger pads of the Thermal Storage Subsystem on the north side; and another loop encircles the thermal storage tanks on the south side of the main access road. These plant roads have a 3.7 m (12.0 ft) wide asphalt concrete paving on crushed rock base and 1.2 m (4.0 ft) wide asphalt concrete paved shoulders. They are designed for single lane, one-way vehicular traffic.

The periphery road and the north-south center road of the Collector Subsystem field provide main access to the field. These roads have 3.7 m (12.0 ft) wide crushed rock paving with rolled in filler, and 0.6 m (2.0 ft) wide crushed rock paved shoulders. In addition, there is one access road for every other heliostat row between rows 1 and 13, and one to each heliostat row between rows 13 and 33, to provide access to individual heliostats and sensors.

Details of site drainage cannot be designed until general site topography and runoff data become available. In view of the extensive area required by the Collector Subsystem, careful considerations should be given to this design, especially if the plant is to be located in an arid desert area of the Southwest, where flash flooding is not an uncommon phenomenon.

The method of supplying process and fire water to the plant cannot be determined at this time. For the present conceptual design stage, it has been assumed that water in adequate quantity and satisfactory quality is obtainable from deep wells. Their assumed location is east of the plant site.

A 1.8 m (6.0 ft) high chain link fence with three strands of barbed wire on top enclose the $3.93 \times 10^5 \text{ m}^2$ (97.3 acre) plant area. Personnel and vehicular gates provide access through this fence and are controlled at the gate house.

8.2.2 General Plant Arrangement

Figure 8-2 shows the details of the major 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 at the origin of the plant grid system. The EPGS plant is located symmetrically about the east-west centerline of the receiver tower on the west of it. It is an open steel structure erected on a reinforced concrete slab on grade, 25.9 m (85.0 ft) along the east-west length and 19.8 m (65.0 ft) along the north-south width. The eastern edge of its foundation pad is 28.2 m (92.5 ft) west of the centerline of the receiver tower, providing space for the expansion loops in the steam and feedwater pipings (not shown here). Road access to this facility is provided on the west side of it.

LEGEND

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

FIGURE 8-1 PAGE 8-3

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11-2-75 ISSUED FOR CONCEPTUAL DESIGN

NO.	DATE	REVISIONS	BY	CHKD	DESIGN	ENGR	APPV

SCALE 1" = 150'

BECHTEL
SAN FRANCISCO

CENTRAL RECEIVER SOLAR THERMAL
POWER SYSTEM PHASE NO. 1

PLOT PLAN

JOB No.	DRAWING No.	REV.
11480	11480-008-C2

FIGURE 8-2 PAGE 8-4

[illegible]

The switchyard/substation is located due north of the EPGS plant. It has a fenced-in area of 12.8 m (42.0 ft) long in east-west by 9.1 m (30.0 ft) wide in north-south. Road access is provided on the north side. Transmission lines running due west connect the substation with a utility grid system to be determined when a site is known.

The 378.5 m³ (100,000 gallon) capacity condensate storage tank is located north of the turbine generator and west of the substation. This location minimizes the piping to the condenser.

The control and administration building is shown as a 12.2 m (40.0 ft) by 18.3 m (60.0 ft) two-storied structure which is set back from the road centerlines on the south and west side by 13.7 m (45.0 ft) to allow for landscaping.

Located immediately to the southeast of the receiver tower are the heat exchanger pads No. 1 and No. 2, both being north of the east-west plant road. They are located to minimize the piping interface with both the receiver tower and the EPGS.

The tank farm of the Thermal Storage Subsystem 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.

The area designated as shops, maintenance and fire truck garage is located on the south side of the main east-west plant road, opposite the control and administration building. The "shops" portion will provide space for the maintenance requirements such as machine and electrical shops while the "maintenance" area will accommodate maintenance work on plant components such as heliostats.

A 2,460 m³ (650,000 gallon) capacity storage tank is provided for the storage of fire water. It is located east of the receiver tower. A water supply line from a well field enters the storage tank from the east. Two fire booster pumps will be located in a pump house south of the tank. One complete fire water loop encircles the hydrocarbon tank farm with a branch extending to the Hitec tank farm. Foam generating equipment will be provided for in this loop. A second loop serves the rest of the plant.

Other facilities shown in Figure 8-2 include a guard house outside of the plant fence, on the south side of the main access road, and a parking area for 40 vehicles. This area is asphalt paved and is located north side of the main plant road.

The main objective of this effort was to develop a cost-effective foundation design of the large number of 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 developed nine design alternatives for the heliostat foundation, and relative costs were determined for these alternatives. This resulted in a preliminary selection of a preferred foundation, the conceptual design of which was then completed by incorporating the associated external electrical junction box and wiring.

8.3.1 Design Criteria and Data

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

The foundation was designed to support the heliostat to operate at 13.4 m/s (30 mph) steady wind and at gusts up to 17.9 m/s (40 mph). When wind speed reaches 22.4 m/s (50 mph), the heliostat will be stowed for protection; and in the stowed position it will survive a wind speed up to 53.6 m/s (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.

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 southwestern U.S. For the purpose of this conceptual design stage, it is 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 is taken to be 191,500 Pa (4,000 lb/ft²), an average value for these soil types.

A flat site is adopted as a baseline for the conceptual design of the collector field.

