305
11/13/80 T.S.

LD. 2011

GA-A-15580

LINE FOCUS SOLAR CENTRAL POWER SYSTEMS. PHASE I

Final Report, September 30, 1978—October 31, 1979

MASTER

September 1979
Date Published

Work Performed Under Contract No. ET-78-C-03-2240

General Atomic Company
San Diego, California

U.S. Department of Energy

Solar Energy

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

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

DISCLAIMER

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

DISCLAIMER

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

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

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

Price: Paper Copy \$32.00
Microfiche \$3.50

LINE FOCUS SOLAR CENTRAL POWER SYSTEMS PHASE I

**FINAL REPORT FOR THE PERIOD
SEPTEMBER 30, 1978 THROUGH OCTOBER 31, 1979**

PROJECT MANAGER: J. R. SCHUSTER

**Prepared under
Contract ET-78-C-03-2240
for the San Francisco Operations Office
Department of Energy**

**GENERAL ATOMIC PROJECT 3295
DATE PUBLISHED: SEPTEMBER 1979**

**General Atomic Company
P.O. Box 81608
San Diego, California 92138**

GENERAL ATOMIC COMPANY

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

ABSTRACT

A conceptual design study was performed of a stand-alone Line Focus Solar Central Power System based on the fixed mirror solar concentrator (FMSC) for heat collection and draw salt (a 50% molar mixture of sodium nitrate and potassium nitrate) for heat transport and storage.

Parametric analyses were performed at the subsystem level, and models were developed that were employed in a computerized simulation to minimize the cost of electricity (COE) by adjusting system design parameters. A design was prepared and costed for a first commercial plant with a rating of 100 MW(e) and a storage capacity equivalent to 420 MW(e)-hr of generation. The resulting plant achieves an annual capacity of 45.6%. Scaling studies indicate reductions in the COE for increased capacity factor and increased plant rating. Assessments of the plant concept indicate it should be acceptable to utilities on the basis of technical and operational considerations, but that reductions from the first 100-MW(e) plant cost would be required to achieve substantial market penetration.

**THIS PAGE
WAS INTENTIONALLY
LEFT BLANK**

ACKNOWLEDGMENTS

General Atomic Company would like to acknowledge the guidance and support provided by Mr. Larry E. Prince and Mr. John V. Coggi during the course of this program. Mr. Prince was the LFSCPS Program Manager at the DOE San Francisco Operations Office. Mr. Coggi represented the Aerospace Corporation, which provided technical program management for DOE.

General Atomic would also like to acknowledge the contributions made by its employees:

Mr. John R. Schuster, Program Manager
Dr. John M. Neill, Lead Engineer
Mr. John C. Bass, Mechanical Design
Mr. Mike Croft, Receiver and Control Analyses
Mr. George H. Eggers, Concrete/Glass Mirror Weathering Tests
Mr. James Kay, Jr., Plant Cost Analysis
Ms. Frances J. Martin, Computer Programming
Mr. Vladimir Nicolayeff, Concentrator Structural Design
Mr. Frank L. Openshaw, Concentrator, Receiver, and Control Analyses
Mr. Donald Schwarzer, Plant Cost Analysis
Mr. William S. Scott, Plant Cost Analysis
Mr. William E. Walker, Operation and Maintenance Studies, Glass
Breakage Evaluations

Finally, General Atomic would like to acknowledge the contributions of its subcontractors:

Dr. Ernest Y. Lam, Project Manager, Bechtel National, Inc.
Mr. Robert L. Lessley, Lead Engineer, Bechtel National, Inc.

Mr. Kenneth Ladd, Jr., Project Manager, Southwestern Public Service
Company

Mr. Cleon Ligon, Lead Engineer, Southwestern Public Service Company

Mr. Alan Higgins, Engineer, Southwestern Public Service Company

Mr. Tom Springer, Project Manager, Energy Systems Group of Rockwell
International

Mr. Anargiros Z. Frangos, Lead Engineer, Energy Systems Group of
Rockwell International

Mr. Hadley K. Wiard, H. K. Design Services

CONTENTS

ABSTRACT	iii
ACKNOWLEDGMENTS	v
1. INTRODUCTION AND SUMMARY	1.1-1
1.1. Description of Project	1.1-1
1.2. Plant Design Summary	1.2-1
1.3. Background and Prior Art	1.3-1
REFERENCES	1.3-7
1.4. Summary of Investigations Conducted.	1.4-1
REFERENCES	1.4-12
2. PARAMETRIC ANALYSIS	2.1-1
2.1. Heat Transport Fluid	2.1-1
2.1.1. Introduction	2.1-1
2.1.2. Fluid Properties	2.1-4
2.1.3. Point Designs	2.1-7
2.1.4. Costs	2.1-10
2.1.5. Fluid Choice	2.1-12
2.1.6. Salt Corrosion Data	2.1-17
REFERENCE	2.1-22
2.2. Concentrator and Receiver Subsystems	2.2-1
2.2.1. Concentrator/Receiver Loss Train	2.2-1
2.2.2. Collector Performance Model (SUNPOW).	2.2-3
2.2.3. Receiver Performance Model (SUNPOW)	2.2-11
2.2.4. Concentrator Subsystem Cost Model	2.2-17
2.2.5. Receiver Cost Model	2.2-19
REFERENCES	2.2-22
2.3. Heat Transport Subsystem	2.3-1
2.3.1. HTS Design Criteria	2.3-1
2.3.2. Approach	2.3-2
2.3.3. Materials	2.3-3

2.3.4.	Piping Arrangements	2.3-4
2.3.5.	Insulation	2.3-14
2.3.6.	Pipe Size	2.3-14
2.3.7.	Trace Heating	2.3-15
2.3.8.	Drain Provisions	2.3-15
2.3.9.	Pumps	2.3-16
2.3.10.	Steam Generator	2.3-18
2.3.11.	Reference HTS	2.3-22
2.3.12.	HTS Cost Model	2.3-24
2.3.13.	Selected HTS Arrangements	2.3-27
REFERENCES		2.3-31
2.4.	Energy Storage Subsystem	2.4-1
2.4.1.	Introduction and Design Criteria	2.4-1
2.4.2.	Salt/Taconite Thermoclines	2.4-2
2.4.3.	All-Salt Thermocline	2.4-8
2.4.4.	Multiple Tanks	2.4-8
2.4.5.	Reference Design Choice	2.4-18
2.5.	Electric Power Generating Subsystem	2.5-1
2.5.1.	EPGS Design Criteria	2.5-1
2.5.2.	Candidate Power Cycles	2.5-2
2.5.3.	Reference EPGS	2.5-3
2.5.4.	Alternate Operating Conditions	2.5-5
2.5.5.	Power Plant Auxiliary Load Model	2.5-6
2.5.6.	EPGS Cost Model	2.5-8
2.6.	Master Control Subsystem (MCS).	2.6-1
REFERENCE		2.6-6
2.7.	Selection of Power Plant Operating Conditions	2.7-1
2.7.1.	ESS Cost Model	2.7-1
2.7.2.	CS/RS Cost Model	2.7-1
2.7.3.	Selected Power Plant Operating Conditions	2.7-2
3.	SYSTEM DESIGN AND ANALYSIS	3.1-1
3.1.	Introduction	3.1-1
3.2.	SS2 Cost and Performance Computer Code	3.2-1
3.2.1.	Code Logic	3.2-1
3.2.2.	SS2 Code Subsystem Models	3.2-4

3.3.	Parameter Selection	3.3-1
3.3.1.	System Analysis Results	3.3-1
3.3.2.	Optical/Thermal Data	3.3-15
REFERENCES		3.3-23
3.4.	Utility Data	3.4-1
3.4.1.	Introduction	3.4-1
3.4.2.	Load Profiles	3.4-1
3.4.3.	Incremental Cost Data	3.4-4
3.5.	Operational and Environmental Considerations	3.5-1
3.5.1.	Collector Row Length Limitations	3.5-1
3.5.2.	Initial Plant Startup	3.5-5
3.5.3.	Collector Field Maintenance	3.5-7
3.5.4.	FMSC Resistance to Hail Damage	3.5-11
3.5.5.	Effect of Wind Loads	3.5-13
REFERENCES		3.5-14
3.6.	System Dynamic Behavior	3.6-1
3.6.1.	Control Philosophy	3.6-1
3.6.2.	Design Transient Events	3.6-3
3.6.3.	Dynamic Analysis Model	3.6-3
3.6.4.	Startup and Shutdown Sequences	3.6-7
3.6.5.	Mode Changing Sequences	3.6-11
4.	CONCEPTUAL DESIGN	4.1-1
4.1.	Introduction	4.1-1
4.1.1.	System Description and Requirements	4.1-1
4.1.2.	System Performance	4.1-5
4.2.	Concentrator Subsystem	4.2-1
4.2.1.	Concentrator Subsystem Requirements	4.2-1
4.2.2.	Concentrator Subsystem Design	4.2-2
4.2.3.	Concentrator Design Studies	4.2-8
4.2.4.	Collector Deflection Analysis	4.2-19
4.2.5.	Concentrator Performance	4.2-26
REFERENCES		4.2-28
4.3.	Receiver Subsystem Design	4.3-1
4.3.1.	Receiver Design Requirements	4.3-1
4.3.2.	Receiver Reference Design	4.3-2

4.3.3.	Receiver Subsystem Studies	4.3-39
4.3.4.	Seismic Analysis	4.3-54
4.3.5.	Receiver Instrumentation and Electrical Equipment	4.3-65
4.3.6.	Receiver Performance	4.3-69
4.3.7.	Concentrator/Receiver Performance	4.3-75
REFERENCES		4.3-82
4.4.	Heat Transport Subsystem	4.4-1
4.4.1.	HTS Requirements	4.4-1
4.4.2.	HTS Design	4.4-2
4.4.3.	HTS Performance, Operation and Control	4.4-54
REFERENCES		4.4-71
4.5.	Energy Storage Subsystem	4.5-1
4.5.1.	ESS Requirements	4.5-1
4.5.2.	ESS Design	4.5-1
4.5.3.	ESS Performance, Operation and Control	4.5-30
4.6.	Electrical Power Generation Subsystem	4.6-1
4.6.1.	EPGS Requirements	4.6-1
4.6.2.	EPGS Design	4.6-1
4.6.3.	EPGS Performance, Operation and Control	4.6-34
4.7.	Master Control Subsystem (MCS)	4.7-1
4.7.1.	MCS Requirements	4.7-1
4.7.2.	MCS Design	4.7-2
4.7.3.	Subsystem Controls	4.7-7
5.	PERFORMANCE AND COST ESTIMATES	5.1-1
5.1.	Introduction	5.1-1
REFERENCE		5.1-3
5.2.	Performance Estimates	5.2-1
5.3.	Cost Estimates	5.3-1
5.3.1.	Basis of Cost Estimates	5.3-1
5.3.2.	Plant Capital Costs	5.3-13
5.3.3.	Operating and Maintenance Cost	5.3-29
5.3.4.	Construction Schedule	5.3-34
5.3.5.	Cost of Electricity	5.3-38
REFERENCES		5.3-41

6.	LIMITED SUBSYSTEM AND COMPONENT EXPERIMENTS	6.1-1
6.1.	Introduction	6.1-1
6.2.	Test Plan	6.2-1
6.2.1.	Objective	6.2-1
6.2.2.	Background	6.2-1
6.2.3.	Justification	6.2-2
6.2.4.	Approach	6.2-3
6.2.5.	Test Program	6.2-3
6.2.6.	Materials to be Tested	6.2-7
7.	COMMERCIAL PLANT ASSESSMENT	7.1-1
7.1.	Potential Improvements	7.1-2
7.1.1.	Improved Performance	7.1-2
7.1.2.	Improved Design	7.1-6
7.1.3.	Improved Manufacturing/Assembly	7.1-8
7.2.	Plant Limitations	7.2-1
7.3.	Utility Perspective of Plant	7.3-1
7.3.1.	Performance	7.3-1
7.3.2.	Economics	7.3-5
7.3.3.	Technical	7.3-10
7.3.4.	Operability	7.3-14
7.3.5.	Reliability	7.3-16
7.3.6.	Summary	7.3-16
7.4.	Preliminary Marketing Analysis	7.4-1
7.4.1.	Introduction	7.4-1
7.4.2.	Marketing Analysis Formalism	7.4-2
7.4.3.	Marketing Assessment Results	7.4-9
	REFERENCES	7.4-12
8.	PLANT DEVELOPMENT	8.1-1
8.1.	Development Issues	8.1-1
8.1.1.	Solar Selective Absorptive Coating	8.1-1
8.1.2.	Primary Concentrator Fabrication	8.1-2
8.1.3.	Collector/Receiver Subsystem Performance Verification	8.1-2
8.1.4.	Corrosion Data	8.1-3
8.2.	Open Questions	8.2-1

APPENDIX I.	LINE FOCUS SOLAR CENTRAL POWER SYSTEMS REQUIREMENTS DEFINITION AND REFERENCE METHODOLOGY FOR CALCULATING THE BUS BAR COST OF ELECTRICITY	I-1
APPENDIX II.	FABRICATION SPECIFICATION SOLAR PANELS-BEAMS FOR FREEZE-THAW TESTS	II-1

FIGURES

1.1-1	Summary schedule	1.1-3
1.1-2	Project team functional organization	1.1-4
1.2-1	LFSCPS schematic	1.2-2
1.3-1	FMSC geometry	1.3-2
1.3-2	Principal angular relationships of the FMSC	1.3-3
1.3-3	FMSC installation at Sandia Laboratories, Albuquerque	1.3-6
1.3-4	FMSC array installed in molten salt test loop	1.3-8
1.3-5	Molten salt storage and supply system	1.3-9
2.1-1	Representative thermal degradation rates of various heat transport oils	2.1-14
2.2-1	Optical/thermal performance train	2.2-2
2.2-2	Ray travel directions and distances	2.2-5
2.2-3	Structural block and structural shadow direction	2.2-6
2.2-4	Image projection from south-most mirror strip of module at receiver	2.2-8
2.2-5	Image projection from mirror strip in mid-northern half of module at receiver	2.2-9
2.2-6(a)	Model initially used in SUNPOW and SS2	2.2-12
2.2-6(b)	Advanced receiver model now used in SOLCOL and SS2	2.2-12
2.2-7	Receiver design evaluated in advanced receiver design model confirmation study	2.2-14
2.2-8	Concentrator panel cost trends	2.2-18
2.3-1	Butterfly-pattern receiver-manifold arrangement	2.3-5
2.3-2	Approximate Size of 100 MWe HTS	2.3-7
2.3-3	Effect of receiver run length	2.3-9
2.3-4	Alternate receiver run configurations	2.3-10
2.3-5	Single vs. multiple trunk supply lines	2.3-12
2.3-6	Reference HTS configuration	2.3-13
2.3-7	HTS pumping system	2.3-17

FIGURES (continued)

2.3-8	Reference steam generator temperature profiles	2.3-20
2.3-9	Steam generator conceptual design	2.3-21
2.3-10	Final HTS configuration	2.3-28
2.4-1	Stabilized performance of a salt/Taconite thermocline	2.4-5
2.4-2	Relation between bed utilization and end temperature drop for a salt/Taconite thermocline	2.4-6
2.4-3	Salt/Taconite thermocline bed discharge performance	2.4-7
2.4-4	Salt/Taconite thermocline storage system costs	2.4-10
2.4-5	Thermal storage costs for all-salt thermocline and all-salt single hot/single cold tanks	2.4-12
2.4-6	Hot/warm tank storage system costs	2.4-14
2.4-7	Cost ratio for multiple versus single tanks with same storage volume	2.4-15
2.4-8	Heat loss per unit area for storage tanks and piping	2.4-16
2.4-9	Draw salt storage tank temperature decay during standby	2.4-17
2.4-10	Cost summary for alternate thermal storage systems	2.4-19
2.5-1	Electric power generating subsystem flow diagram	2.5-4
2.6-1	Conventional plant and direct digital control	2.6-2
2.6-2	Distributed plant control	2.6-5
2.6-3	"Typical" computer costs for various microcomputer and minicomputer configurations and projections of future costs	2.6-7
3.2-1	Overall logic of cost performance optimization code SS2	3.2-2
3.2-2	Heat transport system model	3.2-5
3.2.3	Collector arrangement	3.2-7
3.3-1	Sensitivity of bus bar cost of electricity (BBCOE) to selected parameters (high-sensitivity set)	3.3-5
3.3-2	Gross turbine heat rate	3.3-9
3.3-3	Sensitivity of bus bar cost of electricity (BBCOE) to selected parameters (low-sensitivity set)	3.3-11
3.3-4(a)	Bus bar cost of electricity versus number of collector rows	3.3-14

FIGURES (continued)

3.3-4(b)	Percent annual loss of collectable energy versus number of collector rows	3.3-14
3.3-5	Normal transmission through Pyrex 7740	3.3-20
3.3-6	Effect of absorber coating on heat collection	3.3-24
3.4-1	Daily load profile for SPS system generation for July 24-30, 1977	3.4-2
3.4-2	Daily load profile for SPS system generation for December 18-24, 1977	3.4-3
3.4-3	Hourly incremental cost of SPS system generation for July 24-30, 1977	3.4-5
3.4-4	Hourly incremental cost of SPS system generation for December 18-24, 1977	3.4-6
3.6-1	Collector control system functional groups	3.6-6
3.6-2	Startup transient flows beginning at 7 a.m., December 21	3.6-9
3.6-3	Startup transient salt temperatures beginning at 7 a.m., December 21	3.6-10
3.6-4	Fluid temperatures following operator-initiated defocus at 10 min	3.6-12
3.6-5	Pipe temperatures as sun drops over horizon starting from 2 p.m., December 21	3.6-13
3.6-6	Control system response to a 20 m/sec cloud, 10 miles (or 13 min) wide, traveling inlet to exit, unseen by insolation detector	3.6-14
3.6-7	Control system response to a 20 m/sec cloud, 10 miles (or 13 min) wide, traveling inlet to exit, seen by insolation detector	3.6-15
4.1-1	Seasonal heat collection characteristics of the FMSC	4.1-8
4.2-1	Concentrator module for LFSCPS	4.2-5
4.2-2	Forms for precast concrete panels	4.2-7
4.2-3	Typical spacing of concentrators	4.2-9
4.2-4	General arrangement of LFSCPS	4.2-10
4.2-5	Typical concrete primary concentrator	4.2-13
4.2-6	Typical foamed plastic primary concentrator	4.2-18
4.2-7	Steel primary concentrator assembly	4.2-20
4.2-8	Side view of mathematical model for primary concentrator	4.2-22

FIGURES (continued)

4.2-9	Developed view of mathematical model for primary concentrator	4.2-23
4.2-10	Vertical displacement plots of primary concentrator due to dead weight	4.2-24
4.2-11	Vertical displacement plots of primary concentrator due to 1-in. vertical settlement at node 90	4.2-25
4.3-1	Receiver truss assembly for LFSCPS	4.3-6
4.3-2	Receiver truss assembly for LFSCPS	4.3-8
4.3-3	Piping interface for LFSCPS	4.3-9
4.3-4	Surface for transferring solar energy to heat receiver tube	4.3-10
4.3-5	Receiver splice and support for LFSCPS	4.3-14
4.3-6	Receiver truss assembly for LFSCPS	4.3-17
4.3-7	Long concentrator support post for LFSCPS	4.3-21
4.3-8	Intermediate concentrator post for LFSCPS	4.3-22
4.3-9	Support column and screwjack assembly for LFSCPS	4.3-24
4.3-10	Screwjack-link assembly for LFSCPS	4.3-25
4.3-11	Radial arm angular position and rate of change as a function of jackscrew travel	4.3-27
4.3-12	Receiver support arm and tilt mechanism for LFSCPS	4.3-29
4.3-13	Receiver support arm and tilt mechanism for LFSCPS	4.3-31
4.3-14	LFSCPS HRA drive train	4.3-35
4.3-15	Driveshaft and motor installation for LFSCPS	4.3-38
4.3-16	Initial reference design heat receiver cross section	4.3-41
4.3-17	Receiver trussing	4.3-42
4.3-18	Receiver cable-drum drive	4.3-44
4.3-19	Receiver circular arch drive	4.3-45
4.3-20	Receiver articulated link drive	4.3-46
4.3-21	Articulated link drive with "A" frame	4.3-47
4.3-22	Offset arm HRA support	4.3-48
4.3-23	Roller track HRA support	4.3-49
4.3-24	Parallel linkage HRA support	4.3-50
4.3-25	Guide rod aiming concept	4.3-51
4.3-26	Wire and drum aiming concept	4.3-52

FIGURES (continued)

4.3-27	Pulley and rod concept	4.3-53
4.3-28	Mathematical model of receiver support mechanism . . .	4.3-55
4.3-29	Seismic stress in post	4.3-58
4.3-30	Seismic stress in receiver support	4.3-59
4.3-31	Seismic stress in jackscrew	4.3-60
4.3-32	Seismic stress in link	4.3-61
4.3-33	Seismic stress in tilt linkage	4.3-62
4.3-34	Seismic stress in over-center link	4.3-63
4.3-35	Dynamic amplification factors for receiver truss . . .	4.3-64
4.3-36	Photo cell pair and relation to light beam	4.3-66
4.3-37	Schematic of alarm circuit	4.3-68
4.3-38	Heat receiver subsystem electrical equipment single line drawing	4.3-71
4.3-39	Heat transfer coefficient vs. air gap at various inclinations	4.3-74
4.3-40	Sandia receiver and truncated concentrator	4.3-76
4.4-1	HTS/ESS flow diagram	4.4-3
4.4-2	HTS piping layout	4.4-5
4.4-3	HTS plot plan	4.4-7
4.4-4	Optimum HTS pipe diameters	4.4-27
4.4-5	Arrangement of steam generator heat exchangers	4.4-42
4.4-6	Steam generator temperature profiles	4.4-49
4.4-7	Heat tracing power distribution single-line diagram .	4.4-51
4.4-8	Steam generator salt exit temperature	4.4-60
4.4-9	Auxiliary heater loop	4.4-66
4.5-1	HT/ESS flow diagram	4.5-3
4.5-2	HTS piping layout	4.5-5
4.5-3	Warm salt storage tank design	4.5-20
4.5-4	Hot salt storage tank design	4.5-21
4.5-5	Alternate foundation designs	4.5-24
4.6-1	Turbine-generator and steam generator building	4.6-3
4.6-2	Heat balance at 25% of guaranteed throttle flow . . .	4.6-5
4.6-3	Maximum guaranteed load heat balance	4.6-7

FIGURES (continued)

4.6-4	VWO + 5% O.P. load heat balance	4.6-9
4.6-5	Main single-line diagram	4.6-11
4.6-6	Single-line diagram for 480-Volt power distribution. .	4.6-13
4.7-1	Simplified schematic of master control system	4.7-3
4.7-2	Simplified schematic of receiver position control. . .	4.7-9
5.2-1	March and December loss trains	5.2-2
5.2-2	June, storage, and year average loss trains	5.2-3
5.2-3	Heat and mass balance, CS-RS driving EPGS	5.2-4
5.2-4	Heat and mass balance, CS-RS driving EPGS and charging ESS (normal solar power operation) at equinox noon	5.2-5
5.2-5	Heat and mass balance, CS-RS driving EPGS with ESS discharging (low solar power operation)	5.2-6
5.2-6	Heat and mass balance, CS-RS charging ESS with ESS discharging (intermittent cloudiness prior to equinox noon)	5.2-8
5.2-7	Heat and mass balance, CS-RS charging ESS only at equinox noon	5.2-9
5.2-8	Heat and mass balance, ESS discharging driving EPGS only	5.2-10
5.2-9	Heat and mass balance, overnight shutdown condition. .	5.2-11
5.3-1	Engineering, procurement and construction schedule for solar line focus power plant	5.3-35
5.3-2	BBCOE as a function of capacity factor and plant rating - first plant values	5.3-39
6.2-1	Environmental test chamber freeze/thaw cycle conditions	6.2-4
7.4-1	Generation planning and dispatch method analysis . . .	7.4-3

TABLES

1.2-1	Key parameters for 100-MW(e) LFSCPS	1.2-3
2.1-1	Heat transport fluid alternatives	2.1-2
2.1-2	Draw salt properties	2.1-5
2.1-3	Point design data	2.1-8
2.1-4	Plant cost changes relative to draw salt system . . .	2.1-11
2.1-5	Comparison of operation and maintenance costs	2.1-13

TABLES (continued)

2.1-6	Fluid selection decision matrix	2.1-16
2.1-7	Corrosion rates in heat transfer salts	2.1-18
2.1-8	Rate constants for oxidation of iron alloys	2.1-19
2.3-1	HTS pump requirements	2.3-18
2.3-2	Reference HTS steam generator design	2.3-23
2.3-3	Reference HTS cost	2.3-23
2.4-1	Thermal storage sizing	2.4-3
2.4-2	Thermocline sizing and costs	2.4-4
2.4-3	Cost components for salt/Taconite thermocline storage	2.4-9
2.4-4	All-salt thermocline sizing	2.4-11
2.4-5	Cost summations for hot/warm tank systems for various hours of storage	2.4-11
2.5-1	Reference EPGS cost	2.5-8
2.7-1	Economic impact of plant operating parameters	2.7-3
3.2-1	Independent variables in the SS2 code	3.2-3
3.3-1	Design parameters selected from optimization data	3.3-3
3.3-2	Summary of parametric data on EPGS performance	3.3-7
3.3-3	Optimum pipe diameters predicted by SS2	3.3-12
3.3-4	Reference optical properties	3.3-16
3.3-5	Comparison of reported properties of solar selective absorbing surfaces	3.3-22
3.4-1	Southwestern Public Service Company generating cost data 1977 monthly averages	3.4-7
3.5-1	Concentrator and receiver subsystem maintenance	3.5-8
3.6-1	LFSCPS transient analytical evaluation	3.6-4
4.1-1	Key parameters for 100-MW(e) LFSCPS	4.1-2
4.1-2	LFSCPS daily performance for June 21	4.1-6
4.2-1	Concentrator subsystem specifications	4.2-3
4.2-2	Concentrator subsystem equipment list	4.2-4
4.2-3	Parametric cost estimates of concentrator panels for one 100-MW(e) plant	4.2-12
4.2-4	Properties of Glasspak reinforced foam	4.2-16
4.2-5	Maximum facet distortion	4.2-26

TABLES (continued)

4.3-1	Heat receiver design tolerance	4.3-3
4.3-2	Heat receiver subsystem specifications	4.3-4
4.3-3	Heat receiver subsystem equipment list	4.3-5
4.3-4	Coefficient of friction for Nitronic 60 couples . . .	4.3-12
4.3-5	Receiver truss maximum deflection and twist	4.3-18
4.3-6	Estimate of total control drive motor power consumed in December or June on the LFSCPS	4.3-36
4.3-7	Heat receiver subsystem electrical equipment list . .	4.3-70
4.3-8	Performance comparison of truncated secondary concentrator and reference design concentrator	4.3-78
4.3-9	SUNPOW performance data for reference design FMSC . .	4.3-79
4.3-10	SUNPOW performance data for low-concentration FMSC . .	4.3-81
4.4-1	HTS equipment list	4.4-9
4.4-2	HTS data list	4.4-15
4.4-3	Optimum insulation thickness	4.4-23
4.4-4	Allowable working pressure for schedule 10S pipe . . .	4.4-24
4.4-5	HTS pipe diameters	4.4-28
4.4-6	HTS heat loss summary	4.4-30
4.4-7	HTS heat loss with loose fill insulation in pipe trenches	4.4-31
4.4-8	Time for insulated warm and hot salt lines to cool to 533K (500F)	4.4-62
4.4-9	Freeze protection energy requirements	4.4-64
4.5-1	ESS equipment list	4.5-7
4.5-2	ESS data list	4.5-12
4.5-3	ESS thermal storage requirement	4.5-17
4.6-1	EPGS equipment list	4.6-15
4.6-2	EPGS data list	4.6-22
4.6-3	Plant auxiliary loads	4.6-37
4.6-4	Plant performance summary	4.6-38

TABLES (continued)

5.3-1	Plant cost estimate summary by account	5.3-14
5.3-2	Plant cost estimate summary by category	5.3-15
5.3-3	Plant costs by account number	5.3-16
5.3-4	Functional distribution of capital costs 1979 dollars	5.3-24
5.3-5	Concentrator subsystem cost details	5.3-27
5.3-6	Receiver subsystem cost details	5.3-28
5.3-7	Maintenance labor costs (steady state - excludes shakedown costs)	5.3-30
5.3-8	Yearly expenditures for maintenance materials and spare parts	5.3-32
6.2-1	First test matrix	6.2-5
6.2-2	Test observations and measurements	6.2-6
6.2-3	Test observation and measurement schedule	6.2-8
7.1-1	Sources for cost reduction and learning	7.1-3
7.1-2	Calculated performance improvement effect on BBCOE	7.1-4
7.3-1	Percent available sunshine in SPS service area	7.3-4
7.3-2	Southwestern Public Service Company 1978 average weekly system fuel cost from base case run	7.3-7
7.3-3	Utility's economic assessment of General Atomic LFSCPS	7.3-8
7.4-1	1979 reference cost information	7.4-5
7.4-2	Estimated fuel-escalation-influenced power increase rates (%)	7.4-6
7.4-3	Yearly equilibrium costs for plants operating at 3504 hr/yr	7.4-11

1. INTRODUCTION AND SUMMARY

1.1 DESCRIPTION OF PROJECT

Under Contract ET-78-C-03-2240 with the San Francisco Operations Office of the U. S. Department of Energy, General Atomic Company has conducted a conceptual design study of a 100-MW(e) Line Focus Solar Central Power System (LFSCPS). The reference LFSCPS uses a fixed mirror solar concentrator (FMSC) field to heat molten salt, which is used both to convey heat energy to the power conversion system and to store heat energy in tanks for later use. The power conversion system uses equipment similar to that employed in fossil-fired power plants to convert the heat to electricity.

The major objectives of the study were to:

1. Prepare the conceptual design of an LFSCPS that has performance and cost advantages over prior solar central power systems.
2. Produce analyses and data showing that this system has an economic potential comparable to or better than first generation point focus central receiver systems.

The work was grouped into 8 tasks:

1. Systems requirements definition.
2. Parametric analysis.
3. Selection of commercial system configuration.
4. Commercial plant conceptual design.
5. Commercial plant cost and performance estimates.

6. Limited subsystem and component experiments.
7. Assessment of commercial-scale system.
8. Phase I program planning and management.

Figure 1.1-1 provides the overall schedule for this work. The duration of the program was from September 30, 1978, to October 31, 1979. A midterm topical report (Ref. 1.1-1) presented results of the first 6 months of study. Figure 1.1-2 presents the project organization, which consisted of General Atomic Company (GA), Bechtel National, Inc., Southwestern Public Service Company, Energy Systems Group (formerly Atomics International) of Rockwell International Corporation, and small/minority businesses.

The Aerospace Corporation was the Technical Program Manager on this project for DOE, and Sandia Laboratories, Livermore, California, was a technical monitor.

General Atomic, as prime contractor and systems integrator, performed system tradeoff and performance evaluations and design, analysis, and costing of the concentrator, receiver, and master control subsystems. Bechtel performed design and costing of the energy storage subsystem; design, analysis, and costing of the heat transport and electrical power generation subsystems; and site design; and assembled the plant cost estimates. Southwestern Public Service Company provided information on utility load profiles, the present and future worth of conventional power generation, and utility requirements that could affect the design of the LFSCPS, and performed an assessment of the limitations of the selected LFSCPS concept. The Energy Systems Group analyzed the energy storage subsystem and assessed fluid handling and development requirements. The small/minority businesses performed some of the required computer code modifications and engineering layout work.

Fig. 1.1-1. Summary schedule

Fig. 1.1-2. Project team functional organization

1.2 PLANT DESIGN SUMMARY

The conceptual design of the LFSCPS is illustrated schematically in Figure 1.2-1. The system comprises a solar collector field which heats draw salt from 565 K (557°F) to 843 K (1075°F). The molten salt is then transported either to a thermal storage system of 420 MW(e)-hr capacity or to a salt-to-steam generator. Steam is then generated at 16.5 MPa (2400 psi) and 811 K (1000°F) and passed through a single reheat turbine driving an electric generator. Conventional forced draft wet tower cooling is employed. Extensive feedwater heating is utilized so that good station efficiency is achieved. The plant net output is 100.5 MW(e).

Significant parameters for the system are given in Table 1.2-1, and an overall arrangement of the plant is illustrated in Figure 4.4-3. The collector field utilizes GA's fixed mirror solar concentrator (FMSC). The FMSC comprises a cylindrical array of fixed mirror strips mounted so that their focus varies over a circular arc throughout the day and is tracked by a simple lightweight movable heat receiver. The collector field comprises 966,000 m² (10.4 million ft²) of mirror surface arranged into 972 rows, each 274 m (900 ft) long and spaced 7.62 m (25 ft) apart. The FMSC units in each row are modular, each being 5.49 m (18 ft) long by 3.61 m (11 ft 10 in.) wide, abutting in pairs so as to support an 11-m (36-ft) long receiver. They have a geometric concentration factor of 71. The primary concentrator panels are made of prestressed, reinforced concrete, selected for its durability and ruggedness, and are fabricated at a temporary factory adjacent to the plant site.

The receiver comprises a heat receiver tube to which is clamped a removable nickel-clad copper foot that has a selectively absorptive coating. The coating is a Pt/AlO₂ cermet which is under development by Cornell University. The coating has been exposed for 300 hr at temperatures of 873 K (1111°F) in air without loss of performance. Its properties of α (absorptivity) of 0.94 and ϵ (emissivity) of less than 0.30 have been

Fig. 1,2-1. LFSCPS schematic

TABLE 1.2-1
KEY PARAMETERS FOR 100-MW(e) LFSCPS

Net design output	100.5 MW(e)
Mirror surface area	$9.66 \times 10^5 \text{ m}^2$ ($10.4 \times 10^6 \text{ ft}^2$)
Heat transport/storage fluid	draw salt
Receiver fluid inlet/outlet temperatures	565 K/843 K (557°F/1057°F)
Collector type	FMSC modules
Module size	5.49 m long x 3.61 m wide (18 ft/11 ft 10 in.)
Geometric concentration factor	71
Receiver surface properties	0.94 absorptivity 0.30 emissivity
Receiver length	11 m (36 ft)
Module arrangement	972 rows, 274 m (900 ft) long at 7.62-m (25-ft) pitch
Land required	232 hectares (573 acres)
Energy storage capacity	420 MW(e)-hr
Turbine type	tandem-compound 3600-rpm reheat unit
Turbine throttle steam conditions	16.5 MPa/811 K (2400 psig/1000°F)
Number of feedwater stages	7
Net EPGS efficiency	39.4%
Net collector field efficiency (including piping losses)	62.0% at equinox noon 32.7% year average
Net plant efficiency	23.8% at equinox noon 12.9% year average
First plant	
Capital cost estimate	\$287.0 million
Annual capacity factor	45.6%
Bus bar cost of electricity	151 mills/kW-hr

measured at temperature. These properties are more conservative than reported for other coatings by other researchers but appear to be the best open-literature tested values at this time.

The receiver is insulated on the back with calcium silicate and has a borosilicate glass window. It has a shallow secondary concentrator which is integrated into the thin steel sheet casing of the receiver shell. The receiver obtains its structural support to meet central sag specifications of less than 6.35 mm (0.25 in) by means of tubular trussing. A lightweight, readily fabricable receiver unit results.

The collector field is sized so that on the day of maximum heat collection, March 21, it just fills up the energy storage system of 420-MW(e)-hr capacity while simultaneously generating net power of 100 MW(e). This permits the plant to operate directly from insolation at 100 MW(e) net for approximately 4.5 hr on the day of worst heat collection, December 21. The resulting capacity factor of the plant, based on year-round clear-day insolation levels at Barstow, California, is 45.6%. Flat land has been assumed [232 hectares (573 acres) is required]; however, the design is adaptable to modest slopes.

Energy is collected in the receiver and transported through the field piping using molten draw salt as the fluid medium. Draw salt is a 50% molar mixture of sodium nitrate and potassium nitrate. It melts at 494 K (429°F) and may be employed up to at least 867 K (1100°F). The high density of draw salt plus its good specific heat, reasonable conductivity and viscosity, and modest cost [33.1 cents/kg (15 cents/lb)] make it a suitable medium for transporting and storing thermal energy.

The field piping is arranged (see Figure 4.4-3) so as to minimize the hot piping runs, because the more expensive 316 stainless steel is employed on that side and the less expensive carbon steel on the warm legs. The energy storage system (ESS) is decentralized, which assists in minimizing piping

costs. The ESS comprises two hot tanks, each 18.6 m in diameter by 11.0 m high (61 ft in diameter by 36 ft high) and two warm tanks 17.7 m in diameter by 10.7 m high (58 ft in diameter by 35 ft high). The tanks are unpressurized, although the draw salt is covered by a controlled atmosphere of dry air. The tanks are isothermal (well mixed); no thermoclines are utilized.

Salt is pumped from the hot tank to the centrally located steam generator, which comprises two preheater units, four boilers employing a common drum, two superheaters, and one reheater, the latter two sections being in parallel on the salt side. The materials employed in the steam generator are principally carbon steel and 347 stainless steel. Feedwater is supplied to the preheater at 18.6 MPa (2706 psig) and 506 K (450°F). Steam leaves the superheater at 18.1 MPa (2630 psig) and 811 K (1000°F).

As shown in Figure 4.4-3, the field pumps are located in the warm leg and the steam generator pumps are located in the hot leg. The pumps are each a bank of three 50% capacity units. Each unit is a vertical multistage centrifugal pump driven by a hydraulic motor to achieve a wide range of controlled speed and flow. Overall pump efficiencies are in the 80% to 85% range.

The steam turbine is a commercially available tandem-compound unit operating at 3600 rpm and rated at 106.2 (MW(e) gross. At maximum guaranteed load, the throttle steam conditions are 16.5 MPa/811 K (2400 psig/1000°F) and reheat steam conditions are 2.90 MPa/811 K (421 psig/1000°F). The steam exhausts to a condenser vacuum of 10.2 kPa (3 in. Hg). The steam power conversion cycle employs seven stages of feedwater heating to achieve a turbine gross heat rate of 8509 kJ/kW-hr (8072 Btu/kW-hr), i.e., a gross EPGS efficiency of 42.3%. The net output of the plant at equinox noon (when the field pump power is highest) is 100.5 MW(e); when the plant operates at design power with heat supplied directly from the collector field, the net output rises to 101.0 MW(e). The net EPGS efficiency (accounting for station auxiliary loads) in the latter condition is 39.6%.

The LFSCPS has a year-round average efficiency during the insolation interval of 12.9%. The maximum efficiency achieved is 23.8% on March 21 at noon.

The present cost of the first plant, excluding owner's costs except for land, is estimated to be \$287.0 million. Using this cost estimate, the bus bar cost of electricity (BBCOE) using the DOE-supplied methodology is calculated as 151 mills/kW-hr. When the plant capacity is increased to 80% by increasing the field size and energy storage volume, the BBCOE drops to 143 mills/kW-hr.

The plant cost is considered to be a realistic estimate for the given design. The labor rates, burdens, indirect costs, contingencies, and fee are based on extensive experience by Bechtel in constructing plants having reasonable analogies to the LFSCPS. However, the given design contains areas of high cost which may be reduced by suitable redesign. Identified means to reduce cost include factory instead of field fabrication of the concrete concentrator panels, the possible use of a lighter-weight material for the panels, and simplification of the heat transport system, which comprises concrete-lined trenches covered with checkered steel plate.

The plant, as designed, is durable and conservatively engineered and costed. It is anticipated that it will be competitive with other solar plants when placed on the same basis for comparison.

1.3 BACKGROUND AND PRIOR ART

The design of the LFSCPS, summarized in the preceding section, has been based on two key items:

1. The use of the GA solar collector concept called the fixed mirror solar concentrator (FMSC).
2. The use of draw salt as the heat transport and storage fluid.

The background of these items is described below.

The FMSC concept was conceived at GA in 1971. It comprises a fixed array of long, narrow, flat reflecting mirror strips arranged on a semicylinder to concentrate the sun's light as illustrated in Figure 1.3-1. The mirror array produces a narrow focal line that follows a circular path as the sun moves, and the focal line is tracked by the moving heat receiver. The principles of the concept are illustrated in Figure 1.3-2. Each mirror strip is positioned with its center on the semicircle and oriented so that its normal is offset with respect to the reference or tangent facet normal by an amount that is one-fourth of the angle between the mirror strip and the reference facet. The chord length between the two edges of the semicylinder segment is called the mirror aperture and is so referred to throughout this report.

Early GA-supported work was directed to the development of the principles of the fixed mirror system and construction of two research models to corroborate the theoretical analysis (Refs. 1.3-1 and 1.3-2). Arizona State University then carried out a test program on a one-tenth scale model of the FMSC, which satisfactorily demonstrated the system principles (Ref. 1.3-3).

A study program was then initiated under ERDA sponsorship to develop information leading to the design and evaluation of the FMSC. Under this

Fig. 1.3-1. FMSC geometry

Fig. 1.3-2. Principal angular relationships of the FMSC

program, overall system parameters, the interrelationship between coolant fluid and mirror geometry, fluid piping design, and economics, and the effect of plant size upon overall cost were studied. Trade-offs in methods of suspending the heat receiver assembly were considered, and a conceptual design was evaluated. The means for supporting and rotating the heat receiver assembly were studied. A preliminary mechanical design of the heat receiver was completed, and its thermal performance was analyzed for several thermal configurations and pipe surface solar and thermal properties (Ref. 1.3-4).

A conceptual design of an FMSC was then prepared under a joint utility program sponsored by Arizona Public Service Company, Southern California Edison Company, and Utah Power and Light Company. The design, directed toward demonstrating modular hardware, featured a mirror unit 61 m (200 ft) long with 6.4-m (21-ft) wide aperture and a traversing heat receiver assembly (HRA), suspended from a beam mounted on five support arms. Various methods of supporting the HRA and mounting the mirror strips were investigated in this program (Ref. 1.3-5).

The ERDA- and utility-sponsored R&D programs resulted in the important conclusion that the most economical way to support the mirror strips for power applications was with self-supporting above-ground concrete assemblies. This conclusion led to consideration in subsequent studies of relatively modest size FMSC modules, each having a mirror area of about 23.2 m^2 (250 ft^2).

An investigation of a 100-MW(e) solar power station based on the FMSC was then conducted for the Electric Power Research Institute (EPRI) (Ref. 1.3-6). Alternative coolants were considered in the design comparisons of this investigation. A computer code developed in this study incorporated analytic models for the cost and performance of the major system components in the FMSC power plant with appropriately different arrangements for the coolant fluids selected, which were heat transfer salt and CO_2 . Influence functions identifying those items leading to significant changes in plant

cost and performance were generated with this code. Specifically, these studies concluded that gas cooling of a distributed receiver system such as the FMSC was uneconomical. Heat transfer salt, on the other hand, showed technical and economic merit. This conclusion was in conformance with other GA studies of the transport and storage of heat using heat transfer salt (Refs. 1.3-7, 1.3-8).

After completing the above investigation for EPRI, GA was selected to provide an FMSC array for the Total Energy Demonstration Facility at Sandia Laboratories, Albuquerque (Refs. 1.3-9, 1.3-10, 1.3-11). This array comprised 260 m^2 (2800 ft^2) of mirror surface arranged as two 61-m (200-ft) long rows of FMSC modules oriented in the east-west direction. General Atomic prepared the preliminary and final designs of this array, had the components fabricated and assembled, and made the system operational. This array, illustrated in Figure 1.3-3, utilizes oil (Therminol 66) as the heat transport fluid. Oil is heated from 517 K (470°F) to 589 K (600°F) in the array and delivered to the Sandia hot oil storage tank.

The array at Sandia Laboratories was the first significant demonstration of the FMSC technology, particularly the concrete concentrator panels, the insulated receiver with the Winston type secondary concentrator, and the simple receiver drive mechanism. The array was clearly a demonstration unit and confirmed that the FMSC had a performance potential competitive with steerable parabolic systems. The design exemplified in the array was not, however, suited for low-cost mass production. It was based on soft tooling, was not optimized, and was directed to lower delivery temperatures than required for the LFSCPS. Further, various minor performance and mechanical difficulties arose with that design. Mechanical design studies to address these various questions were thus necessary and have been undertaken at GA. The costs of the 260-m^2 (2800-ft^2) array were high on a unit area basis, but their distribution was helpful in identifying items of high cost which required redesign to meet the cost goals of this program.

Fig. 1.3-3. FMSC installation at Sandia Laboratories, Albuquerque

Background effort at GA related to high temperatures in line focus receivers has included two programs. One was a study (Ref. 1.3-12) for EPRI to investigate the heat losses from receivers for the FMSC at temperatures up to 867 K (1100°F). Simulated receivers and electrical heating were employed in order to develop parametric heat loss data as a function of various geometrical factors and wind speed/direction. General Atomic was also awarded a contract by DOE to construct a solar test loop using heat transfer salt as the fluid medium. This test rig has been completed and has been operated at temperatures in excess of 811 K (1000°F). The design experience with the rig provided useful information on materials selection, valve and pump suppliers, trace heating requirements, and safety and operational factors with draw salt. Figures 1.3-4 and 1.3-5 illustrate the solar salt test loop.

This experience has been integrated into the investigations described in this report.

REFERENCES

- 1.3-1 Russell, J. L., Jr., E. P. DePlomb, and R. K. Bansal, "Principles of the Fixed Mirror Solar Concentrator," General Atomic Company Report GA-A12902 (Rev.), February 1, 1977.
- 1.3-2 Russell, J. L., Jr., "Investigation of a Central Station Solar Power Plant," General Atomic Company Report Gulf-GA-A12759, August 31, 1973.
- 1.3-3 Wood, B. D., R. Pinon, and C. E. Backus, "Evaluation of a Fixed Mirror Solar Concentrator," Arizona State University Report ERC-R-78040, 1978.
- 1.3-4 Potthoff, R., et al., "Investigation of a Central Station Solar Power Plant, Final Report," ERDA Report GA-A13821, General Atomic Company, February 26, 1976.
- 1.3-5 Walker, W. E., "Conceptual Design of a Demonstration Fixed Mirror Solar Concentrator, Final Report," General Atomic Company, unpublished data, July 1, 1976.