Based on these environmental design criteria, the following design data were developed for further specifying the design requirements of the foundations. The axial load is the self weight of the heliostat or the sensor; the shear force and the bending moment were determined by Martin Marietta in wind tunnel testing. An allowable vertical wobble of 0.291 mrad (1 arc minute) was allocated for foundation movements. The fine tracking sensor foundation is designed for the 53.6 m/s (120 mph) wind survival condition and the assigned shear force and bending moment were calculated values.

Upright Heliostat at Operating Condition:

Axial Load = 20,016 N (4,500 lb)
Shear Force = 4,793 N (1,077.5 lb)
Bending Moment = 18,777 J (13,850 ft-lb)
at 13.4 m/s (30 mph) wind
Bending Moment = 33,365 J (24,610 ft-lb)
at 17.9 m/s (40 mph) gust
Vertical Wobbling = 0.291 mrad (1 arc min)

Upright Heliostat at Survival Condition:

Axial Load = 20,016 N (4,500 lb)
Shear Force = 14,145 N (3,180 lb)
Bending Moment = 51,356 J (37,880 ft-lb)

Stowed Heliostat at Survival Condition:

Axial Load = 20,016 N (4,500 lb)
Maximum Uplift
Force = 11,120 N (2,500 lb)
Bending Moment = assumed to be less than
51,356 J (37,880 ft-lb)

Upright Sensor at Survival Condition:

Axial Load = 445 N (100 lb)
Shear Force = 445 N (100 lb)
Bending Moment = 1,898 J (1,400 ft-lb)

8.3.2 Design Alternatives

Nine alternative designs for the heliostat foundation and one sensor foundation design common to all heliostat-sensor combinations were developed. These are described in this section.

Type 1 Mat Foundation for Heliostat and Sensor. The circular mat heliostat foundation shown in Figure 8-3 is a 1.83 m (6.0 ft) diameter by 1.07 m (3 ft 6 in) thick cast-in-place concrete block resting below the grade level. For temperature and shrinkage stresses, the mat foundation is nominally reinforced with preassembled steel reinforcing bars. Under the actions of unsteady wind forces on the heliostat, the foundation will be forced into a rocking motion about an axis perpendicular to the direction of wind. Based on the assumed soil bearing capacity and operating design forces, the vertical wobble angle due to rocking is estimated to be 0.163 mrad for a wind velocity of 13.4 m/s (30 mph) and 0.291 mrad for gusts up to 17.9 m/s (40 mph). At a wind velocity of 17.9 m/s (40 mph), the factor of safety against overturning is 1.63, which is only slightly higher than the minimum recommended value of 1.50. Therefore, even if the allowable foundation wobble angle were increased to greater than 0.291 mrad, the weight of the mat foundation could not be significantly reduced without causing unacceptable overturning conditions.

The sensor foundation is a circular concrete mat which is 1.07 m (3 ft 6 in) in diameter by 0.76 m (2 ft 6 in) thick and is

FIGURE 8-3 TYPE 1 MAT FOUNDATION FOR HELIOSTAT AND SENSOR

FIGURE 8-4 TYPE 2A DRILLED PIER AND GRADE BEAM FOUNDATION FOR HELIOSTAT

reinforced for shrinkage and temperature stresses. This sensor foundation has been adopted in conjunction with all of the various types of heliostat foundations investigated in this study, and its cost is included in all heliostat foundation cost estimations.

The circular mat foundation for the heliostat has the lowest cost of all heliostat foundations considered and is therefore taken as the base (unity) for the cost-ratio comparison study.

Type 2A Drilled Pier and Grade Beam Foundation for Heliostat.

Figure 8-4 shows the type 2A heliostat foundation. It is a 0.61 m (2.0 ft) diameter by 3.05 m (10.0 ft) long drilled and cast-in-place reinforced concrete pier. Pier reinforcing steel is preassembled for placement in previously drilled holes. In order to prevent excessive vertical rotation due to the applied loading, the tops of adjacent piers are connected with 0.41 m (1 ft 4 in) by 0.41 m (1 ft 4 in) precast concrete grade beams in the form of a grid, as shown in the figure.

The vertical wobble angle of the top of the pier is 0.10 mrad at 13.4 m/s (30 mph) wind and 0.19 mrad for gusts up to 17.9 m/s (40 mph).

The cost-ratio is 3.89 with respect to type 1 foundation.

Type 2B Drilled Pier and Grade Beam Foundation for Heliostat.

Since the cost of type 2A was found to be high, an attempt was made to modify its design to reduce the cost. This was done by trimming the size of the pier to a diameter of 0.38 m (1 ft 3 in) and the cross section of the grade beam to 0.3 m (1.0 ft) by 0.3 m (1 ft). This resulted in type 2B, which has a higher wobble angle of 0.65 mrad at 13.4 m/s (30 mph) wind and 1.16 mrad at gusts up to 17.9 m/s (40 mph). However, it was found that the cost ratio was reduced only slightly to 3.55.

Type 3A Grade Beam Grid Foundation for Heliostat. Figure 8-5 shows this foundation type which consists of cast-in-place reinforced 0.61 m (2.0 ft) by 0.76 m (2 ft 6 in) concrete beams laid just below the grade in a triangular grid. Heliostats are installed at the intersection of the beams. This approach integrates the heliostat foundations into one structural system.

The maximum wobble angle is estimated to be 0.22 mrad for wind velocity of 13.4 m/s (30 mph) and 0.33 mrad for gusts up to 17.9 m/s (40 mph).

The cost-ratio is 4.02 with respect to type 1 foundation.

Type 3B Grade Beam Grid Foundation for Heliostat. Again, an attempt was made to reduce the cost of type 3A by trimming the grade beam dimensions to 0.61 m (2.0 ft) by 0.61 m (2.0 ft) to become type 3B. This resulted in a higher wobble angle of 0.44 mrad at 13.4 m/s (30 mph) wind and 0.78 mrad at gusts up to

FIGURE 8-5 TYPE 3A GRADE BEAM GRID FOUNDATION FOR HELIOSTAT

FIGURE 8-6 TYPE 4 PILE FOUNDATION FOR HELIOSTAT

17.9 m/s (40 mph). However, the cost ratio was only slightly reduced to 3.82.

Type 4 Pile Foundation for Heliostat. This foundation is shown in Figure 8-6. It consists of four piles with a 1.22 m (4.0 ft) square by 0.76 m (2 ft 6 in) thick reinforced concrete pile cap. The length of each pile is determinable only by actual pile driving test and is tentatively assumed to be 6.1 m (20.0 ft) for cost estimation purposes. Lacking soil test data and actual pile driving data, the wobble angle cannot be estimated. On the basis of this tentative design, the cost-ratio is estimated to be 2.57 with respect to the type 1 foundation. This foundation type is site sensitive and may not be suitable for sites where soil is primarily medium dense or dense sand because the vibration produced by the pile driver compacts the sand causing it to resist the penetration of the piles beyond a few feet into the soil. In view of these considerations, its applicability and cost can only be considered as tentative at this time.

Type 5 Precast Concrete Anchor Block with Explosive Earth Anchor. This foundation, as shown in Figure 8-7, is a precast concrete block in the shape of a truncated pyramid on a triangular base. Pipe sleeves are embedded in the block for inserting the anchors. The block is positioned on stripped and levelled ground at the location of a heliostat and is anchored by three Martin-Marietta Aluminum earth anchors. After the anchors are installed through the pipe sleeves, they are tensioned by turning the tension-bolts. By tensioning the anchors with proper force, the vertical wobble angle can be well restrained. This type of foundation is suitable for locations where high anchorage force (tensile) can be transmitted to the ground without causing upheaval, e.g., a site where there is a rock stratum close to the ground surface. The cost-ratio for this foundation is 1.30 with respect to type 1 foundation.

Type 6 Structural Steel Anchor System with Vertical Augered Caisson Foundation for Heliostat. This type 6 foundation, shown in Figure 8-8, is a shop-fabricated structural steel tripod mounted on three long drilled caissons. The tripod is fabricated with a special type of structural steel called Cor-Ten which does not require painting for protection against corrosion. During the first year, its exposure to weather develops a thin oxide film which protects the base material from corrosion. The structural steel tripod is rigid enough to carry all external load without exceeding the allowable stress limit.

The drilled piers transfer the external loads to the substratum by skin friction along their length and by bearing on soil below the bell.

The vertical wobble angle is negligibly small.

The cost-ratio for this foundation is 1.67 with respect to type 1 foundation.

FIGURE 8-7 TYPE 5 PRECAST CONCRETE ANCHOR BLOCK WITH EXPLOSIVE EARTH ANCHOR FOR HELIOSTAT FOUNDATION

FIGURE 8-8 TYPE 6 STRUCTURAL STEEL ANCHOR SYSTEM WITH VERTICAL AUGERED CAISSON FOUNDATION FOR HELIOSTAT

Type 7 Structural Steel Anchor System with Slanted Augered Caisson Foundation for Heliostat. Figure 8-9 shows this type, which is a variation of the type 6 foundation. In this case, the caissons are drilled at an angle to vertical. Depending on the soil condition, it may be more desirable to use slanting caissons. For illustrative purposes, two slanting angles are shown. The cost ratio of this type is expected to be the same as type 6, i.e., 1.67.

Of the nine heliostat foundation candidate designs described above, the circular mat foundation (type 1) was found to meet all design criteria and to be the lowest in cost. It was therefore chosen as the baseline design in this conceptual design stage. It must be pointed out that this choice is tentative. As site conditions become better defined in the preliminary design stage when a more accurate cost estimation will be conducted, the cost ratios of these foundation types may be subject to change which could lead to the selection of a different foundation candidate design.

8.3.3 Heliostat and Sensor Foundation Conceptual Design

Adopting the type 1 foundation design alternative, the conceptual design for the heliostat and the sensor combination was completed by incorporating the external junction box and the power and control wiring. This conceptual design is schematically shown in Figure 8-10. Each heliostat is connected to an individual electrical tracking sensor as shown in the figure. Three wires (22 gauge) connect the sensor to the junction box at the base of the heliostat. These wires run about 0.61 m (2.0 ft) underground to avoid interference by people or vehicles. The electrical power for each heliostat is supplied through wires to the same junction box.

8.4 RECEIVER TOWER, RISER AND DOWNCOMER CONCEPTUAL DESIGN

This section describes the conceptual design of the receiver tower and the major piping associated with the tower. Since a site was not specified, environmental criteria were hypothesized for this design effort. Further design specifications were provided by Martin Marietta. Based on this information, Bechtel developed three design alternatives for the tower, which consists of the supporting foundation and the tower shaft from ground level to the top platform, but excludes the receiver and its supporting frame. Of these three designs, one was selected for the conceptual design on the basis of cost and performance. In conjunction with the tower design, a conceptual design for the riser and the downcomer was also developed to meet the design specifications given by Martin Marietta based on system function considerations.