Fig. 1.3-4. FMSC array installed in molten salt test loop

Fig. 1.3-5. Molten salt storage and supply system

- 1.3-6 Russell, J. L., Jr., et al., "Preliminary Systems Analysis of a Fixed Mirror Solar Power Control Station," Electric Power Research Institute Report GA-A13974 (Rev.), General Atomic Company, February 15, 1977.
- 1.3-7 Huntsinger, J., et al., "Process Heat in Petroleum Refinery Applications, Final Report" ERDA Report GA-A13406, General Atomic Company, February 20, 1976.
- 1.3-8 Vrabie, D. L., and R. N. Quade, "HTS Thermal Storage Peaking Plant," ERDA Report GA-A14160, General Atomic Company, April 1977.
- 1.3-9 Eggers, G. H., et al., "Solar Collector Field Subsystem Program on the Fixed Mirror Solar Concentrator, Final Report for Period March 28, 1976, through September 30, 1976," Sandia Laboratories Report GA-A14209, General Atomic Company, December 31, 1976.
- 1.3-10 Eggers, G. H., et al., "Installation and Startup of the Fixed Mirror Solar Concentrator Collector Field Subsystem, Final Report for the Period January 31, 1977 through March 31, 1979," Sandia Laboratories Report SAND-79-7015, General Atomic Company, to be issued (GA-A15344).
- 1.3.11 Dudley, V. E., and R. M. Workhaven, "Summary Report, Concentrating Solar Collector Test Results, Collector Module Test Facility," DOE Report SAND-78-0815, Sandia Laboratories, May 1978.
- 1.3-12 Walker, W. E., J. J. Housman, and J. L. Russell, "Measurement of Heat Loss from a Heat Receiver Assembly of a Fixed Mirror Solar Concentrator," Proceedings of the 13th IECEC Conference, San Diego, California, August 1978, Volume Two, Society of Automotive Engineers, 1978, p. 1554.

1.4 SUMMARY OF INVESTIGATIONS CONDUCTED

The first task in the program was the selection of the heat transport fluid, which is discussed in Section 2.1. After some preliminary screening, three fluids were studied in detail: draw salt, Therminol 88, and sodium. The studies were performed by preparing a point design for each system that met the program design requirements and then comparing the capital and operating costs of the three designs. Operational and development questions associated with each fluid system were also identified. A ground rule in the designs was that the heat transport fluid would also be utilized in the thermal storage system, since it permitted higher plant efficiencies when operating from storage than did indirect storage methods. This had been identified as a key performance problem in the first generation central receiver solar power system design. This ground rule was found to distinctly favor draw salt, which was the heat transport fluid chosen for the program.

Draw salt, the selected fluid, is a 50% molar mixture of potassium nitrate and sodium nitrate. The literature on draw salt was reviewed briefly, and a set of its physical properties was prepared for use by GA and its subcontractors on this program. This data set is given in Section 2.1. A brief survey of industry was undertaken, and plants having long operating experience with draw salt or Hitec* were identified, including one with temperature levels in excess of the 843 K (1057°F) proposed here. Pertinent operational experience from the plant was incorporated into this design report.

A review of experience gained from the 260-m^2 (2800-ft^2) FMSC array installed at Sandia Laboratories in Albuquerque, New Mexico, indicated that the FMSC efficiency was possibly underpredicted owing to conservative allocations of energy loss terms and to the simplification of the receiver

*Similar composition to draw salt, but with sodium nitrite addition. Hitec is made by the DuPont Chemical Company.

modeling. Analysis to quantify these loss terms and to refine the receiver model was therefore undertaken and is reported in Section 2.2. The resulting updated design code, SUNPOW, was employed in the overall system design code, SS2, along with the receiver and concentrator cost models.

The heat transport subsystem (HTS) investigations are described in Section 2.3. Various alternative piping arrangements were investigated. Significant conclusions of the investigation were that:

1. Piping cost economies would result if the collector row length were made as long as possible.
2. The piping should be arranged so that hot manifolds and return lines, having the more expensive alloy steel piping, are shorter than the warm manifolds and supply lines made of carbon steel.
3. The thermal storage should be decentralized to take advantage of the fact that piping between storage and steam generator need be designed only for salt flow corresponding to a solar multiplier of 1.

The insulation thicknesses were investigated to achieve cost optimal values using influence functions for the impact on the rest of the plant. A balance was found between the piping thermal losses and the salt pumping power. The sum of these two losses represents only about 3% of the total heat gathered in the collector field. A cost/performance model of the HTS was prepared for use in the subsystem design code. This work is also discussed in Section 2.3.

Section 2.4 describes the parameter investigation of the energy storage subsystem (ESS). The investigation encompassed a rock-filled salt thermocline, an all-salt thermocline, and separate hot and warm tanks of pure salt. The latter arrangement was recommended because it is operationally the simplest and its computed cost is only modestly more than that of the others without taking any credit for the loss of availability that occurs

in thermoclines at the end of their discharge cycle. A preliminary cost performance model of the ESS was prepared for use in the system design costs.

The parametric analysis of the electric power generating subsystem (EPGS) is described in Section 2.5. In this study, heat balances were determined for a 100-MW(e) plant at a range of feedwater temperatures, reheat/superheat temperatures, and throttle pressures based on a single reheat steam cycle. A constant approach temperature was assumed in the parallel reheater and superheater, a constant pinch point temperature at the boiler (water) inlet, and a constant condenser vacuum. Reference EPGS parameters selected as a result of these investigations led at that point in the program to a hot salt supply temperature of 839 K (1050°F). Further heat balances were then performed in which the approach temperatures, pinch point temperature difference, and condenser vacuums were varied. Using cost influence functions for the remainder of the plant, optimal parameters were selected. Studies of plants larger than 100 MW(e) net were then conducted using the same EPGS operating parameters. Preliminary EPGS costs were developed for the reference plant and for the larger plants to provide a data base for scaling purposes. A cost/performance model incorporating these data was prepared for the system design code. It was also established in these studies through contact with turbine manufacturers that the selected 100-MW(e) turbine of 16.5-MPa (2400-psi) pressure and 811 K (1000°F) temperature with a single reheat is commercially available and may be cycled daily.

A discussion of master control subsystem (MCS) philosophy is presented in Section 2.6 together with the reason for choosing the adopted mixed mode control scheme.

The system design and analysis was performed to a large extent by means of the SS2 code. It was found relatively early in the studies that it was cost effective to increase the hot salt temperature to the maximum

practical extent, i.e., limited by either thermal stability or material corrosion. A nominal limit of 839 K (1050°F) was therefore selected. It was also found cost effective to reduce the warm salt temperature as much as possible in order to reduce HTS and ESS requirements until a practical lower limit was reached. This practical lower limit should be either to provide an adequate margin to prevent freezing of the draw salt under certain off-design conditions (e.g., startup) or to ensure that the warm salt is hotter than the incoming feedwater. The latter condition was found to be the limiting criterion. As a consequence, it was more practical to optimize the EPGS independently of the rest of the plant. This is discussed in Section 2.7. The upshot of this practical approach was to limit the role of SS2 to determining the most cost effective parameters for the concentrator and receiver subsystems and to provide the means for investigating BBCOE as a function of plant rating and/or capacity factor, the latter through varying the size of the ESS.

Section 3.2 describes the SS2 code and the cost and performance models of the various subsystems. The optimization results obtained are discussed in Section 3.3 in conjunction with the selection of parameters for the primary concentrator and receiver subsystem. The selection was based on interfacing practical limitations such as manufacturing feasibility on the system optimization results. The sensitivity of the plant design to the choice of key optical parameters is also described in Section 3.3.

Utility hourly load data supplied by Southwestern Public Service Company (SPS) are presented in Section 3.4. These data showed several items of interest:

1. The peak daily loads occurred typically some 5 hr after solar noon. Thus, energy storage would be required for a solar plant in the SPS grid not just to change plant capacity, but to achieve a necessary translation in time.

2. The peak loads occurred in July and August. This is significant because the selected FMSC design is optimized to maximize the total energy collection during the year and not for the summer months.

Southwestern Public Service also supplied hourly incremental cost of electricity generation information. The data showed that for the SPS system, there is only a modest variation of the incremental generating cost with peak load. This in turn indicates that it may be preferable to design the solar plant for maximum yearly performance rather than for good summer performance.

Overall conclusions from examinations of the SPS data are that each utility's requirements for a solar electric plant will be highly specific. Key areas affecting requirements which differ from utility to utility include:

1. Site availability (proximity to load demand, adequate land area, and topography for plant).
2. Utility load profiles (storage requirements, summer or year-round optimized plant).
3. Fuel costs and present generating capacity mix (displacement by solar electric units of plants having no assured long-term fuel supplies of low costs).
4. Environmental factors (primarily insolation levels and secondarily frequency of snow, hail, and dust storms).

Section 3.5 provides an overview of the response of the FMSC to environmental conditions. In particular, a summary is provided on the hail damage to the FMSC array at Sandia Laboratories. The summary discusses what took place, why it occurred, and what action will ensure that such damage does not occur in future plants.

The collector row length is a key system parameter. System economics indicate that it should be as long as possible and limited only by land topography or access for maintenance from one row to another. However, a further limit on collector row length arises from the response of the collector field to transients during daily plant startup or following complete or partial intermittent cloud coverage. A computer code to calculate these transients was written (based on an earlier version for the array at Sandia) and transient data were calculated. The data discussed in Section 3.6 show that with proper control, collector row lengths in excess of 274 m (900 ft) would be permissible. The data also displayed a significant finding, namely that the plant, as designed, is relatively slow to start up and to recover from cloud coverage. The former is somewhat ameliorated by the fact that insolation levels early in the morning, insufficient to sustain design conditions, do provide heat that assists in the plant startup. Further, bringing the steam turbine onto line from its bedded-down condition takes a similar time duration (1.6 hr), although the operation may utilize stored instead of direct solar energy. Recovery from clouds may be facilitated if means for reducing the temperature fall-off of the receiver in the absence of insolation can be found. Such means are described in Section 3.6. In general, these transient problems are not fully resolved, a situation paralleled in the design of central receiver systems.

Section 4 presents the conceptual plant design. For completeness, the overall plant design requirements are given in Appendix I. Operating and physical parameters of each subsystem are given together with the equipment lists which form the basis for the plant cost estimate. The overall arrangement (plot plan) of the plant is presented together with flow diagrams illustrating the integration of the HTS and ESS auxiliaries.

Section 4.2 presents details of the selected concentrator subsystem. The design studies of alternate concentrator panels and design questions pertinent to the subsystem selection are discussed in this section. Of key importance was the choice of concrete as the material for the concentrator panel. This choice reflected largely that it appeared suitable to meet cost goals and that it was more credible than advanced alternatives such as fiber-glass-reinforced foamed plastic.

Section 4.3 gives details of the selected receiver subsystem and the alternate approaches considered with respect to the receiver support, receiver drive, and actuating and aiming mechanisms. Of key importance in the design was obtaining as long a receiver span as possible while still meeting sag requirements and satisfactory response to the environmental conditions (wind, seismic). The use of a lightweight and low-cost trussing structure permitted these objectives to be attained and was therefore adopted. The selection of materials for the various receiver elements is also discussed in Section 4.3.

Details of the selected HTS are described in Section 4.4. An equipment list is provided which is the basis of the HTS cost estimate. Design studies of the HTS and its auxiliaries are also presented. The steam generator design consists of dual preheaters feeding four boiler units which have a common drum. This is followed by a superheater and reheater which are in parallel on the salt side. The design of the steam generator is not considered to be a significant development item.

Operational aspects of the HTS considered in Section 4.4 include piping startup in the morning and trace heating requirements. Heat losses of the HTS are discussed as are the means for substantially reducing the quoted heat loss values. A description of backup fossil-fired salt heaters which improve the availability of the plant so that it can receive full capacity credit is also presented.

Section 4.5 describes the ESS design comprising two warm tanks and two hot tanks of salt at decentralized locations within the collector field. The design is relatively straightforward. The tank walls are externally insulated so that they generally operate at the contained fluid temperatures. The ESS also contains the salt purification and charging subsystems, and brief descriptions of these are also presented. The ESS is seen to be a low-cost subsystem, comprising only 3% of the total plant costs. Thus, it may be readily enlarged to improve plant capacity factors and system economics.

Section 4.6 gives details of the selected EPGS design together with equipment lists upon which the plant cost estimate is based. Main steam piping diagrams, EPGS arrangement, and support system piping and instrumentation diagrams are also presented. The EPGS design of 811 K (1000°F) superheat/reheat temperature and a condenser vacuum of 10.2 kPa (3 in. Hg) with a throttle pressure of 16.5 MPa (2400 psi) is a current industry standard for which equipment is readily available. Thus, a key element of this solar plant design is that it does not require a new standard or to-be-developed power conversion cycle. The modern steam conditions plus extensive feed-water heating permit a gross station efficiency of 42.1% to be achieved. After subtraction of HTS and ESS heat losses and salt pumping power, the net station efficiency is 39.6%. This value is fully consistent with those predicted for central receiver systems. Section 4.6 also addresses the startup and shutdown of the turbine from the cold condition and from the overnight bedded-down condition.

Section 4.7 describes the master control system and how it functions to integrate the control of the various subsystems into a smoothly running plant. A key element in the control is the use of the ESS to isolate the steam generator and EPGS from the rest of the collector field. This buffering of the solar supply from the energy demand greatly simplifies control and permits reduction of transients on components containing salt. An overall fluid management strategy is therefore required to implement the control philosophy. The elements of this strategy are outlined in Section 4.7.

The plant performance and cost estimates are given in Section 5. The cost estimate of \$287 million is for the first plant and was obtained late in the program, so opportunities to review the design to reduce its cost were limited. Means for reducing the plant cost by a combination of design improvement, performance improvement, and manufacturing improvement have been identified, but their cost impact has not been estimated by GA in view of the considerable design extrapolation required. The capital cost has been translated into BBCOE using the methodology given by DOE and listed in Appendix I to arrive at a value of 151 mills/kW-hr for the first plant. The basis for the operation and maintenance (O/M) cost of the plant is given in Section 5.3.

The yearly average solar-to-electricity efficiency of the plant is 12.9% based on clear day insolation levels at Barstow. Peak plant efficiency and peak daily efficiency occur at equinox and are 23.8% and 13.7%, respectively. These values are slightly lower than predicted previously (Ref. 1.4-1) owing to the adoption of a more conservative set of optical properties for the selective coating on the receiver.

A brief analysis of the construction schedule for the plant is presented in Section 5.3. It is concluded that a 3-yr construction schedule for this size plant could be attained, but that no benefit would occur from incremental startup because of the present long-term delivery of key EPGS components. However, a corollary of the short construction time is that the escalation costs and allowance for interest during construction are less than for a comparable coal-fired or nuclear plant, which modestly offsets at the utility owner's level the otherwise high capital cost per unit kilowatt of the solar plant.

Good long-term behavior of mirrors glued to concrete is an inherent assumption in the plant design. The validity of the assumption is being tested by mirrors on various samples placed in the environmental testing chamber at Sandia Laboratories, Albuquerque. The preparation of the samples and the test objectives and procedures are described in Section 6.

An assessment of commercial-scale plants is presented in Section 7. Section 7.1 discusses potential performance and cost improvements and economies of scale. Sections 7.2 and 7.3 present an assessment, performed by SPS, of the solar plant design detailed in Sections 4 and 5. As mentioned earlier, the specific requirements of each utility will impact the plant design. In the case of SPS, which presently has reasonably low fuel costs and has insolation levels approximately 80% of those at Barstow, California, the LFSCPS does not look economically attractive. From an environmental and licensing viewpoint, SPS views the solar plant favorably.

Section 7.4 presents the results of a marketing assessment performed by investigating the conditions (fuel cost, interest rates, etc.) under which solar plants would become attractive. A combined dispatching and generation planning computer code was written for this purpose and utilized EPRI synthetic utility data (Ref. 1.4-2) for its bases. However, the code was not utilized because the first plant cost was too high to achieve any market penetration. "Nth" plant capital costs of approximately half the \$287 million value calculated here will be necessary, coupled to significant escalation of conventional fuels. Market penetration will then occur because:

1. Solar plants will then be more economical to operate than fossil-fueled power peaking and intermediate load plants.
2. Solar plants tend to displace plants which are obsolete or relatively inefficient and thus may be credited with the fuel savings from these retired plants.
3. Solar plants provide a measure of protection from inflation because they have no escalating fuel cost, which offsets their initial capital cost.

General Atomic has with its own funds briefly summarized the key development issues of the LFSCPS in Section 8. The receiver selective coating development, the demonstration of the concentrator fabrication technology, and the

verification of the advanced concentrator and receiver subsystem performances are the key items. Other issues of less significance but not yet resolved are also addressed in this section.

In summary, an extensive investigation of an advanced solar plant design based on GA's FMSC has been conducted and is reported fully in the following sections. The design and engineering data provide a firm foundation for future development of this plant and other systems using the FMSC or using draw salt as the heat transport and storage media. The design shows that cost reduction occurs as the system capacity factor increases owing to an efficient low-cost thermal storage in which the fluid medium is also the heat transport fluid. The cost of the field piping is found to be relatively low, only 16% of the total cost, and not a limiting factor in the distributed receiver concept at the 100-MW(e) level. The modularity of the design simplifies development; hence, design verification costs and system demonstration costs are low. No significant technical problems are perceived for the design except the development of a reliable selectively solar absorptive coating for the receiver; however, ongoing research elsewhere is already indicating promising results on such a coating. The temperature transient problems typical of centralized receiver systems are not present in the design described herein. Finally, the modular design permits a ready adaptability to applications other than central power stations, such as process heat for the range 533 K to 811 K (500°F to 1000°F), and in size ranges from 293 to 1.39×10^5 kJ/s (1 to 500 MMBtu/hr). This diversity of plant sizes and end uses offers high incentive for the continued development of this design.

REFERENCES

- 1.4-1 Schuster, J. R., et al., "Line Focus Solar Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.
- 1.4-2 Zaininger, H. W., et al., "Synthetic Electric Utility Systems for Evaluating Advanced Technologies," Electrical Power Research Institute Report EM-285, February, 1977.

2. PARAMETRIC ANALYSIS

2.1 HEAT TRANSPORT FLUID

2.1.1 Introduction

General Atomic (Refs. 2.1-1, 2.1-2) and others have performed previous studies of fluids for transporting heat from solar collectors having distributed receivers and from very-high-temperature reactors in refinery applications. In both studies, gases were found to require large uneconomic pumping powers; and in both cases, the recommended fluid was a molten salt. It was appropriate to review this choice for the application to a 100-MW(e) solar electric plant since thermal storage was required in addition to heat transport. This study was performed by developing representative point designs for three selected fluids and intercomparing plant economics and operations.

The review of heat transport fluids was of necessity quite brief and performed early in the program so as not to delay the design work. Consequently, the range of fluids considered was limited and based on the results of the prior work. Table 2.1-1 lists the fluids considered; of these, only Therminol 88, sodium, and draw salt were investigated in detail for reasons that are presented below.

Therminol 66 (T-66) is a heat transport oil manufactured by the Monsanto Company. It has the virtue that it is liquid at room temperatures. However, its pour point is 246 K (-18°F), so flow blockage problems could arise on the coldest days [lowest ambient temperature of 233 K (-40°F)]. On the upper end of the scale, T-66 is limited to a long-term bulk temperature of 617 K (650°F) and a skin temperature of 644 K (700°F). Thus, it does not appear to have the capability to transport heat at sufficient temperature to produce steam continuously at 644 K (700°F), a specified requirement for this program.

TABLE 2.1-1
HEAT TRANSPORT FLUID ALTERNATIVES

Fluid	ρC_p [kJ/m ³ ·K (Btu/ft ³ -°F)]	Reasons For Not Considering Further
Therminol 66	2059 (30.7)	Insufficient high-temperature capability
Therminol 88	2059 (30.7)	--
NaK (43.3 wt % K)	818 (12.2)	More expensive than sodium; poorer heat transport properties
Sodium	1080 (16.1)	--
Silicone oil	1301 (19.4)	More expensive than T-88
Hitec	2749 (41.0)	More expensive than draw salt with reduced temperature capability
Draw salt	2749 (41.0)	--

Therminol 88 (T-88), which has higher temperature capabilities, was therefore selected as a suitable hot oil that would meet the system requirements despite the inconvenience of a higher pour point of 418 K (293°F).

NaK (43.3 wt% potassium) has the virtue of a low melting point of 279 K (43°F). It is considerably more expensive than pure sodium and more chemically reactive, with water in particular. In addition, NaK has a lower density, a lower specific heat, and a lower thermal conductivity, so its heat transport and storage properties are poorer. Since NaK will also freeze on the coldest specified day, pure sodium was chosen as a suitable liquid metal for the parametric analysis.

Silicone oil, such as Dow Corning Syltherm-800, has the virtue of a wide liquid range. Its pour point is 224 K (-57°F), and its upper temperature from a thermal stability viewpoint is not known but is believed to be in excess of 672 K (750°F). Syltherm-800 is an inert substance that does not react with iron, copper, or aluminum. However, its flash point of 428 K (310°F) and its fire point of 467 K (380°F) are nominally lower than those of T-88. In addition, it is priced at about \$4.41/kg (\$2/lb) with little likelihood that the price will drop with solar plant application because the fluid is already quite widely used in industry. The cost factor and the fact that its volumetric specific heat of $1.3 \times 10^{-3} \text{ kJ/m}^3 \cdot \text{K}$ (19.4 Btu/ft³ · °F) is considerably less than that of T-88 were sufficient reasons not to consider silicone oil further in this parametric analysis.

Hitec is a salt made by the DuPont Company that has the virtue of a lower melting point of 416 K (288°F) than draw salt with 494 K (429°F). Hitec is 7% NaNO₂, 40% NaNO₃, and 53% KNO₃ compared with 45% NaNO₃ and 55% KNO₃ for draw salt. The addition of the sodium nitrite considerably increases the cost of the fused salt. In addition, Hitec is believed to have a poorer high-temperature thermal stability than draw salt. Consequently, draw salt was chosen as the representative molten salt for this parametric analysis.

2.1.2 Fluid Properties

Therminol 88 (T-88) data were supplied by the Monsanto Company. The fluid comprises mixed terphenyls plus additives. It has a bulk temperature limit of 672 K (750°F) and a melt point of 418 K (293°F). Its volumetric specific heat at the midpoint of a suitable operating range is $2.06 \times 10^3 \text{ kJ/m}^3 \cdot \text{K}$ (30.7 Btu/ft³·°F). Therminol 88 is a hydrocarbon, with a flash point of 464 K (375°F) and a fire point of 511 K (460°F). Its auto-ignition point is greater than 811 K (1000°F). The price of T-88 in bulk quantities was not well established but is now believed to be about 82 cents/kg (37 cents/lb). It was quoted by the Monsanto sales staff as \$1.43/kg (65 cents/lb) in carload lots of 10,200 kg (22,500 lb). On the other hand, the price of bulk diphenyl in carload lots is quoted in the October 1978 Chemical Marketing Reporter as 36 cents/kg (16-1/2 cents/lb) (molten, 96% purity) and 82 cents/kg (37 cents/lb) [34-kg (75-lb) bags, 99% purity]. For the purposes of this study, a T-88 price of 55 cents/kg (25 cents/lb) was assumed.