FIGURE 8-9 TYPE 7 STRUCTURAL STEEL ANCHOR SYSTEM WITH SLANTED AUGERED CAISSON FOUNDATION FOR HELIOSTAT

FIGURE 8-10 CONCEPTUAL FOUNDATION DESIGN FOR HELIOSTAT AND SENSOR USING TYPE 1

8.4.1 Design Criteria for the Receiver Tower

To develop a structural design for the receiver tower, several wind and seismic conditions were assumed. Forces acting on the tower due to these conditions were calculated. These wind and seismic generated loads were evaluated and used to determine design parameters such as bending moment, shear and uplift forces.

The tower was designed to operate in accordance to specifications at 13.4 m/s (30 mph) steady wind with gusts up to 17.9 m/s (40 mph) and to survive wind velocities up to 53.6 m/s (120 mph). It was also designed to survive a postulated major earthquake of 0.3 g maximum ground acceleration without experiencing functional loss or structural failure. All wind velocities are at 9.1 m (30.0 ft) above ground and all wind loads are assumed to be steady and acting on the structures as equivalent static loads, i.e., neglecting any wind induced dynamical effects. It is further assumed that seismic and wind loads at the survival level do not occur simultaneously.

A minimum soil bearing capacity of 191,500 Pa (4,000 lb/ft²) was adopted. The justification for this assumption was discussed in Section 8.3.1 in the heliostat foundation design. Net allowable soil pressures were increased by one-third for transient loads such as wind and earthquake.

Furthermore, to satisfy lateral stability requirements, a minimum factor of safety against overturning of 1.5 was maintained for the tower conceptual design in all loading conditions.

These environmental design criteria were adopted for the conceptual design stage only and are subject to future verification and revision as actual site data become available.

8.4.2 Design Data for the Receiver Tower

Wind. Wind design data were developed using the ASCE paper "Wind Forces on Structures" published in 1961. Equation 8-1 taken from this paper was used to calculate the variation of wind velocities with height. The maximum wind velocity assumed for the operating condition is 17.9 m/s (40 mph) and that assumed for the survival condition is 40.2 m/s (90 mph) with gusts up to 53.6 m/s (120 mph). On the basis of the 40.2 m/s (90 mph) wind, the exponent 0.233 in Equation 8-1 was determined using the guidelines of the ASCE paper. All specified wind velocities are measured at 9.1 m (30 ft) above ground. Equation 8-1 is valid up to the 183 m (600 ft) level. A constant velocity is used above this level or when the velocity reaches a maximum of 89.4 m/s (200 mph). The wind velocity below 9.1 m (30.0 ft) is taken as uniformly distributed.

$$V'' = V' (Z/X)^{0.233}$$

Equation 8-1

where

V'' = wind velocity at height Z

V' = wind velocity at and below 9.1 m (30.0 ft)

$V' = 17.9$ m/s (40 mph) under operating condition

$V' = 53.6$ m/s (120 mph) under survival condition

$X = 9.1$ m (30.0 ft)

All wind criteria are based on conditions in an open and level terrain as a standard of reference, and the influence of shielding or of deflections and channeling due to unusual topography or to large obstructions are not included.

Proceeding from the wind velocity profile generated using Equation 8-1, the wind load on the tower was determined. Other structural loads include the self weights of the receiver, a service crane and the tower itself. The receiver was assumed to weigh 270,000 kg (300 ton), the crane 9,000 to 18,000 kg (10 to 20 ton). Adopting a concrete tower as the conceptual design, the combined weight of the tower and its auxiliaries was estimated to be 1.4×10^7 kg (15,500 ton).

A maximum allowable lateral sway of 0.53 m (1.75 ft) at the center of the receiver aperture was specified under operating conditions.

Seismicity. The structural integrity of safety-related components of the pilot plant must be assured in the event of earthquakes. This assurance was provided for by designing the plant to withstand the seismic responses resulting from a postulated earthquake.

The Nuclear Regulatory Commission (NRC) Seismic and Geology Siting Criteria requires that, for the purposes of analysis and design, two design earthquakes be specified for each site, viz, the Safe Shutdown Earthquake (SSE), here referred to as the Design Basis Earthquake (DBE), and the Operating Basis Earthquake (OBE). This conservative approach is the standard procedure to nuclear power plant design. Since the pilot plant does not have the potential radiation hazards associated with nuclear facilities, the OBE was not considered in the seismic criteria and only the DBE was adopted. This somewhat relaxed approach facilitates the plant structural design because it eliminates the need to carry out structural analysis at the OBE level. This is possible because the safety requirements for this type of solar plant are not expected to be as stringent as those for nuclear plants where safety related systems must continue to operate at the OBE level.

Lacking specific site data, the following guidelines and data were assumed:

- A site located in Zone 3 according to Uniform Building Code (UBC) excluding sites near active faults.

- A maximum ground acceleration of 0.3 g for survival condition for both horizontal and vertical components.
- Damping values of 2% and 7% critical for steel and concrete structures respectively.
- Bechtel synthetic time histories.