Draw salt is taken as a 50% molar mixture of potassium and sodium nitrates. Since draw salt was the fluid selected for the present investigation, more effort was expended to develop its physical properties than those of other fluids.* The property data set for draw salt in Ref. 2.1-4 was not utilized because:

1. It does not cover the desired temperature range [from the melting point up to somewhat in excess of 839 K (1050°F)], and a uniformly consistent data set is desired.
2. It contains an inconsistency in the viscosity column.

The data set given in Table 2.1-2 was generated for this study, and equations describing the property dependence with temperature were also generated. Density data are from Ref. 2.1-5, p. 676, with suitable extrapolation in the range 720 to 880 K [836° to 1124°F]. Viscosity data are

*The most up-to-date evaluated data are given in Ref. 2.1-3.

TABLE 2.1-2
DRAW SALT^(a) PROPERTIES

Temp [K (°F)]	Density [kg/m ³ (lb/ft ³)]	Viscosity [10 ⁻³ Pa·s (centipoise)]	Conductivity [10 ⁻⁴ kJ/s·m·K (Btu/hr-ft-°F)]	Specific Heat [kJ/kg·K (Btu/lb-°F)]
Molten				
520 (476)	1928 (120.36)	4.725 (4.725)	0.369 (0.23)	1.56 (0.373)
560 (548)	1906 (119.00)	3.506 (3.506)	0.369 (0.23)	1.56 (0.373)
600 (620)	1883 (117.54)	2.657 (2.657)	0.369 (0.23)	1.56 (0.373)
640 (692)	1858 (115.98)	2.100 (2.100)	0.369 (0.23)	1.56 (0.373)
680 (764)	1831 (114.31)	1.755 (1.755)	0.369 (0.23)	1.56 (0.373)
720 (836)	1803 (112.55)	1.540 (1.540)	0.369 (0.23)	1.56 (0.373)
760 (908)	1776 (110.84)	1.377 (1.377)	0.369 (0.23)	1.56 (0.373)
800 (980)	1748 (109.12)	1.241 (1.241)	0.369 (0.23)	1.56 (0.373)
840 (1052)	1721 (107.40)	1.132 (1.132)	0.369 (0.23)	1.56 (0.373)
880 (1124)	1693 (105.69)	1.050 (1.050)	0.369 (0.23)	1.56 (0.373)
Melting Point = 494 K (429°F) Heat of Fusion = 81.4 kJ/kg (35 Btu/lb)				
Solid				
300 (80)	1922 (120.00)	--	0.339 (0.21)	1.34 (0.32)
400 (260)	1922 (120.00)	--	0.339 (0.21)	1.34 (0.32)

(a) 50% molar NaNO₃, i.e., 54.6% KNO₃, 45.4% NaNO₃.

from Ref. 2.1-5, p. 679, with suitable extrapolations for the range 720 to 880 K (836° to 1124°F). Specific heat data were taken to be the same as those for Hitec (Ref. 2.1-6), since no data for draw salt were discovered in the brief literature survey performed and Hitec is close to the draw salt composition. Conductivity data were not initially found for draw salt. Reference 2.1-6 quotes a range for Hitec from 2.73 to 5.63×10^{-5} kJ/s·m·K (0.17 to 0.35 Btu/hr-ft-°F), and the scatter suggested that impurities could affect the conductivity. Impurities such as carbonates, oxides, and hydroxides can readily arise from exposure of hot draw salt to air. Since the data of Ref. 2.1-4 reflect practical experience in constructing heat transfer systems using either Hitec or draw salt, the conductivity values of Ref. 2.1-4 (although on the low side and therefore conservative) were utilized. Subsequently, a complete conductivity data set for draw salt was discovered (Ref. 2.1-7), but it has not been utilized in the design calculations of this Phase I program because system performance is not very sensitive to changes in the draw salt conductivity. A factor of two change in the conductivity was found to change the FMSC performance by less than 1%.

Draw salt has a melting point of 494 K (429°F), the highest of the three fluids studied in detail. It has a volumetric specific heat of 2749 kJ/m³·K (41.0 Btu/ft³-°F) in the middle of its operating temperature range, also the highest of the three fluids. Long-term stability at temperatures up to 867 K (1100°F) requires careful control of the cover gas composition. Draw salt is a very strong oxidant and will react explosively with finely divided carbon. Therefore, all carbonaceous material must be removed from points where leaks could occur.

Draw salt can be made from its component materials, which are available in bulk. A cost analysis was performed by the Energy Systems Group of Rockwell International, who concluded that a bulk price of 27.6 cents/kg (12-1/2 cents/lb) was reasonable. Assuming a West Coast site for importing sodium nitrate, a further 4.41 cents/kg (2 cents/lb) was added for transportation,

giving a total cost of 31.97 cents/kg (14-1/2 cents/lb).^{*} Martin Marietta used a cost of 25.35 cents/kg (11-1/2 cents/lb) in their studies (Ref. 2.1-4).

Sodium data were taken from the Liquid Metals Handbook (Ref. 2.1-8).

Sodium melts at 371 K (208°F) and can be used at quite elevated temperatures. At the upper operating temperature of 867 K (1100°F) chosen for this parametric analysis, sodium has a vapor pressure of 20 mm Hg. The volumetric specific heat of sodium at the midpoint of its operating temperature range is $1080 \text{ kJ/m}^3 \cdot \text{K}$ ($16.1 \text{ Btu/ft}^3 \cdot ^\circ\text{F}$), the lowest of the three fluids studied in detail.

Bulk sodium is readily available since it is commonly used industrially, and a bulk price of 90.39 cents/kg (41 cents/lb) was utilized. Sodium will react explosively with water at all temperatures. Its fire point is 478 K (400°F). It is an alkaline metal that reacts readily with glass or concrete (from which it removes the water of crystallization). When sodium crud (Na_2O , NaOH , Na_2CO_3) is formed, its melting point rises. Thus, ingress of air into a sodium piping system can lead to a subsequent blockage.

2.1.3 Point Designs

Point designs of a 100-MW(e) plant** with 420 MW(e)-hr of storage were prepared for the three different fluid systems, and pertinent parameters are given in Table 2.1-3. The calculations of the collector field performance were made with the SUNPOW code, which is described in Section 2.2.2. In essence, the code provides a complete optical/thermal balance over the collector row length for the various hours of the day and utilized for these particular parametric analysis solar insolation data pertinent to Albuquerque, New Mexico. The receiver absorptivity and emissivity were held constant at 0.93 and 0.13, respectively, for all three systems.

^{*}This cost analysis was reviewed later in the program, and 15 cents/lb delivered to the U.S. West Coast was utilized in the plant cost estimates.

^{**}Rating established at 10 a.m. on December 21.

TABLE 2.1-3
POINT DESIGN DATA

	Therminol 88	Draw Salt	Sodium
Inlet/outlet temperature [K (°F)]	561/667 (550/740)	561/839 (550/1050)	561/867 (550/1100)
Peak collector efficiency (%)	54.6	52.1	51.8
Maximum steam temperature [K (°F)]	656 (720)	813 (1003)	840 (1052)
Power conversion efficiency (%) (net)	33.5	39.2	39.8
Collection area [$m^2 \times 10^6$ (ft ² x 10 ⁶)]	0.9065 (9.757)	0.8288 (8.921)	0.8293 (8.926)
Thermal collection rate (MW)	310.2	262.1	260.3
Piping heat loss (MW)	9.80	6.39	8.35
Pumping power at 10 a.m. Dec. 21 (MW)	0.642	0.227	0.382
Piping diameter [mm (in.)] and length [m (ft)]	762/1652 (30/5420)	406/1541 (16/5056)	508/1823 (20/5982)
	508/3024 (20/9920)	254/2804 (10/9200)	356/1999 (14/6560)
	254/19824 (10/65040)	203/12329 (8/40448)	254/13611 (10/44656)
			305/500 (12/1640)
Thermal storage tanks	13	6	3
Diameter x height [m x m (ft x ft)]	14.9 x 12.2 (48.8 x 40)	12.9 x 10.7 (42.2 x 35)	26.3 x 12.2 (86.2 x 40)

The collector inlet temperature was kept as high as practical in order to achieve a high power conversion efficiency. The collector outlet temperature was set by practical operating limits for the fluid. In the case of T-88, the collector outlet temperature of 667 K (740°F) is actually too high since it promotes thermal degradation. However, it was chosen in order to achieve the stipulated minimum steam temperature of 644 K (700°F).

The specific dimensions assumed for the receiver rectangular heat transport duct were 63.5 mm (2.5 in.) wide by 25.4 mm (1.0 in.) deep. In attempting to obtain the desired temperature rise while maintaining modest pressure drops in the heat receiver, limits on the collector row length arose. Draw salt, having the largest volumetric specific heat, had the longest row length.

Power efficiencies for single reheat steam cycles were calculated with the SCAP code for the three different designs. The SCAP code computes component efficiencies and provides complete heat balances. Feedwater temperatures consistent with the outlet fluid temperature and closure of the heat balance were chosen. Storage was based on rock-filled thermoclines for T-88 and draw salt and on a multiple tank system for sodium.

Observations of interest concerning the data in Table 2.1-3 are as follows:

1. There is more variation in the power conversion efficiency than in the solar collector efficiency.
2. The T-88 system has the largest collector field and largest piping. Draw salt has the smallest piping.
3. The lower pumping power and heat loss of the draw salt system relative to the sodium system compensate for the latter's advantage in power conversion efficiency, so the net collector areas are the same.

4. The storage fluid volume is lowest for draw salt.
5. The thermal losses in the piping field are significant (2% to 3%), and steps should therefore be taken to reduce overnight losses.

2.1.4 Costs

Cost estimates for the three point designs were prepared using cost data based on production estimates for the design of FMSCs delivered to Sandia Laboratories. Since that design was intended for demonstration purposes rather than for large-scale production, the costs are not representative of those for the final design under Phase I of this program. The cost data were nevertheless useful in guiding the selection of one fluid over another. The costs for the three point designs are presented in difference form in Table 2.1-4. Observations regarding these costs are as follows:

1. Account 4200 - The turbine plant equipment for the T-88 system, having the lowest efficiency, handles more heat for a given electrical output and therefore is the most costly.
2. Account 4400 - The collector field cost is linearly proportional to the overall system efficiency, which is highest for sodium at 20.5% and lowest for T-88 at 18.3%.
3. Account 4500 - The receiver costs are proportional to overall collector area, except that the draw salt and sodium systems operate at higher fluid outlet temperatures and require more costly heat transport ducts and valves.
4. Account 4600 - The sodium thermal storage system cannot employ a thermocline because of the very high conductivity of this liquid metal. Further, since sodium has the lowest volumetric specific heat of the three fluids considered, its thermal storage has the highest cost of the three point designs even though it has the largest differential temperature.

TABLE 2.1-4
PLANT COST CHANGES^(a) RELATIVE TO DRAW SALT SYSTEM (\$ MILLION)

Account No.	Description	Therminol 88	Draw Salt	Sodium
4100	Site/structures/misc. equipment	+0.1	0	0
4200	Turbine plant equipment	+1.1	0	0
4300	Electric plant equipment	0	0	0
4400	Collector/concentrator equipment	+6.2	0	0
4500	Receiver equipment	+3.6	0	0
4600	Thermal storage equipment	+3.1	0	+8.7
4700	Heat transport equipment (includes steam generator)	+2.2	0	+6.1
4800	Distributables/indirects	<u>+3.3</u>	<u>0</u>	<u>+3.0</u>
	Total	+19.6	0	+17.8

(a) Increases are positive.

5. Account 4700 - The piping is the smallest and least costly for draw salt even though it requires an expensive material for the high-temperature portion.* The T-88 system has the longest and largest-diameter piping and, even though made of carbon steel throughout, its piping has a higher cost than the draw salt system piping.

Of the three systems, draw salt has the lowest cost, with sodium being \$17.8 million more costly and T-88 \$19.6 million more costly, i.e., differences of about 10%.

Annual operation and maintenance (O/M) costs were estimated and are given in Table 2.1-5. There is negligible difference in the three accounts except in the fluid makeup costs, which are taken as 1% per week for T-88 and 2-1/2% per year for draw salt (the same as in Ref. 2.1-4). The large makeup cost for T-88 arises from its estimated thermal degradation rate at 667 K (740°F), which is deduced from the data of Figure 2.1-1 obtained from Sandia Laboratories. This degradation rate, while uncertain, is very significant. If the point design had not been based on a requirement to produce 644 K (700°F) steam, a lower fluid outlet temperature would have been utilized. If the O/M costs are capitalized with a 15% capital use charge, the O/M cost difference between the T-88 and draw salt systems would be about the same as their capital cost difference.

2.1.5 Fluid Choice

Draw salt was selected as the heat transport and storage fluid. The rationale for this choice is discussed in this section. No consideration was given to using alternate fluids for storage since this causes the system to operate less efficiently. This is a consequence of the temperature differences that are needed to drive heat across the intervening heat exchanger.

*Incoloy 800 was considered at this point in the program.

TABLE 2.1-5
COMPARISON OF OPERATION AND MAINTENANCE COSTS (\$ THOUSAND)

Item No. and Description	T-88	Draw Salt	Sodium
0100 Operations/Supervision	693	693	693
0200 Maintenance Materials			
210 Spare Parts			
0211 Electric Power Generating System	320	320	320
0212 Solar Equipment	240	264	264
0213 Heat Transport System	70	84	84
0214 Thermal Storage System			
0215 Heat Transport Fluid Makeup	2304	37	0
220 Material for Repair			
230 Other			
0300 Maintenance Labor	1112	1112	1112
0310 Scheduled Maintenance	--	--	--
0320 Corrective Maintenance	--	--	--
Total	4739	2510	2473

Fig. 2.1-1. Representative thermal degradation rates of various heat transport oils (from Sandia Laboratories)

A decision matrix was set up to quantify factors that affected fluid choice. These factors and the maximum weightings are given in Table 2.1-6. It is significant that cost factors 1, 2, and 3 contribute only about 40% to the total weighting. The balance is determined by judgment. In essence, the collective GA assessment was that the T-88 system suffered so much by comparison in terms of the first three cost factors that it was not able to make up the disadvantage by superior ratings in the other factors. It is noted further that preference for draw salt over sodium is affected if one has a successful working experience with liquid sodium.

The main reasons for choosing draw salt were as follows:

1. It has the lowest cost thermal storage and lowest sensitivity to change in storage requirements (some electric grids may need 5 to 6 hr of storage in order to translate the peak solar power generation to the peak demand). Sodium is particularly sensitive here.
2. Therminol 88 has large cost and degradation rate uncertainties and, unlike T-66, requires a trace heating system.
3. A distributed piping system is vulnerable to being damaged. Because the piping system has more bellows, valves, and seals, it will exhibit more leakage. This leads to a preference for a hot fluid which does not interact with the air or the environment. Hot draw salt is a powerful oxidant, but the absence of carbonaceous material around potential release points can minimize leakage consequences.
4. Draw salt has been selected as the heat transfer fluid for a point focus central receiver design (Ref. 2.1-4). It is a favored fluid for thermal storage applications, so the probability for the technology base to increase is very good. Also, Hitec, which is very similar to draw salt, has been employed by GA for a solar test rig that has been built and operated.

TABLE 2.1-6
FLUID SELECTION DECISION MATRIX

Cost Factor	Total Number of Points	Weighting Points		
		T-88	Draw Salt	Sodium
1. Capital cost	20	10	20	9
2. Operating cost	15	0	15	15
3. Cost uncertainties	17	0	17	17
4. Development requirements	30	24	6	13
5. Design flexibility	8	7	6	3
6. Credibility	15	11	8	6
7. Reliability/availability	17	15	7	4
Total	122	67	79	67

5. Line focus systems tend to be modular and can therefore be expected to have more diverse applications than point focus central receiver types. Thus, the fluid choice should be consistent with these other applications in order to further spread out development costs. Draw salt is considered more suitable for smaller, urban-sited applications.

2.1.6 Salt Corrosion Data

The choice of materials for components in a 100-MW(e) solar power plant which uses draw salt as the heat transport and storage fluid is important because of the large potential impact on plant capital cost and reliability. In this section, a brief review is presented of the factors leading to the materials selection.

A summary of corrosion rates obtained from the literature for carbon steel, 2-1/4 Cr - 1 Mo steel, and Incoloy 800 is given in Table 2.1-7. Several different salts were used as an electrolyte in obtaining these data. The composition of these salts is not always clear owing to differences in nomenclature (see Table 2.1-8). For example, Silverman (Ref. 2.1-11) describes draw salt (or eutectic salt by his definition) as having a composition of 46% NaNO_3 -54% KNO_3 , while Martin Marietta (Ref. 2.1-15) and others describe draw salt as having a composition of 60% NaNO_3 -40% KNO_3 . Nevertheless, based on the data in Table 2.1-7 and on the test work and conclusions of Martin Marietta (Ref. 2.1-15), carbon steel and 2-1/4 Cr - 1 Mo steel appear to be unsuitable for service in draw salt at 756 K (900°F). Even though Bohlmann predicts corrosion rates of 0.180 mm/yr [7.1 mils per year (mpy)] and 0.061 mm/yr (2.4 mpy) for carbon and 2-1/4 Cr - 1 Mo steel, respectively, at 756 K (900°F) (Ref. 2.1-9), Martin Marietta observed flaking and very deep pitting (in carbon steel) under these conditions (Refs. 2.1-15, 2.1-16).

Solar central power systems can be considered in two temperature sections: (1) high temperature - greater than 811 K (1000°F), and (2) low temperature - less than 589 K (600°F) (Ref. 2.1-15). The above discussion indicates

TABLE 2.1-7
CORROSION RATES IN HEAT TRANSFER SALTS

Material Tested	Salt Used for Electrolyte	Length of Test	Corrosion Rate at Temperature (mpy)						Ref.
			750°F	850°F	900°F	1000°F	1058°F	1100°F	
Carbon steel	Hitec	Probably a few hundred hours		3.6		12-24		120-600	2.1-9, 2.1-10
Carbon steel	Eutectic salt	24 hr			36			385	2.1-9
Carbon steel	Eutectic salt	100 hr			15.5			159	2.1-9
Carbon steel	Eutectic salt	500 hr			18.2				2.1-9
Carbon steel	Eutectic salt	1 yr ^(a)			7.1 ^(a)			18.4 ^(a)	2.1-9
Carbon steel	Heat transfer salt	700 hr				39-78			2.1-11, 2.1-12
Carbon steel	Heat transfer salt	600 hr			7.8 ^(b)				2.1-11, 2.1-13
Carbon steel	Heat transfer salt	600 hr			15.6 ^(c)				2.1-11, 2.1-13
Carbon steel	Heat transfer salt	1 yr				No corrosion damage, walls burned through in 20 days ^(d)			
Carbon steel		5 yr			(842°-932°F)				2.1-11, 2.1-14
Carbon steel	Draw salt		Wt gain		1.0 ^(f)				2.1-15
					13.5				
2-1/4 Cr - 1 Mo steel	Eutectic salt	24 hr			36			165	2.1-9
(Esshete CRM.2)	Eutectic salt	100 hr			13.4			52.2	2.1-9
	Eutectic salt	500 hr			9.5			20.7	2.1-9
	Eutectic salt	1 yr			2.4 ^(a)			4.1 ^(a)	2.1-9
Incoloy 800	Draw salt	391 hr						(g)	2.1-15
	Draw salt	3585 hr						(h)	2.1-15

(a) Extrapolated, assumes adherence to parabolic rate law.

(b) Unstressed.

(c) Stressed.

(d) Results of 1-yr plant tests in molten salt baths (assumed to be exposed to air) in which the alkali content was not allowed to go over 0.3%.

(e) Alkali content reached 1.5%.

(f) Pipe exposed to discharge line of salt pump, exposed to temperatures between 842° and 932°F for 80% of the time; velocity = 8.2 ft/sec.

(g) Stable dark oxide layer 0.5 mils thick.

(h) Stable dark oxide layer 0.7 mils thick.

TABLE 2.1-8
RATE CONSTANTS FOR OXIDATION OF IRON ALLOYS

Temperature (K)	Duration of Experiment (hr)	Medium	Material	K_p $\left(\frac{\text{mg}^2}{\text{cm}^4 \cdot \text{s}} \right)$	b $(\text{mg}^2/\text{cm}^4)$
823	4506	Hitec	1018 steel	0.09	5.6
823	4506	Hitec	316 stainless	0.0028	0.43
823	4506	Hitec	310 stainless	0.0011	-0.11
823	8	KNO_3	Fe	2.1	(a)
823	8	NaNO_3	Fe	8.6	(a)
823	8	$(\text{Na/K})\text{NO}_3$	Mild steel	150	(a)
773	100	O_2	Fe-0.1 C	0.48	(a)

(a) Not reported.

that carbon steel is unsuitable for the high-temperature section. Martin Marietta tested carbon steel (A516) at 589 and 672 K (600° and 750°F) and found that the corrosion film was stable and did not flake during the 2300 hr of testing at these temperatures (Ref. 2.1-15). The weight gain dropped to zero after 1000 hr of testing and no corrosion penetration was observed. Therefore, carbon steel appears to be a suitable material for use in the low-temperature section of the system.

Martin Marietta investigated the galvanic compatibility of Incoloy 800 and carbon steel using electrochemical techniques. The laboratory test was performed at 614 K (645°F), and a corrosion rate of 0.0122 mm/yr (0.48 mpy) (assuming iron is oxidized to Fe^{+3}) or 0.0183 mm/yr (0.72 mpy) (assuming iron is oxidized to Fe^{+2}) was indicated. However, the tests were performed under stagnant conditions. Higher rates should be expected under dynamic conditions.

When tested in draw salt for 3585 hr at 894 K (1150°F), Incoloy 800 showed no evidence of corrosion penetration (Ref. 2.1-15). Although Incoloy 800 reportedly experienced a small amount of carbide precipitation, no intergranular corrosion was observed during the test period. No stress corrosion cracking was observed in a stressed Incoloy 800 sample exposed to draw salt for 1400 hr at 894 K (1150°F) (Ref. 2.1-15). Similarly, no corrosion problems were observed in a test simulating the effects of thermal stress which occurs when heat is applied to one side only of a tube. In this test, molten salt at 803 K (985°F) was circulated through an Incoloy 800 tube at 3.47 m/s (11.4 ft/sec). The tube was subjected to 10,000 cycles, lasting 2 min each, of heat having an incident flux of $363 \text{ kJ/m}^2 \cdot \text{s}$ (115,000 Btu/hr-ft²). The outside and inside tube wall temperatures were 867 and 841 K (1100° and 1054°F), respectively. Based on this test work by Martin Marietta, Incoloy 800 appears to be suitable for the high-temperature section of the system.

Since Incoloy 800 did experience a small amount of carbide precipitation in the 894 K (1150°F) test, Incoloy 801 might be preferable. Incoloy 801 is a titanium-stabilized modification of Incoloy 800 and is therefore less

susceptible to sensitization and intergranular corrosion than Incoloy 800. However, Incoloy 800 is an expensive material, about 7 to 10 times more expensive than carbon steel. Its employment as the hot section of the heat transport duct in the receiver and the hot side piping of the heat transport subsystem would have a significant impact on the plant cost. Therefore, the stainless steels, which are less costly, merit consideration if they have adequate corrosion resistance. The corrosion of various iron alloys has been investigated at Sandia Laboratories (Ref. 2.1-17) from which Table 2.1-8 has been reproduced. Since the resistance of the alloys to corrosion appears to depend on the tenacity of an impervious oxide layer, the oxidation weight gains of the alloys inversely reflect on the corrosion resistance of the alloys. Based on these studies, Sandia Laboratories recommends SS-316 as a suitable material for the high-temperature portion of a solar electric power system.

Struthers-Wells reportedly has information that 9 Cr steel is suitable for the high-temperature section, although others do not accept this view. In addition, GA has a section of SS-304 pipe, schedule 40, exposed to flowing Hitec for over 7000 hr at temperatures ranging from 822 to 867 K (1020° to 1100°F). No deleterious effects are evident. It can be concluded that there is not a strong industrial data base for the selection of materials for use with draw salt, particularly in the 839 K (1050°F) temperature range.

Corrosion rates may be sensitive to impurity levels in the draw salt that build up with exposure to a cover gas of air and nitrogen (such as nitrites, oxides, hydroxides, and carbonates) and to trace impurities in the original stock (such as chlorides). The corrosion rates can be expected to be affected by salt velocities, salt temperatures, temperature cycling, galvanic effects (different materials around the loop), and base stress in the material. Crevice corrosion may also be a factor. The question of material selection may be addressed by measurements that provide a suitable data base or possibly by assessing and publishing the full extent of industrial experience with draw salt at high temperatures.

The information presented in this section and in the references supports the employment of carbon steel at draw salt temperatures up to 700 K (800°F) and type 316 stainless steel at temperatures up to 867 K (1100°F). These choices have been utilized throughout the system except in some highly stressed locations where Incoloy 800 was chosen instead.

REFERENCES

- 2.1-1 Huntsinger, J., et al., "Process Heat in Petroleum Refinery Applications, Final Report," ERDA Report GA-A13406, General Atomic Company, February 20, 1976.
- 2.1-2 Russell, J., et al., "Preliminary System Analysis of a Fixed Mirror Solar Power Central Station," Electric Power Research Institute Report EPRI ER-434, June 1977.
- 2.1-3 Janz, G. J., et al., "Physical Properties Data Compilation Relevant to Energy Storage. II. Molten Salts: Data on Single and Multi-Component Salt Systems," Purdue University, Thermophysical Properties Research Center, NSRDS-NBS 61, Part II, April 1979.
- 2.1-4 "Conceptual Design of an Advanced Central Receiver Power System," Martin Marietta Company, DOE Contract EG-77-C-03-1742, January 1979.
- 2.1-5 Journal of Physical and Chemical Reference Data, Vol. 1, No. 3, 1972.
- 2.1-6 "Hitec. Properties Data Brochure," DuPont Chemical Company.
- 2.1-7 McDonald, J., and H. T. Davis, "Thermal Conductivity of Binary Mixtures of Alkali Nitrates," J. Phys. Chem. 74 (4), 725 (1970).
- 2.1-8 Liquid Metals Handbook, 3d ed., U. S. Government Printing Office, Washington, D. C., 1955 (TID-5277).
- 2.1-9 Bohlmann, E. G., "Heat Transfer Salt for High Temperature Steam Generation," USAEC Report ORNL-TM-3777, Oak Ridge National Laboratory, 1973.

- 2.1-10 "DuPont Hitec, Heat Transfer Salt," E. I. DuPont de Nemours and Company, Explosives Department, Chemical Products Sales Division, Wilmington, Delaware 19898.
- 2.1-11 Silverman, M. D. and J. R. Engel, "Survey of Technology for Storage of Thermal Energy in Heat Transfer Salt," ERDA Report ORNL/TM-5682, Oak Ridge National Laboratory, January 1977.
- 2.1-12 Sorokin, Y. I., and K. L. Tseitlin, "Corrosion of Metals in Nitrate-Nitrite Melts at 500°C," Khim. Prom. 41 (1), 64 (1965).
- 2.1-13 Sukhotin, A. M., et al., "Stability of Materials of Construction in Contact with Nitrate-Nitrite Melts," Sov. Chem. Ind. 6, 1 (1974) (translation of Khim. Prom.).
- 2.1-14 Silverman, M. D., and E. C. Fox, "Visit to General Electric Noryl Plastics Plant," Oak Ridge National Laboratory, unpublished data, June 15, 1976.
- 2.1-15 "Conceptual Design of Advanced Central Receiver Power Systems," Martin Marietta Corporation Report, Contract No. EG-77-C-03-1724, September 1978.
- 2.1-16 Madsen, A. C., Oak Ridge National Laboratory, private communication to T. Tracy of Martin Marietta, February 9, 1979.
- 2.1-17 Kramer, C. M., W. H. Smyrl, and W. B. Estil, "Corrosion of Fe Alloys in Hitec at 823K," DOE Report SAND 79-825 6, Sandia Laboratories, February 1979.

2.2 CONCENTRATOR AND RECEIVER SUBSYSTEMS

2.2.1 Concentrator/Receiver Loss Train

A necessary prelude to improving the performance modeling of the concentrator and receiver subsystems was to identify significant items in the energy loss train. This section discusses the loss train and the factors of significance.

Figure 2.2-1 illustrates the optical/thermal performance train of the FMSC. Owing to the fixed aperture position, the first loss encountered is the cosine of the solar angle to the aperture (QCOS). The stepped mirror strip structure results in an edge loss (QEDGE) due to direct or reflected light impinging on the structural material. A third source of direct loss (FGLASS) comes from light striking the edge of the finite thickness back-silvered mirror strips, which is then refracted away from the receiver.

Light is reflected with a specific reflectivity (FREFL) and undergoes a loss from dirt on the mirror (FDIRT). The reflected energy (QMIROR) has subtracted from it an estimated shadowing loss (QSHADO). The remaining reflected energy (QM1) suffers losses due to image spread (finite sun diameter) and aiming tolerance causing some of the light to miss the aperture (QMISS). An additional miss loss occurs from misalignment of the receiver aperture due to control malpositioning and receiver assembly sag and twist (QRPOS). The miss losses (QMISS) are functions of the distance traveled by rays of light from each mirror strip and are calculated using three-dimensional analyses.

The light impinging on the receiver aperture (QRCVIN) undergoes several losses before being absorbed in the heat transport fluid. The first loss (QSECLS) is the absorption in the secondary concentrator of a portion of the light impinging on it. The optical losses (QROPT) are from the direct and multiple reflections in the receiver assembly. Radiative, conductive, and convective heat losses occur through the receiver glass cover and

Fig. 2.2-1. Optical/thermal performance train

through the insulation and back cover. Additional losses arise from the unilluminated ends of the receiver (QEND, calculated from each mirror strip) caused by the east/west solar elevation angle.

Subsequent to the actual collector/receiver system calculation, heat losses (QH_{DR}) through the flexible and rigid piping connecting to the supply and return manifolds are subtracted to obtain the overall collector/receiver performance.

Code improvements required to describe the FMSC performance more accurately involved quantifying the "secondary losses" noted above and ray tracing the individual mirror strip images to the receiver assembly. The improvements allowed assessment of the edge loss of various module designs, the effects of mirror strip aiming tolerance, and the energy distribution in the receiver assembly.

2.2.2 Collector Performance Model (SUNPOW)

Modeling the collector performance requires extensive three-dimensional geometric calculations as a function of the daily and annual solar position.

The SUNPOW code models the performance of the concentrator, receiver, and header system. Code input includes module aperture, arc, tilt angle, tangent facet angle, length and spacing, the secondary concentrator/receiver geometry, and optical properties of the mirror strips and receiver assembly. The code calculates the collector system geometry and then determines the optical/thermal performance for the given insolation data at any input times of the day and year. The performance is calculated for specified inlet and outlet temperatures of the selected fluid (several sets of fluid property data are provided in the code). Several secondary loss factors are accounted for in the code, including mirror dirt, glass edge thickness effects, receiver shadow loss, position error, unilluminated end loss, and secondary concentrator loss.

Figure 2.2-2 illustrates the basic geometry pertinent to determining the system performance which the collector model must provide. The major relations may be seen from this two-dimensional cross section, which depicts a view looking toward the west, although the code accounts for three-dimensional effects. "Shadowing" and "blocking" of portions of mirrors by the vertical facet lips result in a displacement of the reflected image from each mirror center as well as a reduction in image size. The reflected image from each mirror approaches the receiver aperture at a different angle and through a different traversal distance, both in the plane shown and in the east/west direction. The solar image spread and displacement at the receiver due to the aiming tolerance must be calculated for each mirror. In addition, the projection of the receiver aperture for each facet, as well as the fraction of the aperture showing the reflective secondary concentrator and the relative position of the secondary concentrator sides and receiver cover glass, must be calculated for each image projection.

Blocking, shadowing, and mirror strip edge blocking are illustrated in Figures 2.2-3(a) and (b). The "structural block direction" shown in Figure 2.2-3(a) is the major edge loss of the FMSC. The portion of the solar rays labeled "block direct on structure" or "block reflection" is absorbed or reflected in noncollection directions and is thus lost from collection. Also lost from collection are those rays directly striking or being reflected into the edges of the finite thickness mirror strips. The loss is due to the fact that the glass refraction will result in the rays missing the collector. However, the glass edge effects are relatively small; they are shown exaggerated in Figures 2.2-3(a) and (b) for clarity. The "shadow by structure" in Figure 2.2-3(b) is not a loss since this light has actually been reflected by the mirror of the adjacent facet. The glass edge blocking of direct and reflected light in this figure is a loss. As

Fig. 2.2-2. Ray travel directions and distances

(a) STRUCTURAL BLOCK DIRECTION

(b) STRUCTURAL SHADOW DIRECTION

Fig. 2.2-3. Structural block and structural shadow direction

may be seen in Figures 2.2-3(a) and (b), the reflected images are significantly different from projected facet widths and have their centers displaced from the facet center.

The finite size of the solar disk results in a roll-off of intensity starting at one projected solar disk distance from the edge of a finite width mirror projection and decaying to zero at one projected solar disk distance beyond the edge of mirror projection. Since the solar disk projection is an angular function, this "blurring" of energy from the facet projection edges increases with the distance of travel from the facet. Figures 2.2-4 and 2.2-5 present facet images calculated by SUNPOW as they would exist at the receiver aperture. Figure 2.2-4 pertains to an image which has traveled a relatively short distance. The flat area of the image is the infinite mirror solar intensity since the solar image projection is less than one-half the facet image projection. Also shown in the figure are the projections of the receiver assembly in the plane of the image projection. As may be seen, this image is aimed predominantly at the secondary concentrator north wing. Figure 2.2-4 also shows the offset (DELTA) of the projected image from the center of the receiver aperture due to aiming and image offset (in this case ~ 0).

Figure 2.2-5 presents another typical calculated facet image for a narrower image which has traveled a greater distance. The solar image spread is larger than one-half the projected facet image and no flat area is seen. The net image offset and aiming displacement from the receiver aperture center (DELTA) is shown as about 6.35 mm (1/4 in.). This offset DELTA is still relatively small because the coincidence of the offset of the image center from the facet center (due to shadowing and blocking) is partially cancelled by the aiming error of the particular facet chosen for plotting. From the figure it may be seen that this facet is "aimed" essentially at the juncture of the receiver window and secondary concentrator south wing projection.

*DELT = MIRROR STRIP IMAGE DISPLACEMENT FROM CENTER OF APERTURE DUE TO INDIVIDUAL MIRROR STRIP AIMING ERROR (≈ZERO THIS CASE)

Fig. 2.2-4. Image projection from south-most mirror strip of module at receiver

SC NORTH WING → | | ← RECEIVER WINDOW → | | ← SC SOUTH WING → | | ← RECEIVER PROJECTION

*DELTA = MIRROR STRIP IMAGE DISPLACEMENT FROM CENTER OF APERTURE
DUE TO INDIVIDUAL MIRROR STRIP AIMING ERROR

Fig. 2.2-5. Image projection from mirror strip in mid-northern half of module at receiver

The numerous angles, distances of travel, image energy distribution, and location and differing area of impact in the receiver assembly have been illustrated in Figures 2.2-2 through 2.2-5. In order to fully quantify performance and to identify areas for design improvement, the complexity of the optical energy passage necessitated refinements to models that existed at the onset of this program. The SUNPOW code was improved so as to provide ray tracing for each mirror on the module. The improved code:

1. Replaces the edge loss curves for a specific module design developed by Georgia Institute of Technology (Ref. 2.2-1) that were previously utilized. This allows calculation of glass edge effects and mirror image shape, location, and direction.
2. Replaces the optical miss fraction from image spread previously calculated by a trapezoidal approximation to the image at the projected receiver aperture. The replacement consists of calculating the mirror image projection (truncated by edge loss, shadow, and glass edge loss) in a plane perpendicular to a line from the individual mirror strip and the reference cylinder (reference plane) at the nominal receiver location. This includes the effect of mirror strip aiming tolerance.
3. Calculates, as part of the replacement in item 2, the projection of the individual parts of the receiver in the reference plane of the individual mirror strip. This allows calculation of energy impinging on the receiver window and secondary concentrator surfaces and that which misses the aperture. The capability to accept average offsets of receiver position due to control positioning and receiver sag and twist is also included, although the feature is not yet used (a conservative allowance is used).
4. Performs appropriate summation of individual mirror strip image components at the receiver window and secondary concentrator and outside so as to calculate detailed losses directly.

5. Includes provision to calculate the effects of displacement of the receiver aperture and/or the receiver pivot point away from the reference cylinder.

The modifications to the performance codes were made so that the improved modeling could be integrated into the collector/receiver performance code (SUNPOW) and the cost/performance optimization code (SS2), described in Section 3.2.

With the improved descriptions of phenomena controlling system performance, several design improvements and system changes were made which resulted in performance gains and cost reductions. Examples are:

1. Using a shallower (more desirable) secondary concentrator.
2. Placing the receiver pivot point on the reference cylinder.
3. Confirming that adequate system performance can be obtained with presently specified mirror angular tolerances.
4. Quantifying the improvement to be obtained from smaller angular tolerances.

2.2.3 Receiver Performance Model (SUNPOW)

In the early parametric analyses, an existing analytic performance model, illustrated in Figure 2.2-6(a), was used to calculate the receiver energy balance and material temperatures. Recent receiver studies under GA funding have developed a more complete analytic capability, namely a generalized algorithm for an arbitrary number of conducting, convecting, and radiating surfaces. The algorithm has been applied to alternate receiver designs being considered. This included evaluating the receiver design being adopted under this program and assessing the adequacy of early analyses

R_1 = CONDUCTION/CONVECTION FROM FLUID THROUGH "HALF CYLINDER" OF INSULATION
 R_2 = FIN CONDUCTION THROUGH SECONDARY CONCENTRATOR

NOTE: ALL CONDUCTION/CONVECTION AND RADIATIVE CONNECTIONS ARE SHOWN.

Fig. 2.2-6(a). Model initially used in SUNPOW and SS2

Fig. 2.2-6(b). Advanced receiver model now used in SOLCOL and SS2

with the simplified receiver model. The algorithm, illustrated in Figure 2.2-6(b), has been integrated into the SOLCOL receiver performance computer program and into the SS2 cost performance computer program.

Confidence in the advanced algorithm was gained by demonstrating good agreement with the annular tube receiver shown in Figure 2.2-7. A point check of the simplified receiver model with the new algorithm indicated that the accuracy of the simplified model was adequate for conceptual design studies. However, the new algorithm was adopted for the performance assessments performed as a part of Task 5.

Original Receiver Performance Model. The early simplified model was developed for receiver designs that could adequately be described in a one-dimensional heat transfer model. The approximations required are as follows:

1. Side surfaces between the receiver pipe and the glass are refractory surfaces that do not absorb energy, i.e., zero absorptivity and emissivity and unit reflectivity.
2. Radiative heat transfer to the secondary concentrator is negligible.
3. Radiative heat transfer to the back side of the receiver is negligible.
4. Two-dimensional effects are unimportant.

With the possible exception of item 1, all receiver designs considered previously are consistent with these approximations.

Advanced Receiver Performance Model. The generalized algorithm developed under GA funding has the capability to analyze a wider range of receiver designs. Significant features of the algorithm are the simultaneous solution of all energy balance equations and a completely general solution

NOTE: ALL SOLID NODES FROM THE INSULATION INSIDE SURFACE ON DOWN ARE INTERCONNECTED BY THE INFINITE REFLECTION RADIATIVE HEAT TRANSFER SUM. ONLY THE CONDUCTION/CONVECTION RESISTANCES ARE SHOWN FOR CLARITY.

Fig. 2.2-7, Receiver design evaluated in advanced receiver design model confirmation study

for an arbitrarily connected network of reflecting, absorbing, and transparent (or translucent) surfaces. The latter feature is a substantial advance over the earlier model.

It is not a sufficient solution to the optical/thermal balance to know the radiative view factor from one surface to another. The sum of the energy absorbed on a given surface after an infinity of reflections and "transmissions" (i.e., through glass) is the required information. In addition, although most surface reflections can be considered diffuse (due to either the diffuseness of incoming light or surface irregularities), reflections from the secondary concentrator surface have a pronounced directional nature. For these reasons, the advanced receiver energy summation starts from the geometric two-dimensional view factors, corrects for directional reflection, and then, by matrix manipulation, derives a general relation for the infinite summation of energy leaving any one surface and its subsequent absorption on all other surfaces (including itself) for given (constant) optical properties.

Because of the generality of this algorithm, improvements in any section of a model do not require extensive model revision. The region of interest is simply divided into more "finite elements," and only the directly connected heat transfer terms must be altered.

The additional detail of the advanced receiver algorithm model of the reference receiver design [Figure 2.2-6(b)] compared with the earlier model [Figure 2.2-6(a)] is permissible because the advanced algorithm has the capability to evaluate the complex radiative heat transfer. Particular areas of detail are the off-axis sections of the secondary concentrator (i.e., wings), the off-axis sections of the glass, and the preferred direction of radiative heat transfer to the secondary concentrator. The wing detail was required to evaluate the "horizontal" temperature gradient in the secondary concentrator. The "edge" sections of the glass were detailed so as to evaluate the effect of close proximity to the wings. The secondary

concentrator details were specifically developed to upper bound the concentrator surface temperature because the temperature strongly affects the choice of concentrator reflective coating or surface finish. The detail "above" the shield is primarily a convenience to aid in temperature evaluation.

Within the framework of a given configuration, the dimensional geometry of the receiver assembly may be varied and the code will solve for the new view factors associated with the change. Basic configuration changes require modification of a section of the code to enable consistent view factor calculations. The geometric variation capability allows assessment of the impact of changes in size, gaps, surface configuration, thickness, and other aspects of the receiver design.

The code provides for parametrically variable properties by an iterative solution technique. Initial values of the system parameters at a given node along the length of the receiver are estimated from the conditions of previous nodes, variable properties are calculated, and the matrix coefficients are determined. Following the solution for the estimated node condition, the variable properties are updated and the solution is reinitiated. The procedure is repeated until the change in any parameter between successive temperature solutions is less than a predescribed (very small) absolute value.

Printout is available of the data for the conduction/convection and radiation net energy transfer array as well as the temperature distributions. These data are used to closely examine the heat transfer and energy loss contributions of the individual parts of the receiver assembly. Desirable geometric features have been identified via these examinations in addition to identification of the critical characteristics and properties within the assembly.

2.2.4 Concentrator Subsystem Cost Model

The cost model for the primary concentrator subsystem used in the design code SS2 is described briefly in this section. The performance model for the concentrator subsystem in SS2 employed the SUNPOW code directly, albeit stripped of input/output routines.

The cost model was generated by a fit to the parametric cost data presented in Figure 2.2-8. These early cost data were generated by concrete quantity take-offs from the different-size panels and their correspondingly different foundations. A simple stress relationship was used to couple panel thickness to panel length and width. Concrete pouring and placement costs were estimated as well as form shipping costs. Foundation excavation costs were estimated and panel handling and installation costs were determined. The labor cost was based on the panel fabrication method outlined in Section 4.2.3. The costs of the steel support posts were estimated along with the associated installation and assembly labor.

Steel master and concrete secondary form costs were determined and amortized over the quantity of concentrator panels needed for the plant. The form costs included a modest dependence on the number of mirror facets. The cost data of Figure 2.2-8 do not include the cost of mirror strips or their installation. Mirror costs were determined using a unit (weight) cost for the glass, a unit (area) cost for silvering, a unit (length) cost for cutting, a cost for handling, and a breakage factor. Thinner glass is handled only in shorter sections and has higher breakage costs. The intent of the mirror glass model was to conform to the industrial observation that 3.18-mm (125-mil) thick glass has about the lowest cost per unit area. The placement of the mirrors on the panel facets was assumed to be semiautomated with a fixed application rate of one mirror strip per minute irrespective of mirror length, width, or thickness. The man-hours to place each unit area of mirror were then duly calculated and costed.

Fig. 2.2-8. Concentrator panel cost trends

2.2.5 Receiver Cost Model

The cost model of the receiver subsystem prepared for the SS2 code was based on an early version of the receiver design described in Section 4.3. The cost model utilizes the same independent geometrical and operating parameters as the performance model and is split into the elements described below, which were largely normalized to early cost estimates of the receiver.

Receiver. The heat absorber duct cost is the sum of the costs of the carbon steel and SS-316 portions. The respective lengths of these portions are calculated assuming carbon steel below 700 K (800°F) and SS-316 above that value. The duct weight is calculated by means of a constant wall thickness of 1.07 mm (0.042 in.) set primarily by corrosion allowances plus a manufacturing margin. The unit costs employed allow for both materials and labor.

The weight of the nickel-clad copper absorber strip is calculated by a geometrical relationship that relates the variable duct diameter to the variable receiver inlet aperture width. The unit cost allows for both materials and labor and for plating with a selectively absorptive coating. The cost of the rollers and internal supports is taken as a fixed cost per unit receiver length. The attachment of the absorber strip to the duct is costed as a fixed cost per unit length of receiver.

The receiver window width is taken as 12.7 mm (0.50 in.) wider than the receiver inlet aperture with a constant thickness of 2.54 mm (0.10 in.). The unit glass cost employed includes materials and installation labor. The window clip cost is taken as a fixed cost per unit length of receiver.

The weight of the steel sheet metal shell of the heat receiver is determined from the height and width of the shell, which are in turn related to the variable duct diameter and receiver aperture. A constant sheet metal wall thickness of 1.02 mm (0.040 in.) is assumed. The unit steel cost employed includes materials and fabrication labor.

The cost of adding a reflective coating (Kinglux) is determined by multiplying the secondary concentrator wing area by an appropriate unit cost that encompasses the cost of that reflective material and its installation. The wing area depends on the variable dimensions of the receiver aperture and secondary concentrator depth.

The dimensions of the calcium silicate insulation inside the receiver shell are calculated based on a minimum insulation thickness set as an independent variable. The dimensions allow for a 9.53-mm (0.375-in.) thick gap between the duct and its insulation canning, a gap selected to minimize convective heat transfer.

The insulation weight is then calculated and its cost determined with a unit cost allowing for both materials and labor. The width of the stainless steel sheet that cans the insulation is determined, then its weight and cost.

The costs of fasteners and general assembly labor are taken as a fixed percentage of the sum of all preceding receiver assembly costs. The resulting total cost of the receiver shows only a weak dependence on duct pipe diameter, receiver aperture, secondary concentrator depth, and inlet and outlet fluid temperatures, but shows a strong dependence on the insulation thickness, which generally controls the dimensions of the receiver shell.