The DBE was specified by a suitable set of idealized smooth curves, known as the "design spectra", and a compatible time history of motion. The design spectra adopted for the analysis were taken from NRC Regulatory Guide 1.60. They were obtained on the basis of statistical evaluation of the actual response spectra of many strong-motion earthquakes recorded at sites underlying various geologic materials in the western United States. Hence, in using these curves, the variations in site conditions and foundation properties and the effects of distant and nearby earthquakes were taken into account.

Figure 8-11 and 8-12 show the horizontal and vertical design spectra respectively for damping values of 2 and 7 percent of critical, for a maximum ground acceleration of 0.3 g. The damping values adopted in the analysis are specified in NRC Regulatory Guide 1.61.

The response spectra of the time history are such that they envelop the design spectra at a sufficient number of frequencies of which only the specific structural frequencies are of engineering significance in the seismic analysis. Since no recorded earthquake motions would fulfill this criterion, synthetic time history motions were generated by modifying actual records or other digitally simulated motions according to a technique proposed by N.C. Tsai ("Spectrum-Compatible Motions for Design Purposes", J. of Eng. Mech. Div., ASCE, Vol. 98, No. EM2, Proceedings Paper 8807, April 1972, pp. 345-356).

Figure 8-13 shows one of such synthetic time history motion, which was generated by modifying the 1952 Taft earthquake and has a total duration of 24 seconds. The maximum integrated velocity of the time history is about 1.5 m/s (5.0 ft/s) for maximum ground acceleration of 1.0 g. Its response spectra generally envelop the design spectra for damping values equal to or greater than one percent of critical. The response spectra occasionally fall below the corresponding design spectra, but the amount is always less than 10 percent.

Figure 8-14 shows the vertical component of the synthetic time history motion for a 24-second duration generated according to the same technique. The maximum integrated velocity of this time history is about 1.2 m/s (3.8 ft/s) for a maximum vertical ground acceleration of 1.0 g.

For engineering design purposes, the design earthquake is assumed to be the free-field motion at the structural basement level, without accounting for soil-structure interaction effects.

FIGURE 8-11 HORIZONTAL DESIGN RESPONSE SPECTRA

FIGURE 8-12 VERTICAL DESIGN RESPONSE SPECTRA

SYNTHETIC TIME HISTORY OF THE HORIZONTAL
COMPONENT OF THE DESIGN EARTHQUAKE

FIGURE 8-13 SYNTHETIC TIME HISTORY (HORIZONTAL)

SYNTHETIC TIME HISTORY OF THE VERTICAL
COMPONENT OF THE DESIGN EARTHQUAKE

FIGURE 8-14 SYNTHETIC TIME HISTORY (VERTICAL)

8.4.3 Tower Design Alternatives

Three tower design alternatives were identified as possible candidates for supporting the receiver. These are a guyed steel tower, a free-standing steel tower, and a reinforced concrete tower. These three tower types were compared in terms of deflection during operating conditions and structural integrity under survival conditions. Relative costs were developed for the free-standing steel tower and the reinforced concrete tower.

Guyed Steel Tower. A guyed steel tower is a high trussed structural steel shaft supported at suitable intervals with high strength wire strands. This is illustrated in Figure 8-15. The structure is usually very slender and light and carries a small weight, usually an antenna mast, at its top. Although this type of construction is highly economical, it is limited to the support of a small top load and requires adequate ground space access for anchoring the guys, and is applicable only when the lateral sway is of no concern in design.

In the present case, the heavy weight of the receiver (2.7×10^5 kg or 300 tons) and the weight of the supporting platform (about 45,000 kg or 50 tons) require many columns to carry the forces to the foundation. Consequently, the design of the main structure approaches that of a free-standing steel tower. Furthermore, since the guys require considerable ground space for anchorage, they interfere with the heliostat field. Previous design experience suggests that the lateral sway (an important criterion in the design) of this type of tower cannot be restricted easily. Another matter of concern is the potential hazard of accidental damage to a guy wire which could cause collapse of the structure. For all these reasons, this alternative was not considered further as a feasible candidate.

Free-Standing Steel Tower. A free-standing steel tower is a self-supported cantilever structure. The tower designed for this study is shown in Figure 8-16. It has a uniform 12.2 m (40.0 ft) square cross-section throughout its height. Twenty columns arranged along the periphery of the cross-section carry the loads to the foundation. A service elevator (not shown) is located inside the tower and is supported by four separate columns. Four levels of column bracing are provided to carry the lateral loads induced by wind or earthquake. The column size is uniform between consecutive bracing levels.

The pile foundation was designed as a matrix of reinforced concrete piles arranged in a pattern of 15 by 15 rows (total of 225). A length of 12.2 m (40.0 ft) was assumed for all piles on the basis of their carrying capacity due to skin friction only. The piles are held together by a 24.4 m (80 ft) square by 2.1 m (7.0 ft) deep reinforced concrete pile cap. This foundation type is site sensitive, and therefore it was analyzed and its cost was estimated separately from the tower to facilitate possible design and cost adjustments when site conditions become better defined. The total lateral sway, under the "operating condition" wind

velocity of 17.9 m/s (40 mph), was found to be 0.08 m (0.26 ft) which is less than the maximum allowable deflection of 0.53 m (1.75 ft). Under the survival wind velocity of 53.6 m/s (120 mph) the total lateral sway was estimated to be 0.71 m (2.32 ft).