Support and Activating Components. The cost of each support post is taken as a constant times the focal circle radius. This assumed proportionality reflects the fact that a bigger focal circle requires a heavier and longer receiver arm with an increased load on the support posts. The cost of each receiver arm is taken as a fixed cost plus a further cost proportional to the receiver weight multiplied by the arm length. The cost of each jack is treated in a manner similar to the receiver arm cost. Unit pivot and bearing costs are taken as fixed. A noteworthy feature in the cost model is that in each row of N modules, there are $N+1$ sets of support and activating components.

Drive System. The drive shaft cost per module is taken as a fixed cost plus a further cost proportional to the square of the module length. The latter accounts for the strengthening of the shaft to avoid sag as the module length increases.

The drive shaft couplings have a fixed unit cost, but there are only $N-1$ of them in an N module long row. The drive shaft bearings have a fixed unit cost and have $N+1$ of them in an N module row.

The drive motor cost per row is taken as a fixed cost plus a cost proportional to row length, the latter proportionality reflecting the increasing load on the motor. The drive control instrumentation is taken to be a fixed cost per collector row.

There is no strong cost dependence in the drive system on values of the independent variables describing the collector and receiver.

Valves and Their Control. The unit cost per row of control and isolation valves is taken to be a constant plus a cost proportional to the flow rate per row. The control electronics is taken as a fixed cost per collector row.

Receiver Instrumentation. The number of thermocouples per row is assumed to be five plus $1/100$ of the row length in feet. The receiver instrumentation cost per row is taken to be equal to a fixed cost times the number of thermocouples, plus a further fixed cost representing the balance of the receiver instrumentation.

Downcomers. The cost of the downcomers which connect the receiver heat duct to the heat transport subsystem is taken as a fixed cost plus a cost proportional to the flow per row plus a further cost proportional to the pressure level in the inlet downcomer.

Transportation and Erection. The receiver shipping installation and erection cost per unit is taken as a fixed cost plus a cost proportional to the shipping distance [161 km (100 miles) assumed] plus a cost proportional to the receiver weight.

Summary. The receiver subsystem cost depends mostly on the number of modules. However, the subsystem cost declines as the receiver length is increased and as the collector rows are made longer.

REFERENCE

- 2.2-1. Williams, J. R., and J. F. Rollins, "Experimental Solar Heat Supply System With Fixed Mirror Concentrators," Progress Report No. 1, NSF Grant GI-43936, Georgia Institute of Technology, September 1974.

2.3 HEAT TRANSPORT SUBSYSTEM

The Heat Transport Subsystem (HTS) conveys heat from the Receiver Subsystem (RS) to the Electric Power Generating Subsystem (EPGS) and/or the Energy Storage Subsystem (ESS) while solar energy is being collected by the Collector Subsystem (CS). When the CS is shutdown, the HTS conveys stored heat from the ESS to the EPGS.

The HTS accomplishes the energy transport indicated above by means of a pumped loop of draw salt (sodium nitrate-potassium nitrate eutectic) operating between temperatures of 839K (1050F) and approximately 561K (550F). The upper temperature of 839K (1050F) is selected based on state-of-the-art of presently functioning salt loops. The final selection of the lower temperature limit is to be based on the impact of that selection on overall power plant cost as revealed by the Task 2 Parametric Analysis.

2.3.1 HTS Design Criteria

The HTS must satisfy the following functional criteria:

- Convey thermal energy from the CS/RS to the EPGS and ESS
- Convey thermal energy from the ESS to the EPGS

- Provide for heat tracing of piping to prevent freezing of the salt
- Provide for drainage of salt from the HTS
- Provide adequate venting of the piping for refilling the HTS with salt
- Provide steam for the EPGs.

It follows that the HTS consists of the following elements:

- Piping and pumping to convey molten salt from the steam generator and warm storage to the CS/RS
- Piping and pumping to convey molten salt from the CS/RS to hot storage and to the steam generator, and from hot storage to the steam generator
- Heat tracing
- Drain sumps and pumps
- Venting and cover gas provisions for HTS piping and ESS tankage
- A molten salt steam generator.

2.3.2 Approach

The primary objective of the Task 2 Parametric Analysis is to create performance/cost models for the plant subsystems which will provide the basis for selection of the most economic operating conditions for the Fixed Mirror Solar Concentrator (FMSC) Line Focus Central Power Plant.

This is done by defining the performance and cost of all subsystems for a reference set of plant operating conditions, and

by then constructing performance/cost models which measure the effect of variations from the reference operating conditions.

The reference operating conditions selected for the Task 2 Parametric Analysis were:

- Feedwater temperature, TFW 506K (450F)
- Turbine throttle pressure, PT 16547 kPa
(2400 psig)
- Superheat/reheat temperature TSH/TRH 783K (950F)
- Pinch point temperature difference ΔT_P 11C (20F)
- Condenser pressure PC 3-inch Hg
- Hot salt temperature TH 839K (1050F)
- Warm salt temperature TW 572K (569F)
- Solar multiple 1.7
- Heat receiver run length 274 m (900 ft)

2.3.3 Materials

Draw Salt Properties. The parametric analysis of the HTS was based on the draw salt properties presented in Table 2.1-2.

Containment Materials. Investigations conducted by Oak Ridge National Laboratory (Reference 2.5-1, 2.5-2) and by Martin Marietta Corporation (Reference 2.5-3) indicate that carbon steel is a satisfactory containment material for draw salt at temperatures below 644K (700F). Incoloy 800 (Reference 2-3) is recommended for long life service at 839K (1050F).

These materials were selected for the reference HTS of the Task 2 Parametric Analysis. Further consideration was given to alternatives to the Incoloy 800 selection (e.g., stainless steel, 9 Cr 1-Mo) in the Task 4 Conceptual Design and is reported in section 4.4.

2.3.4 Piping Arrangements

In order to minimize the cost of the HTS, the piping arrangement must tend to minimize the aggregate effects of the following cost factors:

- Total piping and manifold run lengths
- The fraction of piping which contains the hot 839K (1050F) salt and requires the use of expensive high alloy pipe
- Piping heat loss
- Pumping required to overcome pipe friction
- The pumping component which must be throttled to maintain equal flow through a large number of parallel paths.

Taking into account these factors, the following piping arrangements were considered.

The Butterfly Configuration. The butterfly patterned receiver-manifold arrangement of Figure 2.3-1 was the selected receiver-manifold arrangement for a small solar powered irrigation pumping facility (Reference 2.5-4) in Arizona. The supply and return manifolds extend along the north-south mid-plane of the collector

Figure 2.3-1 BUTTERFLY-PATTERN RECEIVER-MANIFOLD ARRANGEMENT

field. Receiver flow departs in both directions from the supply manifold along one row of line focus receivers. It then doubles back along the adjacent receiver row and discharges into the return manifold.

This arrangement tends to minimize the aggregate costs due to pipe run length, heat loss, and pipe friction. Since flow does not approach the supply and return manifolds from opposite directions, the arrangement tends to pass greater flow through those receiver paths closest to the EPGs. It therefore does not tend to minimize the pressure throttling required to balance the flow through parallel paths. Nevertheless, it is the economic choice for small distributed collector fields with relatively short manifolds.

Increase in size of a collector field requires increase in receiver run length, manifold lengths or both. The maximum permissible length of the receiver runs tend to be limited by a variety of factors; such as pressure drop, elevation change required for receiver gravity drainage, and thermal expansion of the receiver run. Once the maximum practical receiver run length is adopted, further increase in field size requires lengthening of the manifolds. This increases manifold pressure loss and the throttling pressure required for flow balancing until HTS economics require the adoption of multiple tiers of receiver rows as shown in Figure 2.3-2.

Figure 2.3-2 APPROXIMATE SIZE OF 100 MWe HTS

This arrangement, illustrative of what is required for the 100 MWe power plant of this study, is based on a maximum permissible receiver run length of 274 m (900 ft), adopted for the Task 2 Parametric Analysis. The 274 m (900 ft) receiver run length limit was selected by GA based on the considerations mentioned above and their analysis of the effects of run length on receiver start up operations.

One of the considerations was the effect of varying the receiver run length on piping weight, shown in Figures 2.3-3. While the 274 m (900 ft) receiver run limitations is not an economic optimum from this consideration alone the figure shows that it offers a considerable savings in pipe weight over that resulting from shorter receiver run lengths.

Straight Run Configuration. Given the multi-tier receiver-manifold arrangement dictated by the 100 MWe power plant size and the 274 m (900 ft) receiver run length limitations, the relative merits of the butterfly and straight-run receiver manifold configurations were investigated. These configurations are illustrated in Figure 2.3-4. The comparison revealed that the total weight of piping required for the two configurations is identical. However, only 44 percent of the straight run configuration piping contains hot 839K (1050F) salt; compared to 50 percent for the butterfly configuration. It follows that the straight run configuration requires less of the expensive alloy piping and is therefore the economic choice.

Figure 2.3-3 EFFECT OF RECEIVER RUN LENGTH

Figure 2.3-4

ALTERNATE RECEIVER RUN CONFIGURATIONS

Multiple Trunk Lines. The merits of multiple east-west trunk lines were investigated. Comparison of the two configurations of Figure 2.3-5 showed that the weight of piping required for the double east-west trunk configuration exceeds that required for the single trunk configuration by over 20 percent. The single east-west trunk configuration was therefore selected for the Task 2 reference arrangement.

Reference HTS Piping Arrangement. The reference HTS piping arrangement of Figure 2.3-6 was selected based on the considerations discussed above. The discussion which follows will identify optimum insulation thicknesses and line sizes for a reference HTS based on the operating conditions set forth in section 2.3.2 and a gross EPGS generator output of 105 MWe. The HTS cost model which was subsequently developed provides the basis for estimating HTS costs for a variety of EPGS operating conditions; and, along with other subsystem models, serves as a vehicle for selecting the most economic operating conditions for the plant.

Subsequent to selection of the reference HTS, and concurrent with development of the HTS cost model, further evolution of the HTS concept was considered in an effort to minimize throttling of pump generated pressure for balancing of flow through parallel flow paths. This subsequent modification of the HTS is discussed in section 2.3.13.

Figure 2.3-5 SINGLE VS. MULTIPLE TRUNK SUPPLY LINES

Figure 2.3-6 REFERENCE HTS CONFIGURATION

2.3.5 Insulation

All HTS piping must be insulated to control heat loss. The optimum insulation thickness is defined as that which minimizes the combined cost of the insulation, installed with protective sheathing, and the additional solar collector, receiver and heat transport hardware required to supply the heat which is lost through the insulated pipe walls. The thermal conductivity of .00546 W/m-C (0.034 Btu/hr-ft-F) used for the warm side 561K (550F) piping and .00723 W/m-C (.045 Btu/hr-ft-F) used for the hot side 839K (1050F) piping, was based on the use of calcium silicate insulation. The cost of CS, RS and HTS hardware required to supply solar heat lost through the insulation was assumed to be \$300/kWt. Based on this analysis, an insulation thickness of 64 mm (2-1/2 in) was selected for all 561K (550K) piping, and 114 mm (4-1/2 in) was selected for all 839K (1050F) piping.

2.3.6 Pipe Size

Optimum pipe size was determined as a function of volume flow rate by selecting diameters which minimize the aggregate cost of pipe, insulation; and the additional CS, RS and HTS equipment required to replace the piping heat loss and to generate the electricity to provide pumping necessary to sustain the assumed flow rate. All costs were calculated on a per foot basis and the minimum aggregate cost per foot of pipe determined the optimum

piping diameter for each assumed flow rate. The results of this procedure, discussed in greater detail in section 4.4.2.1, were used to determine the pipe sizes for the reference HTS. The relationships were based on the use of Incoloy 800 for the hot piping and carbon steel for the warm piping.

2.3.7 Trace Heating

Trace heating of all HTS piping is required to prevent freezing of the draw salt in the pipes and to facilitate re-melting of the salt should freezing occur. Freezing occurs if the pipe temperature dips below 494K (429F). The reference HTS is based on the use of electrical trace heating sized to provide heating equal to that lost through the insulation from 561K (550F) molten salt.

2.3.8 Drain Provisions

Provisions are necessary for draining the HTS for occasional maintenance. In addition, the system will be drained each night when the HTS is shutdown.

The heat receivers must be drained, when the HTS is shutdown, in order to prevent freezing of salt in the exposed receiver tubes. There is no threat of overnight freezing in the 839K (1050F) salt lines, and overnight-freezing in the 561K (550F) salt lines could be avoided, with some expense and difficulty, by use of thicker

insulation and selective operation of the trace heating system. But since approximately half of the field salt inventory must be drained from the heat receivers, there is little reason to avoid draining the rest of the field.

The reference HTS is drained by gravity to a sump at the center of the field. Sump pumps transfer inventory to the warm salt ESS tank for overnight storage. Calculated drainage times are 45 minutes for the heat receivers and an additional 90 minutes for the field piping.

2.3.9 Pumps

The Reference HTS pumping system is illustrated in Figure 2.3-7. Three 50 percent capacity field pumps (warm salt), three 50 percent capacity steam generator pumps (hot salt), and four 25 percent capacity sump pumps (not shown in the figure) are required. The flows and heads required for the Reference HTS are shown in Table 2.3-1.

Figure 2.3-7 HTS PUMPING SYSTEM

TABLE 2.3-1
HTS PUMP REQUIREMENTS

	Flow per Pump m ³ /s (qpm)	Head m (ft)	Stages
Field Pumps (3 required)	.2681 (4250)	143 (470)	5
Steam Generator Pumps (3 required)	.1830 (2900)	57.9 (190)	2
Sump Pumps (4 required)	.0631 (1000)	24.4 (80)	1

Since the collector field flow varies with the solar insolation over a considerable range, the collector field pumps are equipped with variable speed electric motor drives. The remaining reference HTS pumps operate at fixed speed.

All of the reference HTS pumps are vertical centrifugal designs, fabricated of 316 SS and use carbide bearings and Colmonoy sleeves. Bearings operate in the molten salt. Shaft seals located above the liquid level on the vertical shaft, control the leakage of nitrogen from the molten salt cavity above the uppermost bearing.

2.3.10 Steam Generator

A conceptual design for the reference HTS steam generator was completed. The design operating conditions for this steam

generator and for the reference HTS are as follows:

Feedwater temperature TFW	506K (450F)
Turbine throttle pressure PT	16547 kPa (2400 psig)
Superheat/preheat temperature TSH/TRH	783K (950F)
Pinch point temperature difference ΔTP	11C (20F)
Superheater/reheater approach temperature	56C (100F)
Salt inlet temperature	839K (1050F)
Salt exit temperature	572K (569F)

The corresponding salt and steam temperature profiles in the steam generator are shown in Figure 2.3-8. Entering salt flow passes through the superheater and reheater in parallel, as is shown in Figure 2.3-9. Separate heat exchangers are used for the preheat, boiling, superheat and reheat sections. The boiling and preheat sections each use dual heat exchangers operating in parallel. The dual boilers share a single steam drum.

All of the steam generator heat exchangers are horizontally oriented, U-tube designs.

The preheater shell cavity is divided by a longitudinal baffle between the straight runs of the U-tube bundle to maintain counter current flow in the shell. Water flows on the shellside of this exchanger, with salt in the tubes, because the arrangement best accommodates the relative magnitudes of the two flows. The shell and the tubes are carbon steel.

Figure 2.3-8 REFERENCE STEAM GENERATOR TEMPERATURE PROFILES

Figure 2.3-9 STEAM GENERATOR CONCEPTUAL DESIGN

The boiler consists of dual horizontally oriented U-tube heat exchangers located below a single horizontally oriented steam drum. The water is on the shell side of the heat exchangers. Hot water and steam are conveyed to the drum through multiple risers by natural convection. Cool water returns to the boilers through multiple downcomers. The shells are carbon steel and the tubes and salt headers are stainless steel.

The superheater and reheater are of the U-tube heat exchanger type, with longitudinal baffles to create counter current flow. The HTS salt flows in parallel through the superheater and reheater. The salt is on the shell side and the steam is on the tube side for each of these heat exchangers. Both tubes and shell are Incoloy 800. The steam bonnets are stainless steel.

Reference HTS steam generator design data are given in Table 2.3-2.

2.3.11 Reference HTS

All of the considerations discussed above contributed to the definition of the reference HTS design that is to provide the basis for the HTS cost/performance model of the Parametric Analysis. The cost breakdown for the reference HTS is presented in Table 2.3-3. The HTS cost model, discussed below, uses the costs of Table 2.3-3 as a basis for calculation of HTS costs for alternate EPGS operating conditions.

TABLE 2.3-2

REFERENCE HTS STEAM GENERATOR DESIGN

Section	Duty MWt (Btu/hr)	U W/m ² (Btu/hr-ft ²)	Area m ² (ft ²)	Materials	
				Tubes	Shells
Preheater	67.15 (229.2 x 10 ⁶)	807 (256)	1505 16200	CS	CS
Boiler	67.77 (231.3 x 10 ⁶)	949 (301)	1078 (11600)	347SS	CS
Superheater	73.66 (251.4 x 10 ⁶)	631 (200)	948 (10200)	Incoloy 800	Incoloy 800
Reheater	43.36 (148.0 x 10 ⁶)	432 (137)	567 (6100)	Incoloy 800	Incoloy 800

TABLE 2.3-3

REFERENCE HTS COST (105 MWe GROSS EPGS OUTPUT)

Item	Cost (\$)
Field Piping and Valves	5,762,000
Thermal Insulation	1,821,000
Pumps and Motors	1,379,000
Salt Treatment and Makeup Equipment	62,000 Tank & Sump Only
Steam Generator	2,629,000
Salt	231,000
Design and Engineering	2,585,000
Heat Tracing	2,191,000
Purge and Vent Equipment	97,000
Pump and Field Electric Service	369,000
Heat Transport Equipment	17,126,000

2.3.12 HTS Cost Model

The total cost of the HTS was estimated by using the following relationship:

$$\begin{aligned} \$HTS = & \left(\frac{100 + KWL}{105} \right) \times \\ & [\$ TRUNKS + \$ MANIFOLDS + \$ PUMPS + \$ SG + \\ & \$ MSC + \$ DE] \end{aligned}$$

where KWL is plant auxiliary power requirements, MWe
SG is steam generator
MSC is miscellaneous and
DE is design and engineering

All EPGS heat balances for the Parametric Analysis are based on 105 MWe gross output from the alternator. The multiplier $(100 + KWL)/105$ adjusts HTS cost to correspond to 100 KWe net plant output.

Piping costs are separated into trunk line related costs and manifold related costs. Since the east-west supply trunks parallel the four 274 m (900 ft) tiers of receivers, their length is assumed to remain fixed. Only the pipe diameter varies as necessary to accommodate flow rates corresponding to different EPGS operating conditions.

The cost of the trunk lines is:

$$\$TRUNKS = \$0.634 \left(\frac{WS}{WSD} \right)^{.55} + \$0.718 \left(\frac{WS}{WSD} \right)^{1.05}$$

in millions of dollars, where WS is the salt flow rate and WSD is the salt flow rate for the reference HTS operating conditions. The first term represents the warm trunks; the second term represents the hot trunks.

The manifolds vary not only in diameter, as required to accommodate alternate EPGS operating conditions, but also in length, as necessary to accommodate the changes in collector field area due to altering of the power plant heat rate. The cost of the manifolds is:

$$\text{\$MANIFOLDS} = \$2.608 \left(\frac{\text{HR}}{\text{HRD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right)^{.325} + 5.385 \left(\frac{\text{HR}}{\text{HRD}} \right) \left(\frac{\text{WS}}{\text{WSD}} \right)^{.725}$$

in millions of dollars, where HR is the EPGS gross heat rate and HRD is the gross heat rate for the reference EPGS operating conditions. The two terms represent warm and hot manifolds, respectively.

The pump costs are a function of salt flow rate:

$$\text{\$ PUMPS} = \$1.524 \left(\frac{\text{WS}}{\text{WSD}} \right)^{0.3}$$

in millions of dollars.

The cost of a number of miscellaneous items (including purge facilities, drain sump, makeup tank and miscellaneous piping

insulation and heat tracing in the vicinity of the turbine building) is assumed to be constant:

\$MISC = \$1.043, millions of dollars.

The steam generator heat exchanger costs are assumed to be functions of required heat transfer area. Required heat transfer area is proportional to heat exchanger duty divided by log mean temperature difference. Thus a steam generator cost model of the following form was obtained.

$$\begin{aligned} \$SG = \left(\frac{PT}{2400} \right)^{.65} & \left[.239797 \left(\frac{Q}{\Delta T_m} \right)^{.65}_{PH} + .551774 \left(\frac{Q}{\Delta T_m} \right)^{.65}_B \right. \\ & \left. + .292225 \left(\frac{Q}{\Delta T_m} \right)^{.65}_{SH} + .376944 \left(\frac{Q}{\Delta T_m} \right)^{.65}_{RH} \right] \end{aligned}$$

in millions of dollars, where Q is heat exchanger duty in millions of Btu/hr, ΔT_m is LMTD in degrees F, and the four bracketed terms refer to the preheater, boiler, superheater and reheater, respectively. The multiplier containing PT, the turbine throttle pressure, accounts for the pressure dependence of the steam generator cost.

Allowance for design and engineering was taken as a fixed fraction of the remaining costs:

$$\$DE = 0.178 (\$STRUNK + \$MANIFOLD + \$PUMPS + \$MISC + \$SG).$$

2.3.13 Selected HTS Arrangement

The reference HTS configuration (Figure 2.3-6) selected for use in the parametric analysis, was chosen for its merits regarding four of the five piping arrangement criteria stated at the beginning of section 2.3.4. The selected configuration rated high regarding the following criteria:

- Reduction of total piping length
- Reduction of length of hot piping requiring high alloy materials
- Reduction of piping heat loss, and
- Reduction of piping pressure loss due to friction.

The fifth criterion, reduction of pressure throttling required to balance flow among parallel paths, had not been addressed when the selection of the Task 2 reference HTS had to be made. So the selection of the reference HTS was made, based on consideration of the first four criteria, while consideration of alternate configurations, chosen to reduce the need for pressure throttling, continued concurrently. The alternate configuration finally selected for economic evaluation and comparison with the reference HTS, is shown in Figure 2.3-10. This HTS arrangement differs from the reference HTS arrangement, shown in Figure 2.3-6, in the following respects:

Figure 2.3-10 FINAL HTS CONFIGURATION

Elimination of Throttling for Balancing Flow Among Tiers. By placing the warm salt pumps equidistant between the inner and outer tier warm salt manifolds, the need for throttling pump pressure to equalize flow to inner and outer tiers is avoided. This reduces the required pumping power.

Inversion of Flow Direction in Inner Tiers. Reversal of the flow direction in the inner tier permits sharing of a single enlarged hot manifold for inner and outer tiers, thereby reducing hot manifold cost.

Location of Storage and Pumping in the Field. HTS hot piping consists of a portion which delivers an essentially constant flow of salt to the steam generator, and the remaining field piping which accommodates a salt flow rate which varies with insolation. The flow rate in field piping peaks each mid-day at a value substantially exceeding that required by the steam generator. Therefore, field piping trunk lines are larger in diameter, and hence, more expensive than steam generator (salt) piping.

The location of the energy storage tanks and pumps in the field, as in the final HTS configuration, reduces the extent of the required field piping and increases the extent of the required steam generator salt piping so as to reduce overall piping and pumping costs.

Cost Benefits of Selected HTS Arrangement. The net effect of switching from the reference HTS configuration (Figure 2.3-6) to the final HTS configuration (Figure 2.3-10) was shown to be the reduction of power plant cost by approximately one million dollars.

The adverse factors tending to increase power plant cost,

- larger number of ESS tanks,
- greater number of HTS salt pumps,

are overbalanced by the beneficial factors,

- less high temperature HTS piping,
- less auxiliary power for salt pumping.

The reduction in auxiliary power leads to reduced cost for the CS, RS and EPGS.

REFERENCES

- 2.3-1 Heat Transfer Salt for High Temperature Steam Generation, E.G. Bohlmann, Oak Ridge National Laboratory, ORNL-TM 3777, December 1972.
- 2.3-2 Survey of Technology for Storage of Thermal Energy in Heat Transfer Salt, M.D. Silverman and J.R. Engel, Oak Ridge National Laboratory, ORNL-TM-5682, January 1977.
- 2.3-3 Conceptual Design of Advanced Central Receiver Power System, Phase I, Martin Marietta Corporation, Final Report, DOE Contract, EG-77-C-03-1724, September 1978.
- 2.3-4 150 kWe Solar Powered Deep Well Irrigation Facility Phase I, Acurex Corporation, Final Report, DOE Contract EG-77-C-04-3917, August 1978.

2.4 ENERGY STORAGE SUBSYSTEM

2.4.1 Introduction and Design Criteria

The energy storage subsystem (ESS) is required to provide thermal storage capacity equivalent to 4.2 hr of operation at the nominal plant net electrical output of 100 MW(e). This 420 MW(e)-hr corresponds to 1200 MW(t)-hr of thermal storage capacity including 80 MW(t)-hr for startup and shutdown.

The parametric analysis of the ESS was performed by comparing three alternatives: a molten salt/Taconite thermocline system, an all-salt thermocline system, and an all-salt single hot/single warm tank system. The salt/Taconite thermocline system was sized for assumed bed utilizations ranging from 35% to 100%, and thermocline performance was calculated using a storage efficiency program developed for the Hewlett-Packard HP-9825A programmable calculator and modified specifically for the salt/Taconite system. The all-salt thermocline system was sized with bed utilizations ranging from 60% to 100%.

All systems were sized and costs were estimated for a thermal capacity ranging from 4.2 to 12 hr using a power conversion efficiency (PCE) of 0.392 and a heat recovery efficiency (HRE) of 0.96.

Two values of Taconite specific heat were assumed, 0.837 and 0.963 kJ/kg·K (0.20 and 0.23 Btu/lb-°F). Costs assumed were 33.1 cents/kg (15 cents/lb) for draw salt and 6.61 cents/kg (3 cents/lb) for Taconite, including shipping. Cost algorithms for the hot tank included 305 mm (12 in.) of calcium silicate insulation with aluminum weather protection, trace heating, and foundations. For the warm tank, the insulation thickness was reduced to 152 mm (6 in.) and less expensive material was used for the containment vessel.

In developing costs for the all-salt thermocline, it was assumed that the external calcium silicate insulation was replaced by an internal layer of solid salt which functioned as an insulation and permitted a low-temperature alloy to be used for the primary containment. The following three tank algorithms were obtained (V = tank volume in cubic feet):

Hot tank	$\$ = 101.8 V^{0.8}$,
Warm tank	$\$ = 47.02 V^{0.8}$,
All-salt thermocline tank	$\$ = 31.00 V^{0.8}$.

2.4.2 Salt/Taconite Thermoclines

A method for storage and retrieval of heat energy (referred to here as a salt/Taconite thermocline) consists of a packed bed of Taconite pellets through which the molten draw salt fluid medium is made to flow. Heat energy is stored (sensible heat) in both the solid Taconite pellets and the molten draw salt and is recovered upon demand when colder fluid is passed through the packed bed. The temperature gradient which results in the bed is actually the thermocline.

Table 2.4-1 presents details for determining the volume of thermal storage required in a thermocline system. Table 2.4-2 presents data for thermocline sizing and costs versus bed utilization. Bed utilization is defined as the percent ratio of the energy exchanged in a stabilized bed to the maximum energy content of the bed (when cycled between the maximum and minimum system temperatures). Cost ranges are shown for varying prices of salt and Taconite. The performance of a typical thermocline is shown in Figure 2.4-1. As the allowable end temperature drop is increased, the bed utilization will increase, as shown in Figure 2.4-2. The effluent temperature from a thermocline during "discharge" degrades or drops off with time, as shown in Figure 2.4-3.

TABLE 2.4-1
THERMAL STORAGE SIZING

Power Conversion Efficiency (PCE) = 0.392

Heat Recovery Efficiency (HRE) = 0.96

Utilization (U) = $\frac{\Delta \text{Thermal Energy Exchanged}}{\text{Total Energy in Storage System}}$ (for Salt Storage, U = 1; ϵ = 1)

Temperature Difference: 288° to 566°C = 278°C
(550° to 1050°F = 500°F)

Thermophysical Properties at T Average = 426°C (800°F):

<u>Property</u>	<u>Salt</u>	<u>Taconite</u>
Specific Heat, kJ/kg·K (Btu/lb-°F)	1.56 (0.373)	0.963 (0.23)
Kinematic Viscosity, m ² /s (ft ² /hr)	8.85 x 10 ⁻⁷ (0.0343)	--
Density, kg/m ³ (lb/ft ³)	1810 (113)	3797 (237)
Thermal Conductivity, kJ/s·m·K (Btu/hr-ft-°F)	3.70 x 10 ⁻⁵ (0.23)	--
Absolute Viscosity, kg/m·s (lb/hr-ft)	4.13 x 10 ⁻⁴ (3.87)	--
Porosity of Packed Bed (ϵ)	--	0.40
Taconite Pellet Diameter, mm (in.)	--	12.7 (0.50 in.)

TABLE 2.4-2
THERMOCLINE SIZING AND COSTS
C_p Taconite = 0.963 kJ/kg·K (0.23 Btu/lb·°F); 4.2 hr of storage

	Bed Utilization (%)						
	30	45	50	55	60	65	100
Volume, m ³ (ft ³)	14,504 (512,188)	9,669 (341,459)	8,702 (307,312)	7,911 (279,375)	7,252 (256,093)	6,694 (236,394)	4,351 (153,656)
Diameter, m (ft)	33.3 (109.26)	29.1 (95.45)	28.1 (92.15)	27.2 (89.27)	26.4 (86.72)	25.7 (84.44)	22.3 (73.14)
Height, m (ft)	16.6 (54.6)	14.5 (47.72)	14.0 (46.08)	13.6 (44.64)	13.2 (43.36)	12.9 (42.22)	11.1 (36.57)
Mass of salt, 10 ³ kg (tons)	10,500 (11,575)	7,001 (7,717)	6,300 (6,945)	5,728 (6,314)	5,251 (5,788)	4,847 (5,343)	3,151 (3,473)
Mass of Taconite, 10 ³ kg (tons)	33,032 (36,412)	22,025 (24,278)	19,822 (21,850)	18,020 (19,864)	16,518 (18,208)	15,248 (16,808)	9,911 (10,925)
Containment cost (\$10 ⁶)	3.76	2.72	2.50	2.32	2.16	2.03	1.44
Salt cost (\$ x 10 ⁶)							
44.1c/kg (20c/lb)	4.63	3.09	2.78	2.53	2.32	2.14	1.39
33.1c/kg (15c/lb)	3.47	2.32	2.09	1.90	1.74	1.61	1.04
22.0c/kg (10c/lb)	2.32	1.55	1.39	1.27	1.16	1.07	0.70
Taconite cost (\$ x 10 ⁶)							
8.82c/kg (4c/lb)	2.91	1.94	1.75	1.59	1.46	1.34	0.87
6.61c/kg (3c/lb)	2.18	1.46	1.31	1.19	1.10	1.01	0.65
4.41c/kg (2c/lb)	1.46	0.97	0.88	0.80	0.73	0.67	0.44
Drain tank							
Volume, m ³ (ft ³)	5,801 (204,875)	3,868 (136,583)	3,481 (122,925)	3,164 (111,751)	2,901 (102,437)	2,678 (94,558)	1,740 (61,462)
Cost (\$10 ⁶)	1.81	1.31	1.20	1.11	1.04	0.97	0.69
Total cost (\$10 ⁶)							
High	13.11	9.05	8.23	7.54	6.97	6.48	4.39
Middle	11.23	7.80	7.09	6.51	6.03	5.61	3.82
Low	9.39	6.54	5.96	5.49	5.08	4.74	3.26

Fig. 2.4-1. Stabilized performance of a salt/Taconite thermocline

Fig. 2.4-2: Relation between bed utilization and end temperature drop for a salt/Taconite thermocline

Fig. 2.4-3. Salt/Taconite thermocline bed discharge performance

Table 2.4-3 shows salt/Taconite thermal storage costs for various hours of storage at a constant utilization of 60% and salt/Taconite costs of 33.1 cents/6.61 cents per kilogram (15 cents/3 cents per pound), including shipping. These data are plotted to show cumulative and component costs in Figure 2.4-4.

2.4.3 All-Salt Thermocline

An all-salt thermocline as opposed to a salt/Taconite thermocline is one in which there is no solid material. In this case, the porosity or void fraction = 1. The all-salt thermocline storage system differs from the hot/warm storage tank concept in that only one tank is required. The success of an all-salt thermocline system is dependent upon the degree to which stable thermal stratification can be attained. This requires careful attention to inlet and outlet manifold design to avoid turbulence or mixing of the hot and warm fluids, which would lower the extraction temperature of the fluid medium and result in decreased utilization. It is anticipated that utilization values for the all-salt thermocline are somewhat higher than those for salt/Taconite systems, although experimental work to date has been limited to relatively small-size systems.

Table 2.4-4 presents the results of all-salt thermocline sizing calculations for various hours of storage and utilization. These data were translated into costs and are plotted in Figure 2.4-5. Also shown are costs for the single hot/warm tank thermal storage concept, which will be discussed later.

2.4.4 Multiple Tanks

The thermal storage system utilizing single hot/single warm storage tanks (or multiple hot/multiple warm tanks) was also examined from a cost and effectiveness standpoint and compared with the alternative ESS concepts discussed previously. Cost summations for the all-salt hot/warm storage tank concepts are given in Table 2.4-5 and plotted to show incremental

TABLE 2.4-3
COST COMPONENTS FOR SALT/TACONITE THERMOCLINE STORAGE

Utilization: 60%
Salt Cost: 33.1¢/kg (15¢/lb)
Taconite Cost: 6.61¢/kg (3¢/lb)

	Hours of Storage		
	4.2	8	12
Volume, m ³ (ft ³)	7,252 (256,093)	13,813 (487,796)	20,719 (731,694)
Mass of salt, 10 ³ kg (tons)	5,251 (5,788)	10,001 (11,024)	15,001 (16,536)
Mass of Taconite, 10 ³ kg (tons)	16,518 (18,208)	31,463 (34,682)	47,194 (52,023)
Containment cost (\$10 ⁶)	2.16	3.62	5.00
Salt cost (\$10 ⁶)	1.74	3.31	4.96
Taconite cost (\$10 ⁶)	1.09	2.08	3.12
Drain tank			
Volume, m ³ (ft ³)	2,901 (102,437)	5,525 (195,118)	8,288 (292,678)
Cost (\$10 ⁶)	1.04	1.74	2.40
Total cost (\$10 ⁶)	6.03	10.74	15.49

Fig. 2.4-4. Salt/Taconite thermocline storage system costs

TABLE 2.4-4
ALL-SALT THERMOCLINE SIZING^(a)

Hours of Storage								
4.2			8			12		
V [m ³ (ft ³)]	U (%)	d [m (ft)]	V [m ³ (ft ³)]	U (%)	d [m (ft)]	V [m ³ (ft ³)]	U (%)	d [m (ft)]
8505 (300,365)	60	22.1 (72.6)	16,201 (572,125)	60	27.4 (89.98)	24,301 (858,188)	60	31.4 (103)
7290 (257,456)	70	21.0 (68.9)	13,886 (490,393)	70	26.1 (85.47)	20,830 (735,589)	70	29.8 (97.84)
6379 (225,274)	80	20.1 (65.95)	12,151 (429,093)	80	24.9 (81.75)	18,226 (643,640)	80	28.5 (93.58)
5103 (180,219)	100	18.7 (61.22)	9720 (343,274)	100	23.1 (75.89)	14,581 (514,913)	100	26.5 (86.87)

(a) V = volume, U = utilization, d = diameter.

TABLE 2.4-5
COST SUMMATIONS FOR HOT/WARM TANK SYSTEMS
FOR VARIOUS HOURS OF STORAGE

	Cost (\$10 ⁶)		
	4.2 hr	8 hr	12 hr
Hot tank	1.63	2.74	3.78
Warm tank	2.38	4.00	5.52
Salt	5.44	9.83	14.26

Fig. 2.4-5. Thermal storage costs for all-salt thermocline and all-salt single hot/single cold tanks

costs in Figure 2.4-6. Using a cost algorithm with a tank volume of $V^{0.8}$, the cost of N multiple tanks to provide the same constant total volume varies as $N^{0.2}$. This is shown in Figure 2.4-7 and is probably accurate for small numbers of N since other factors in construction costs would enter for larger numbers of tanks.

Figure 2.4-8 was developed to facilitate calculations of heat loss per unit area for storage tanks and piping runs. Application of these data is presented for a multiple tank system consisting of two hot tanks 18.29 m (60 ft) in diameter by 10.97 m (36 ft) high and two warm tanks of approximately the same dimensions. For 256 K (0°F) ambient temperature, a value of $Q/A = 1.75 \times 10^4 \text{ MW(t)}/\text{m}^2$ is taken and results in about 0.63-MW(t) loss from the ESS tanks. If this loss rate is applied to the hot storage tanks during a standby mode, the temperature decay versus storage time plot shown in Figure 2.4-9 is obtained. The decay rate is well within acceptable values for very large (and unlikely) standby times.

The use of a solid layer of draw salt as internal insulation was considered at one time. This internal layer of draw salt would replace the calcium silicate external insulation and would permit the use of a low-temperature alloy for the primary containment. However, upon reconsideration this approach has been discarded in favor of the more conventional use of external insulation for the following reasons:

1. Some type of internal liner would be required to provide an interface for the insulating salt layer.
2. Since this liner would be in contact with the salt at 841 K (1053°F), it would have to be fabricated using a high-temperature alloy.
3. The salt immediately adjacent to the liner (toward the outside) would be liquid, and unless the liner provided molten salt integrity from the main body of the molten salt, the insulating layer of the salt would drain into the main tank area during operation from storage during normal cyclic operation.

Fig. 2.4-6. Hot/warm tank storage system costs

Fig. 2.4-7. Cost ratio for multiple versus single tanks with same storage volume

Fig. 2.4-8. Heat loss per unit area for storage tanks and piping.

Fig. 2.4-9. Draw salt storage tank temperature decay during standby

4. The liner would have to be designed to take the hydrostatic head of the molten salt in compression.
5. The radial location of the solid/liquid interface in the insulating salt liner would vary during the cyclic operation of the storage system from full to empty to full again.
6. During standby with the hot tank filled, the liquid/solid interface would penetrate into the solid insulating salt at a rate of approximately 6.35 mm/hr (0.25 in./hr) during the first 24 hr. In an 8-hr period, the interface would penetrate about 76.2 mm (3 in.).
7. Any cost advantage in implementing a concept using internal insulation would probably be offset by the material and fabrication costs of the liner assembly, excluding consideration of the design difficulties and uncertainties of this concept.

2.4.5 Reference Design Choice

Figure 2.4-10 compares costs for the three alternative systems considered and indicates that the all-salt single tank thermocline system cannot become competitive unless the utilization is about 75%, while the salt/Taconite system, although always less expensive than the all-salt thermocline system, cannot be considered a viable alternative unless the utilization is greater than 55%. The salt/Taconite thermocline does not offer a sufficient cost incentive over a separate tank system to outweigh its disadvantages: (1) difficulty in starting up with cold Taconite pellets, (2) difficulty in repairing a tank containing hot Taconite pellets, and (3) unknown long-term chemical interaction between Taconite and hot draw salt.

The all-salt thermocline offers marginal cost advantage even at utilizations above 75%. In the event that a separate drain tank is required or if performance degradation due to the inherent temperature drop at the end of discharge is considered, this small advantage is dissipated, and increased costs are incurred for a manifold system to minimize mixing. Therefore, separate hot and warm tanks have been selected for the ESS.

Fig. 2.4-10. Cost summary for alternate thermal storage systems

2.5 ELECTRIC POWER GENERATING SUBSYSTEM

The following is a discussion of the parametric analysis leading to the selection of the electric power generating subsystem (EPGS) configuration and operating conditions which minimize overall cost of the Line Focus Solar Central Power Plant. The parametric analysis involves selection of an EPGS configuration, selection of reference operating conditions, determination of performance and costs for the corresponding reference EPGS, and formulation of a performance/cost model, for evaluation of alternate EPGS operating conditions.

2.5.1 EPGS Design Criteria

The EPGS must fulfill the following requirements:

- Convey main steam from the HTS steam generator to the turbine generator
- Generate sufficient electricity to deliver 100 MWe to the utility grid
- Generate sufficient additional electric output to supply all auxiliary electric loads within the plant
- Reject EPGS waste heat to the environment by means of a condenser and wet cooling tower
- Supply the HTS steam generator with deaerated, regeneratively heated feedwater.

2.5.2 Candidate Power Cycles

Power cycles without reheat and with one and two stages of reheat were considered for use in the FMSC Line Focus Solar Central Power Plant. The selection was made on the basis of equipment availability, performance and resulting power plant cost.

Turbine generators utilizing the double reheat cycle have occasionally been used by the power industry in supercritical cycles 400 MWe and larger. Although double reheat gives such units a performance advantage, they have proven relatively unreliable in practice, resultingⁿ in low plant availability. Moreover, double reheat turbine generators are not available for the desired 100 MWe service.

The availability of turbine generators utilizing single reheat was verified for sizes down to 75 MWe and for turbine throttle pressures of 12,411 and 16,547 kPa (1800 and 2400 psig). Such units give gross cycle efficiencies in the range of 40-43 percent.

Turbine generators without reheat are commonly available with turbine throttle pressures of 8274 to 9653 kPa (1200 to 1400 psig). Such units are considerably less expensive than those utilizing reheat, but gross turbine cycle efficiencies are only in the 36-38 percent range. As a result, the cost of collector and receiver hardware required to drive a turbine

generator without reheat exceeds that required to drive a turbine generator with reheat by a large margin. A margin exceeding entire cost of a single reheat turbine generator.

The above considerations led to the selection of a single reheat power cycle for the reference EPGS.

2.5.3 Reference EPGS

The reference EPGS utilizes a tandem-compound, 2-flow single reheat turbine generator with high, intermediate and low pressure sections. It includes a water cooled condenser, 7 stages of feedwater heating, condensate pumps, boiler feed pumps, mechanical draft wet cooling towers, and circulating water pumps. The general arrangement of the feedwater train and of the turbine extraction flows is illustrated in Figure 2.5-1.

The feedwater train consists of 3 stages of low pressure heaters, a deaerator heater stage, and 3 stages of high pressure heaters. High pressure heater drains cascade to the deaerator and are pumped forward with the feedwater stream. Low pressure heater drains cascade to the condenser.

The operating conditions selected for the reference EPGS are as follows:

Figure 2.5-1 ELECTRIC POWER GENERATING SUBSYSTEM FLOW DIAGRAM

Feedwater temperature	506K (450F)
Turbine throttle pressure	16547 kPa (2400 psig)
Superheat/reheat temperature	783K (950F)
Condenser pressure	3 in.Hg.
Hot salt temperature	839K (1050F)
Pinch point ΔT	11C (20F)

The warm salt temperature associated with the above operating conditions is 572K (569F). The corresponding steam generator temperature profiles are shown in Figure 2.3-8.

The heat rejection system is based on the use of concrete counter flow mechanical draft cooling towers. The towers consist of four 15.5 m (51 ft) square modules, each driven by a 8.5 m (28 ft) diameter fan. The towers are assumed to be located on a down wind edge of the collector field. Circulating water is conveyed to and from the condenser by means of a 1.5 m (5 ft) diameter concrete pipe.

2.5.4 Alternate Operating Conditions

The following EPGS operating condition alternatives were investigated during the parametric analysis:

Feedwater temperature	450K, 478K, 506K, 533K (350F), (400F), (450F), (500F)
Turbine throttle pressure	11032, 13790, 16574 kPa (1600), (2000), (2400 psig)
Superheat/reheat temperature	783K, 811K (950F), (1000F)
Condenser pressure	2.5, 3.0, 3.5 in.Hg.
Hot salt temperature	839K (1050F)
Pinch point ΔT	6C, 11C, 17C (10F), (20F), (30F)

Heat balances were run for all combinations of the first three parameters listed above. Selected heat balances were investigated to evaluate the effect of various condenser pressures and pinch point ΔT values on subsystem costs and performance.

2.5.5 Power Plant Auxiliary Load Model

The total auxiliary power requirements for the power plant, KWL, consists of the summation of power required to drive the following equipment:

- Feedwater pump, KWBFP
- Condensate pump, KWCP
- Cooling tower fans, KWCTF
- Circulating water pumps, KWCWP
- Miscellaneous EPGS equipment, KWMISC
- Heat transport subsystem pumps, KWHTSP
- Heat receiver subsystem motors, KWHRSM

- Heat receiver subsystem control valves, KWHRSC
- Master control subsystem, KWMC.

$$\text{Hence, } KWL = KWBFP + KWCP + KWCTF + KWCWP \\ + KWMISC + KWHTSP + KWHRSM + KWHRSC + KWMC$$

All of the heat balances of the parametric analysis were based on a gross alternator output of 105 MWe, which presumes an auxiliary power requirements of 5 MWe. Modeling of the actual auxiliary load therefore involves adjusting the auxiliary loads, calculated for a plant with a gross electric output of 105 MWe, by a factor of $(100 + KWL)/105$. This is accomplished by applying the following equations:

$$KWBFP = KWBFP \text{ (heat balance)} \left(\frac{100 + KWL}{105} \right)$$

$$KWCP = KWCP \text{ (heat balance)} \left(\frac{100 + KWL}{105} \right)$$

$$KWCTF = .398 \left(\frac{HR - 3412}{HRD - 3412} \right) \left(\frac{100 + KWL}{105} \right)$$

$$KWCWP = .398 \left(\frac{HR - 3412}{HRD - 3412} \right) \left(\frac{100 + KWL}{105} \right)$$

$$KWMISC = .800$$

$$KWHTSP = 1.690 \left(\frac{WS}{WSD} \right) \left(\frac{100 + KWL}{105} \right)$$

$$KWHRSM = .750 \left(\frac{HR}{HRD} \right) \left(\frac{100 + KWL}{105} \right)$$

$$KWHRSC = .400 \left(\frac{HR}{HRD} \right) \left(\frac{100 + KWL}{105} \right)$$

$$KWMC = .500$$

where HR = EPGS gross heat rate
 HRD = gross heat rate for reference EPGS
 WS = salt flow rate
 WSD = salt flow rate for reference HTS
 KWL = plant auxiliary load in megawatts

and where EPGS heat rejection is proportional to
 (HR - 3412).