Reinforced Concrete Tower. The reinforced concrete tower considered in this study is a hollow, vertical cantilever as shown in Figure 8-17. For simplicity in stress analysis and for the purpose of first order cost comparison, the tower was conceptualized as a truncated cone of revolution. Actual construction methods may indicate that an exponential shape is more desirable. A more detailed analysis will be performed in the preliminary design stage to refine this conceptual design.

The tower tapers from an inside base diameter of 19.8 m (65.0 ft) to a top inside diameter of 15.2 m (50.0 ft). The wall thickness also decreases from 0.9 m (3.0 ft) at the base to 0.6 m (2.0 ft) at the top. The top diameter was established from the requirement to support the receiver and the associated platform structure. The wall thickness at the top was established from consideration of the allowable bearing stress on concrete. The bottom diameter and wall thickness were established from consideration of allowable tensile stress in the reinforcing steel. A service elevator (not shown) is located inside the tower and is supported by four separate structural steel columns.

The foundation is a 45.7 m (150.0 ft) diameter circular mat of variable thickness. The thickness of the mat increases from 0.91 m (3.0 ft) at its edge to 1.83 m (6.0 ft) at a point 12.2 m (40.0 ft) from the tower center line. The central region of radius 12.2 m (40.0 ft) is 3.35 m (11.0 ft) thick, as shown in Figure 8-17.

This reinforced concrete tower is heavier and considerably more rigid than the free-standing steel tower described before. The estimated lateral sway is 0.010 m (0.033 ft) under operating and 0.09 m (0.30 ft) under non-operating (survival) conditions. The factor of safety against overturning is 1.70. A preliminary cost estimation revealed that this tower, constructed by jumpforming, is approximately 20% less expensive than the free-standing steel tower shown in Figure 8-16.

8.4.4 Selection of Tower Design Alternative and Conceptual Design of Tower

The three design alternatives described in Section 8.4.3 were reviewed in order to select a preferred approach to the conceptual design of the receiver tower. For reasons which were discussed in Section 8.4.3, the guyed steel tower design was discarded. A comparison of the free-standing steel tower and the concrete tower led to the choice of the latter as the preferred approach. The rationale and parameters considered to arrive at this selection are as follows:

FIGURE 8-17 REINFORCED CONCRETE TOWER

FIGURE 8-18 COMPUTER PROGRAM GEOMETRIC CAPABILITIES

- The free-standing steel tower is approximately 20 percent more expensive than the concrete tower.
- The steel tower has a higher lateral sway. For normal operating conditions, its lateral deflection is 0.08 m (0.26 ft) whereas the corresponding value for the concrete tower is only 0.01 m (0.033 ft).
- The maintenance for the steel tower appears to be higher than the concrete tower especially when painting is required periodically.
- The wind-induced vibration transmitted to the receiver at the interface cannot be estimated unless a detailed dynamic analysis is performed. However, the high rigidity of the concrete tower minimizes this effect better than the steel tower.
- Another effect that was not evaluated at this conceptual design stage is the consideration of tower movements due to heat differentials. Again, it is expected that the concrete tower will be less influenced by this effect.
- Finally, the concrete tower is an enclosed structure as opposed to the open steel tower. Hence, it will provide better protection from environmental hazards--such as wind-generated missiles, wind, hail and lightning, as well as misfocused solar radiation--to critical systems like piping, elevator well, stairs, working platforms, electrical wiring and personnel.

Proceeding from the selected design approach, the conceptual design of the receiver tower was conducted. This effort involved the following tasks:

- A computer program was developed to calculate effects on the tower due to wind loads.
- A simple dynamic model was developed for the tower, and a time-history dynamic analysis was performed using as forcing functions the synthetic time-history earthquakes described in Section 8.4.2. These seismic inputs were applied at the tower base, assuming a rigid foundation, to obtain the resulting time-history responses and response spectra at the top of the tower where the receiver is mounted.
- Conceptual designs were developed for (1) the tower top to show the receiver supporting frame, access requirements and auxiliaries, and (2) the piping support for the riser and the downcomer.

Wind Generated Loads. A computer program was developed to conduct a simplified analysis of wind-generated loads on the tower. Basically, the tower was considered to be a cantilever

beam fixed at the base. The differential equations used included beam-column effects and variable cross sections and were numerically integrated. The program can handle gravity and variable wind loads, as well as concentrated bending moment, axial and shear forces applied at the tower top. The program outputs consist of displacements, total bending moments, as well as shear and axial forces at different elevations. In order to verify this program, two known closed form solutions were implemented to check the program formulation and the accuracy of the numerical integration routines. These two solutions are for the first two shapes shown schematically in Figure 8-18; i.e., a hollow circular cylinder and a solid cone. The third shape shown is a truncated cone and was the one used for the conceptual design of the concrete tower. During the preliminary design stage, a more sophisticated exponential shape shown in this figure may be required.

Using this computer program, the concrete tower design was analyzed to determine design parameters such as lateral deflection, bending moment, axial force and shear force for a survival wind condition of 53.6 m/s (120 mph). The results are presented diagrammatically in Figure 8-19. The maximum horizontal sway computed for the top of the tower is 0.088 m (0.29 ft), which compares well with the 0.091 m (0.30 ft) estimated previously (see Section 8.4.3) during the initial tower design selection study.

Similar calculations for the operating condition of 17.9 m/s (40 mph) wind indicate that the top sway is 0.013 m (0.042 ft), somewhat higher than 0.010 m (0.033 ft) estimated previously (see Section 8.4.3). This is still considerably lower than the allowable limit of 0.53 m (1.75 ft), and confirms the high rigidity of the reinforced concrete tower.

Seismic Response. A mathematical model was developed to predict the seismic response of the tower when subjected to the design earthquakes described in Section 8.4.2. This model is illustrated in Figure 8-20. Since the tower is designed to remain essentially elastic under seismic loadings, it is assumed that the structural deformation is small and linearly elastic. The model is a lumped parameter representation of the structure. A total of mass points were chosen to enable the definition of five vibrational modes. The tower was divided into ten parts of equal height and their weights were lumped into ten points. The receiver was treated as a point weight of 2.67×10^7 N (600 kip) connected to the tower top by a rigid link.

Using this dynamic model, the tower structural response to the specified ground accelerations of the design earthquakes was computed. To provide pertinent data for the structural design of the receiver, a transient (time-history) response at the top of the tower was generated. The horizontal and vertical response spectra at the point where the receiver is mounted to the top of the tower are illustrated in Figures 8-21 and 8-22. Both two and seven percent of critical dampings were given.

FIGURE 8-19 WIND GENERATED LOADS: SURVIVAL CONDITION

FIGURE 8-20 TOWER DYNAMIC MODEL

FIGURE 8-21 HORIZONTAL RESPONSE SPECTRA AT RECEIVER/TOWER INTERFACE

FIGURE 8-22 VERTICAL RESPONSE SPECTRA AT RECEIVER/TOWER INTERFACE 8-27

FIGURE 8-23 EARTHQUAKE GENERATED LOADS: SURVIVAL CONDITION

FIGURE 8-24 TOWER TOP LAYOUT

In order to evaluate the design parameters such vertical deflection, axial force, bending moment and shear force due to seismic loadings, a dynamic analysis was conducted using the model shown in Figure 8-20. The results are shown in Figure 8-23. These results were compared with those obtained in the wind load analysis. This comparison provided the maximum set of forces required for the subsequent stress analysis which in turn furnished a check on the concrete stresses and the required quantification of steel at different tower elevations.

A comparison of Figures 8-23 and 8-19 shows that the seismic loading generates higher values at almost every section and therefore controls the design.

Tower Top Layout. This is illustrated in Figure 8-24. It shows how the receiver interfaces with the top of the tower. To avoid interference of the solar radiant flux directed from the adjacent heliostats with the edge of the tower, the receiver is eccentrically located with respect to the tower center line.

A service elevator well is located behind the receiver. The elevator reaches the top of the receiver; it also provides access to the intermediate working platforms.

A rectangular access hatch is provided to allow bulky equipment to pass through the supporting platform. Hoisting equipment is provided above the access hatch for lifting receiver components either to the support platform for final assemblage or directly to their final installation positions. Portable hoisting equipment will be used as required to supplement the drum hoist. Figure 8-24 also shows the stairway and the positions of the riser and downcomer pipes. The stairway provides alternate access to the receiver from ground elevation.

Receiver Tower Layout. Figure 8-25 shows the receiver tower layout. The inside diameter at the base is 19.8 m (65.0 ft) tapering to become 15.2 m (50.0 ft) at the top. The wall thickness tapers correspondingly from 0.91 m (3.0 ft) at the base to 0.61 m (2.0 ft) at the top. A typical tower reinforcement detail at the base is also given in this figure showing the positions and sizes of meridional and hoop reinforcement bars for both inside and outside tower walls. This reinforcement is similar at higher elevations, but the amount of steel decreases in relation to the reduction in forces and bending moment. Several access platforms are located at approximately every 16.8 m (55.0 ft) for maintenance purposes. Both an elevator and a stairway are provided for alternate access to these platforms and the receiver.

Slipforming and jumpforming are possible methods for the construction of the tower. Typical advance rates are from 3.7 m (12.0 ft) to 7.3 m (24.0 ft) per day for slipforming and 2.1 m (7.0 ft) a day for jumpforming. However, due to the high cost of slipforming, jumpforming is generally recommended for a tapered tower of this size.

TYPICAL TOWER REINFORCEMENT AT BASE
SCALE: NONE

FIGURE 8-25

[illegible]

8.4.5 Riser and Downcomer Piping Conceptual Design

The feedwater will be carried up the receiver tower in a riser pipe and the superheated steam will be carried from the receiver to the Electrical Power Generation Subsystem and the Thermal Storage Subsystem in a downcomer piping system. In designing this piping system the following items must be addressed:

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

A single pipe arrangement has a lower capital cost, a lower installation cost, and it requires less pump power. The use of single piping runs will simplify the design of flexible configurations and provide structural optimization for support and pipe restraints. In addition, the associated valving and controls to common distribution headers will also be simpler and less costly.

The conceptual design of the riser and downcomer pipes and required insulation was primarily based on specified water and steam flow rate, pressure, temperature and velocity set points at the receiver, the Electrical Power Generation Subsystem and the Thermal Storage Subsystem piping interface points at the bottom of the tower. These set points are as follows:

Design Data for Riser (Feedwater Pipe)

At Feedwater Pump:

Flow Rate = 69,400 kg/h (153,000 lb/h)
Pressure = 13,380 kPa (1,925 psig)
Temperature = 539 K (510°F)

At Receiver:

Flow Rate = 69,400 kg/h (153,000 lb/h)
Pressure = 12,000 kPa (1,725 psig)
Temperature = 536 K (505°F)

Design Data for Downcomer (Steam Pipe)

At Receiver:

Flow Rate = 69,400 kg/h (153,000 lb/h)
Pressure = 9,240 kPa (1,325 psig)
Temperature = 789 K (960°F)

At Turbine:

Flow Rate = 69,400 kg/h (153,000 lb/h)
Pressure = 8,890 kPa (1,275 psig)
Temperature = 783 K (950°F)

The sizing of pipe normally requires analyzing 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, which requires knowledge of both plant cost and plant annual output. With no conceptual design data available in these areas, the alternative method used in this study was to assume reasonable pressure and temperature losses in the pipes. The assumed values are:

Pressure Drop in Riser	= 1,379 kPa (200 psi)
Temperature Loss in Riser	= 2.8 K (5°F)
Pressure Drop in Downcomer	= 345 kPa (50 psi)
Temperature Loss in Downcomer	= 5.6 K (10°F)

The 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 of the pipe. The total friction head loss and effective length of the pipe is a result of the accumulated friction losses due to the number and type of valves, the number of pipe elbows and expansion loop bends, in addition to friction loss in the actual pipe length.

On the basis of these considerations, the pipe size required was calculated. The riser was determined to be a carbon steel nominal four inch diameter schedule 80 pipe with a one inch insulation, and the downcomer a one and one quarter percent chrome-one half percent moly nominal eight inch diameter schedule 100 pipe with a six inch 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 the remaining 4.0 K (7°F). Calcium silicate was chosen as the insulation material since it is generally used in power plant piping and appears to be more economical than alternative materials such as mineral wool and fiberglass.

With the pipes operating at elevated temperatures, they will experience considerable thermal expansion. Two methods of absorbing this expansion have been considered:

- Expansion loops.
- Bellows-type expansion joints.

Although expansion joint design has a lower capital cost, bellows-type expansion joints are not commercially available with pressure-temperature rating above 2,800 kPa/250 K (400 psi/400°F). The conceptual design therefore uses an expansion loop design.

Some considerations were also given to the option of a straight run piping without thermal expansion loops in an attempt to reduce the pipe length and to minimize energy loss.

The total vertical piping growth due to thermal expansion in the downcomer main steam line is 1.0 m (3.25 ft) and half that amount is in the feedwater line. The total horizontal piping growth due

FIGURE 8-15 GUYED STEEL TOWER

FIGURE 8-16 FREE-STANDING STEEL TOWER

to thermal expansion in the main steam line is 0.33 m (1.08 ft) and 0.20 m (0.67 ft) in the feedwater line.

With the thermal growth and movement indicated above, the use of rigid anchors for pipe hangers and seismic restraints would be extremely limited. Only fixed piping end points could be so supported. Variable spring hangers would have similar limitations.

The use of constant support flexible hangers is limited to 0.51 m (1.67 ft) of total travel (including thermal) with a mechanism weight of 4,100 kg (9,000 lb). The use of seismic restraint snubbers is limited to a total travel of 0.30 m (1.0 ft).

The accommodation of total thermal expansion without expansion loops would require development and testing activities and costs. Constant support and variable spring hangers and seismic restraint snubbers would be required with a 1.0 m (3.25 ft) thermal travel.

In view of the considerations delineated above, this design option was abandoned.

Since the pressure drop depends on the number of bends and total pipe length, the pipe size depends on the expansion loop design. The expansion loop design, however, depends on the pipe size. Therefore, the design of pipe size and expansion loop parameters is an iterative process. In this conceptual design piping layout, initial thermal expansion calculations, as well as plant design and piping analysis experience on hangers and supports, have been involved in producing this piping arrangement. This is presented in Figure 8-25 showing the number of thermal expansion loops for the riser and downcomer.

A major design concern is thermal cycling. The pipes must be designed to be sufficiently flexible to accommodate the thermal cycles and to ensure that reaction effects are adequately designed for. In the design of supports and restraints, the effects of gravity loads, thermal movement, and seismic disturbance of these two pipes were considered. The effects of pipe reactions on the receiver, EPGRS components and the tower were also considered. Applicable codes and standards such as the ASME Boiler and Pressure Vessel Code and ANSI B31.1 Power Piping Code were used to determine material selection, design allowable stress and other applicable design criteria.

Detailed calculations for flexibility, considering thermal cycling, supported weight and seismic effects will be conducted in the subsequent preliminary design tasks. Initial review and consideration of these design criteria has been conducted in the conceptual design efforts. The conceptual design piping layout is subject to further verification and revision when preliminary design stress analysis results become available.

An important consideration in the development of this novel pilot plant is the early identification of potential environmental impacts. The conceptual design of the pilot plant was reviewed to determine the type of data that would be needed to assess the impact of the pilot plant on the environment.

A preliminary study is being conducted to identify major environmental impacts. These include:

- Changes in local microclimatology.
- Disruption of existing site terrestrial ecology.
- Degradation of water and soil quality due to accidental spillage of thermal storage media.
- Alteration in existing land use.

Using these environmental impacts, as well as other project engineering information, environmental siting criteria are being prepared. These criteria will delineate beneficial and adverse site environmental characteristics which need to be considered by the ERDA site selection team.

This preliminary study of environmental impacts and siting criteria will be concluded in January 1976 and will be reported in more detail in the next quarterly technical progress report.