2.5.6 EPGS Cost Model

The cost of the reference EPGS (105 MWe gross) is presented in
 Table 2.5-1.

TABLE 2.5-1

REFERENCE EPGS COST (105 MWe GROSS OUTPUT)

Item	Cost (\$)
Turbine Generator and Accessories	9,828,000
Heat Rejection System	2,562,000
Condensing System	497,000
Feed Heating System	967,000
Plant Auxiliary Subsystems	970,000
Design and Engineering	<u>2,547,000</u>
Turbine Plant Equipment	17,371,000

The effects of alternate operating conditions on EPGS cost
 are represented by the following parametric equations:

$$\text{Hence: } \$EPGS = (\$TF + \$COND + \$HREJ + \$FW + \$DE) \left(\frac{100 + KWL}{105} \right)$$

2.6 MASTER CONTROL SUBSYSTEM (MSC)

Operation of the LFSCPS will be very similar to conventional fossil-fired power plant operation. Operations and conditions will be prescribed by the plant control operators, and it will be the task of the master control subsystem (MCS) to minimize the burden on the operators in performing these functions. The fact that the "single heater" of a fossil plant is replaced by many small, geometrically dispersed "heaters" presents a unique data assimilation problem as a control/operation requirement.

A number of arrangements and implementations of information, control, and operator interface are possible. The extremes to be considered range from the very conventional (though somewhat old-fashioned) analog control of each subprocess to centralized direct digital control (DDC). In the "conventional" system, the setpoint of each subprocess is operator set based on analog display of subprocess and system level parameters. This type of control/data display is illustrated in Figure 2.6-1(a). The other end of the spectrum is a completely automated central DDC in which the plant operators simply initiate total plant operation or suboperations, monitor the operation via data display, and, if necessary, modify the operation by demand to the central processor. Control/data display of a DDC system is illustrated in Figure 2.6-1(b).

The conventional analog type of control is characterized by a chief operator monitoring system level (plant) parameters and certain main subsystems, adjusting the setpoints of subsystems under his control, and communicating to several roving operators the plant requirements for their subsystem setpoints. Individual roving operators monitor subsystem displays for each controllable element and adjust setpoints for these elements. The very large number of individual control subsystems of a distributed solar collection system, the limited flexibility, and the mass of data which would need to be displayed preclude practical application of this type of conventional control.

(a) CONVENTIONAL PLANT CONTROL

(b) DIRECT DIGITAL CONTROL

Fig. 2.6-1. Conventional plant and direct digital control

Direct digital control may be described as gathering both subsystem and plant data and operating on the combined data to determine the commands to the final control actuators in order to effect regulatory control in a unified form. In addition to top level status, operator demand and off-normal (exception) displays are generated. Various levels of operator setpoint adjustment are available through keyboard and/or status change pushbutton entry. The advantages of DDC lie in reducing demands on the operator from routine tasks and accentuating the operator skills of deductive reasoning, judgment, and short-term proficiency in tactile/visual manipulation. This is accomplished by DDC processing of essentially all information within normal bounds and alarming and displaying of information regarding "exception" conditions for operator action.

Another consideration in the choice of control and data acquisition is the data transmission/data processing. Transmission of the nearly 10,000 measurements within the distributed collector system alone would prohibit hardwired control data processing. Multiplexing would relieve the massive cable pulling and termination requirements; however, large data processing rates would still be required.

A major disadvantage of DDC is the source of a "single point failure." While the reliability of modern computers has increased dramatically, the fact that failure of the central processor in DDC removes all control (unless "drop down" is provided) appears unacceptable for the reliability/availability requirements of a central power station.

An alternate type of control configuration which retains the basic features of DDC, avoids most of the "single point failure" problems of central computer DDC, and actually provides certain additional features is distributed control in a supervisory mode. Here semiautonomous process controllers (PC) units are placed in control of individual portions of the system. The PC units are placed in setpoint or supervisory control by the central unit and/or by direct operator demand. The PC maintains control of all

subelements of the subprocess under its control per the top level demand. Fault conditions within the subprocess are corrected by the PC control where appropriate, or dropout of PC control is made with a transfer of control actuators to central control, or alarm status is sent to the central system operator for decision. In all cases of fault conditions, alarm and exception data are transmitted to the control center. Data processing in the PC allows routine data to be sent to the control center at much less frequent intervals than would be required to provide data for central control and protective action (summary data). The summary data rate is only that required to record routine operation. The exception data transmitted by the PC when out of normal bound conditions are encountered and the associated alarm allow both operator action and the higher-speed data recording needed to evaluate the cause and consequence of malfunction and failures within the system.

Distributed supervisory control is illustrated in Figure 2.6-2. Ignoring the "direct data signals required for drop down control" and the "actuator demand," it may be seen that the configuration looks like the conventional control configuration except that the computer supplies the setpoints. The difference is that the setpoint is for a complete subprocess rather than an individual controller.

Distributed supervisory control retains the flexibility, application of advanced control techniques, improved data processing/display, and reduction of operator burden of DDC while effectively eliminating the single point failure mode. Data processing/logging requirements are more easily met. Advantage is taken of modern microprocessor technology owing to the repetitive character of the LFSCPS collector system in that 128 identical units will provide the control of 972 individual collector rows.

* DROP AT PC INDICATES DROP DOWN TO REMOTE ACTUATOR CONTROL.

** THE DATA LINK CARRIES SET POINT DATA TO THE PC FROM THE CENTRAL UNIT AND SUB-PROCESS DATA TO THE CENTER FROM THE PC. THE LINK IS A MODIFIED DROP LINK COMMUNICATION TIE.

Fig. 2.6-2. Distributed plant control

Initial evaluation of the cost of the microprocessor PC distributed supervisory control system indicates a current cost which is competitive with that of a conventional system. However, the future cost of PC units is expected to decrease dramatically. This is illustrated in Figure 2.6-3. The data in Figure 2.6-3 were based on an extensive survey of PC manufacturers.

REFERENCE

- 2.6-1 Garen, Eric, "Microcomputers: The Ubiquitous Control Element," Control Eng. 25, 53 (June 1978).

REPRODUCED FROM
CONTROL ENGINEERING,
VOLUME 25, JUNE 1978,
PAGE 54.

Fig. 2.6-3. "Typical" computer costs for various microcomputer and minicomputer configurations and projections of future costs

2.7 SELECTION OF POWER PLANT OPERATING CONDITIONS

The most economic power plant operating conditions were selected using the HTS and EPGS cost models of section 2.5 and 2.6 and the ESS and CS/RS cost models presented below.

2.7.1 ESS Cost Model

The Energy Storage Subsystem cost model is based on a reference system tank cost of \$2,851,000 and an associated salt inventory cost of \$3,081,000. Using the 0.8 exponent for tank cost given in section 2.4, the following ESS cost model is obtained:

$$\$ESS = 2,851,000 \left(\frac{\Delta T_D}{\Delta T} \cdot \frac{HR}{HR_D} \right)^{.8} + 3,081,000 \left(\frac{\Delta T_D}{\Delta T} \cdot \frac{HR}{HR_D} \right)$$

where the first term represents tank costs, and the second term represents salt inventory costs. The required volume of salt is directly proportional to EPGS heat rate (HR) and inversely proportional to HTS salt loop temperature difference (ΔT).

2.7.2 CS/RS Cost Model

The Collector and Receiver Subsystem costs are primarily a function of peak collection temperature as follows:

<u>Peak Collection Temperature F</u>	<u>CS/RS \$/Mwt</u>
850	222,200
950	227,400
1050	232,700
1150	238,600

Since the maximum collection temperature of 1050F was selected earlier, based on currently established state-of-the-art, the cost model for the CS/RS is simply,

$$\text{\$ (CS/RS)} = \$232,700/\text{Mwt.}$$

The cost models presented above were used to make the economic comparison of alternate operating conditions shown in Table 2.7-1.

2.7.3 Selected Power Plant Operating Conditions

The power plant economic trade off's reported in Table 2.7.1 result in selection of the following operating conditions as the end result of the Task 2 Parametric Analysis.

Feedwater Temperature	506K (450F)
Turbine Throttle Pressure	16547 kPa (2400 psig)
Pinch Point ΔT	5.6K (10F)
Superheat/Reheat Temperature	811K (1000F)

Condenser Pressure

76 mm (3.0 in.) Hg.

Steam Generator

Inlet Salt Temperature

839K (1050F)

Exit Salt Temperature

561K (550F)

The power plant design described in section 4 is based upon these maximum guaranteed load operating conditions.

TABLE 2.7-1

Economic Impact of Plant Operating Parameters

Operating Parameter	Power Plant Capital Cost Increment, \$
Feedwater Temperature, TFW	
350F	+ 1,063,000
400F	+ 404,000
450F	+ base
500F	85,000
Turbine Throttle Pressure, PT	
2000 psig	+ 512,000
2400 psig	base
Superheat/Reheat Temperature, TSH/TRH	
950F	base
1000F	- 2,046,000
Pinch Point Temperature Difference, ΔTP	
5F	+ 300,000
10F	base
20F	+ 2,000
30F	+ 125,000
Condenser Pressure, PC	
2.5 in.hg.	+ 310,000
3.0 in.Hg.	base
3.5 in.Hg.	+ 195,000

3. SYSTEM DESIGN AND ANALYSIS

3.1 INTRODUCTION

The system design has been accomplished by establishing basic operating parameters for the collector field and then suboptimizing the concentrator and receiver subsystems as a unit and separately suboptimizing the HTS, ESS, and EPGS. The means by which the parameter selection and the system suboptimization were performed is described in the remainder of Section 3. Also included are various data, such as environmental response, operational factors, and utility load data, which influenced the selection of system physical and operating parameters.

The main tool used in the system analysis is a cost performance optimization code called SS2. Originally, a cost performance code called SOLSAL was proposed for use in the system design. However, examination of that code showed it would be unsatisfactory because it incompletely modeled the system and because its cost models were obsolete. Therefore, SS2 was written to perform the economic optimization of the 100-MW(e) LFSCPS.

The basic intent of SS2 was to enable selection of optimum parameters for the system, principally the combination of operating conditions and component sizes and geometries that give the minimum bus bar cost of electricity (BBCOE)--in other words, to choose parameters by optimizing at the system level instead of at the subsystem level. However, it was not possible to achieve the full intent of SS2 directly in a single code because the accurate calculation of the EPGS efficiency is complex and does not lend itself to modeling within the framework of an overall system optimization code. It was therefore proposed to describe this complexity by parametric data

generated from separate computerized calculations of the EPGS performance. This approach was initiated but not carried through, because a simpler model of the EPGS within SS2 showed that the EPGS efficiency so dominated plant economics that the efficiency should have been made as high as practical. The result was that the operating conditions of the molten draw salt which interfaces the EPGS to the rest of the system were chosen based on practical limits. The nominal limits chosen were 839 K (1050°F) inlet temperature to the steam generator and a 561 K (550°F) exit temperature. This selection, which is discussed later, had the practical effect of separating the system optimization into two parts: the EPGS on the one hand and the collector field, heat transport, and energy storage systems on the other hand. Thus, Bechtel National, which had the design responsibility for the EPGS and HTS, was able to perform the EPGS optimization using influence functions, provided by GA, that describe the collector field and was able to calculate accurate efficiency values with existing computer codes encompassing the EPGS modeling complexity. In addition, the optimization of the HTS turned out to be more a study of the economics of different field layouts than the optimization of parameters for a given layout.

The preceding discussion provides the rationale for what SS2 could be and was used for. It became a code which contained relatively simple cost and performance models of the HTS, EPGS, and ESS and a sophisticated cost and performance model for the collector field. The calculated EPGS efficiency was normalized at the design point to the accurate value generated by Bechtel National. Cost data for the EPGS and HTS were supplied by Bechtel. Thus, SS2 was used to select the most cost effective parameters for the receiver and collector, to determine the required size of the collector field, to calculate the total system BBCOE and the yearly capacity factor, and to assess the impact of plant and thermal storage sizes on the total plant capital cost and BBCOE.

While more limited in scope than originally intended, SS2 has been a useful tool in this LFSCPS program. The limitations of the code should be understood to provide insight and judgment regarding the system parametric data that it was employed to generate.

Section 3.2 gives details of the subsystem cost and performance models within SS2. Many of these models and data utilize the parametric data given in Section 2 on the early component design described in Section 4. The integration of these models and the overall logic of the code are also described.

Section 3.3 presents the results of the parametric study and describes the selection of parameters for the concentrator and receiver subsystems. The sensitivity of the plant design to various optical parameters is also explored.

Representative loads placed hourly on the Southwestern Public Service grid are presented in Section 3.4 together with incremental fuel cost data. Generalizations are drawn from these data.

Section 3.5 presents a discussion of operational factors that impact the design including response to wind and hail. [Seismic response (of the receiver) is analyzed in detail in Section 4.3.] The operating and maintenance requirements for the collector field are also discussed.

Section 3.6 addresses the dynamic behavior of the system, including a discussion of control philosophy. The dynamic model is described, the transient data generated are reviewed, and their impact on the design is assessed. These data had a significant role in confirming that a 274-m (900-ft) collector row length would be satisfactory.

3.2 SS2 COST AND PERFORMANCE COMPUTER CODE

3.2.1 Code Logic

Figure 3.2-1 illustrates the overall structure of the SS2 code. Basically, the code calculates the size of the collector field for initial values of the variables specified in Table 3.2-1 in order to meet the design requirements.

Initially these requirements (Appendix I) were interpreted as sizing the collector field to produce 100 MW(e) net without the aid of storage at 10 a.m. on the worst clear day of the year. Such a plant has the smallest capital cost but does not fill the specified storage capacity of 420 MW(e) any day of the year if it is simultaneously producing 100 MW(e). Therefore, the code was modified to size the field so as to just fill up the storage on the equinox day (the day on which the most energy is collected) while simultaneously producing 100 MW(e) net. In sizing the field, thermal losses from the piping and the salt pumping power are included. It is to be noted that pumping the salt adds energy to the salt and increases its total temperature. The pumping losses at the peak heat collection time, 12 p.m. March 21, are calculated in order to determine required pump capacity. The system elements are then costed and the cost of electricity (COE) is determined using the formalism in Appendix I. The cost data utilized in these optimization studies were based on early first plant cost estimates. These estimates were subsequently found to be lower than the final plant cost estimate described in Section 5. However, the initial costs are believed to have provided suitable influence functions.

The BBCOE is the objective function which is then optimized by a packaged algorithm that adjusts the independent variables so as to minimize the COE. The optimization algorithm is a direct search method using a Fibonacci sequence as a means for accelerating the search when favorable trends have been identified. The algorithm converts all independent variables into

Fig. 3.2-1. Overall logic of cost performance optimization code SS2

TABLE 3.2-1
INDEPENDENT VARIABLES IN THE SS2 CODE

FMSC panel width
FMSC panel length
Number of panels per collector row
Collector row pitch
Panel aperture angle
Number of facets
Mirror strip thickness
Tangent facet rotation angle
Warm salt temperature
Hot salt temperature

Secondary concentrator aperture
Secondary concentrator depth
Heat transfer duct diameter
Receiver insulation thickness
Copper absorber plate thickness

Reference warm trunk pipe diameter
Warm feed pipe diameter
Reference hot trunk pipe diameter
Hot return pipe diameter

Warm pipe insulation thickness
Hot pipe insulation thickness

Thermal storage capacity
Plant rating

integers to be varied over a discrete range. Upper and lower bounds of the variables which may arise from practical considerations are used as input to the code.

3.2.2 SS2 Code Subsystem Models

For the SS2 code, subsystem models were prepared to reflect performance and cost. The concentrator and receiver subsystems were modeled as described in Section 2.2. The HTS, ESS, and EPGS models used in the code differed in some respects from the models that resulted from subsystem studies. This is because the code was required before the Task 2 subsystem models were complete. However, there were no substantial differences in the models.

Heat Transport Subsystem Model. A cost and performance model of the heat transport subsystem (HTS) was prepared for use in the SS2 system design code. This model was based on a flow distribution scheme considered early in the field layout studies but not in fact finally adopted. Nevertheless, the model provides an adequate influence function for the purposes for which SS2 was used. These purposes include determining the impact on the COE from large capacity factors and large plant ratings, both of which require larger collector fields and hence larger heat transport systems. The SS2 code was not used, however, to set the specifications of the HTS.

The piping layout of the HTS model is shown in Figure 3.2-2. It has a centralized thermal storage. In essence, the number of collector rows is determined from the collector area required to give the specified output. The collectors are arranged in east-west rows into tiers. Figure 3.2-2 shows 8 such tiers, which is the minimum. As the field grows, the vertical depth of the tiers increases, but then the model adds additional pairs of tiers in both north and south sectors when the aspect ratio of the field Y/X becomes so large that it can be brought closer to unity by the additional tiers. The piping is arranged so that the flow path from each warm manifold inlet to its hot manifold outlet is the same for all interconnecting rows. This arrangement tends to balance pressure drops and minimize

Fig. 3.2-2. Heat transport system model

pumping power. The flow is arranged so that the more expensive hot side piping of type 316 stainless steel is the shortest and nearest to the hot storage tank. The less expensive warm side pipe of carbon steel is longer and furthest from the warm storage tank. There is an axial spacing of 6.1 m (20 ft) between tiers and a vertical spacing of 15.2 m (50 ft) between sectors. The collector-to-collector spacing is the reference value of 7.62 m (25 ft) illustrated in Figure 3.2-3. Throughout all the above modeling, the collector row length remains an independent variable.

The piping sizes are defined in terms of two independent variables: the diameter of the warm manifold pipe and the diameter of the hot manifold pipe. The other hot and warm piping sizes are simply scaled up from these diameters, hot and warm, respectively, so as to have the same flow velocity in the manifolds as in the trunk lines. The warm side and hot side piping insulation thicknesses are two additional independent variables.

Although each pipe diameter is a continuously changing variable, the code always selects the nearest standard pipe size. Pipe weight and pressure loss are also determined from that pipe size. Valve selections are also based on the standard pipe sizes. The field piping costs are then determined using suitable unit cost data for the 316 stainless steel and carbon steel portions that allow for fittings and expansion loops.

The cost of the piping foundations is based on the total length of the piping runs. Valve costs are determined considering the size and number of valves and whether they are for the hot or warm side of the loop. Insulation costs take into account thickness as well as temperature rating. The fluid pump costs utilize a reference value with a cost scaling relationship proportional to $E^{0.7}$, where E is the pumping power (electrical).

The piping fluid inventory is calculated and costed. The costs of the salt purification system, the salt makeup system, and the gas cover and purge system are based on reference values which are scaled up (nonlinearly) with the piping fluid inventory.

TYPICAL SPACING OF CONCENTRATORS

Fig. 3.2-3. Collector arrangement

The head losses in each pipe length are computed using the standard Moody formula with the appropriate temperature-dependent density and viscosity for draw salt. The pressure losses in the manifolds, which are of constant diameter, take into account the reduced flow velocity at the end of the manifold. In practice, the manifold would be stepped, but the modeling for the stepping or tapering would have added a complexity that did not seem justified in terms of the system optimization.

The computed pumping power includes the pressure loss in the receivers, the field piping losses, the static head loss in the storage tank, and the pressure loss in the steam generator. The power is calculated for the maximum heat collection rate (12 p.m., March 21), which determines total pump capacity and cost, and then for 10 a.m., December 21. Pumps are assumed to have an overall 65% efficiency. The latter power is subtracted from the gross electric power as are the station auxiliaries.

The steam generator sizing, performance, and pressure loss characteristics were established for the reference design of 100 MW(e). Scaling relationships were utilized to adjust the cost of the steam generator for different plant ratings.

A further step in the HTS performance model is to compute salt temperatures around the system. The computation accounts for thermal losses in the insulation, thermal losses from storage, and the temperature rise across the salt pumps. These calculations are performed for the flow rate which gives the desired net output. During the day, flow rate varies so these temperature drops will be smaller or larger, but no account is made of such changes.

The trace heating system cost is determined by applying a unit cost to the rate of heat lost from the piping during a startup. During the startup, the ambient temperature is taken as 256 K (0°F) and both hot and warm piping temperatures are taken as 561 K (550°F). The unit cost comes from normalization to an earlier cost estimate of the total trace heating system.

An allowance is made for field-located buffer tanks and storage pumps which is proportional to the total heat collection rate of the field. Design and engineering costs are fixed irrespective of the size of the field.

Energy Storage Subsystem Model. The simple cost and performance model of the energy storage subsystem (ESS) described in this section was prepared for use in the system design code SS2. Earlier ESS models for SS2 had considered a pure thermocline, a rock-filled thermocline, and an N+1 tank system. In the latter case, the tanks may be used for either hot or warm storage. For practical reasons described in the previous section, these various models were abandoned in favor of the system described below. This system is comprised of the same number of hot and warm tanks, each dedicated to storage of hot or warm draw salt, respectively.

The desired heat storage H_s is

$$H_s = E_s / \eta_s,$$

where E_s = desired electric storage of 420 MW(e)-hr,

η_s = plant electric efficiency when operating from storage
(but burdened with collection pumping losses).

η_s is given by

$$\eta_s = \eta_N \cdot \eta_{st},$$

where η_{st} = storage efficiency, initially taken as 0.965 but found to be closer to 0.99 average instead.

The net plant efficiency is

$$\eta_N = \frac{100 \text{ MW(e)}}{Q},$$

where Q = total heat rate delivered to the steam generator.

The draw salt weight is

$$W_s = H_s / C_p (T_H - T_W),$$

where C_p = average specific heat between hot and warm storage temperatures, T_H and T_W , respectively.

The storage volume is calculated at the hot temperature as

$$V_s = W_s / \rho_s (T_H),$$

where $\rho_s (T_H)$ = density of the draw salt.

A 10% ullage is then added, i.e., $V_{s1} = 1.1V_s$.

The storage tank is based on an American Petroleum Institute design with a height limit of 12.19 m (40 ft) and a maximum wall thickness of 25.4 mm (1 in.) at the bottom in which the loop stress is limited to 68,945 kPa (10,000 psi). These requirements have the practical effect of setting the tank diameter D_s , in feet, to

$$D_s = 6000 / \rho_s.$$

The number of tanks n (either hot or warm) is then given by

$$n = V_{s1} / (10\pi D_s^2).$$

n is then rounded upward to the next integer n_1 and the tank diameter is reduced to D_{s1} , so that

$$n_1 \cdot 10\pi D_{s1}^2 = V_{s1}.$$

The heat loss from these tanks is then calculated assuming 102 mm (4 in.) of insulation around the walls and an average thermal conductivity of 9.65×10^{-6} J/m·K·s (0.06 Btu/hr-ft-°F).

The cost of the energy storage system is computed as follows. The hot tank is assumed to be made of stainless steel and the warm tank of carbon steel. The side wall thicknesses are assumed to reduce uniformly from 25.4 mm (1 in.) at the bottom to 6.35 mm (0.25 in.) at the top, i.e., an average thickness of 15.9 mm (0.625 in.). The weight of the tanks is calculated; and their costs are calculated by applying hot and warm unit costs which include allocations for labor.

The cost of the storage tank insulation is determined by applying an insulation cost per unit area to the total area of all the tanks.

Each tank is assumed to have a dike and a wall around it to contain any leakage. These items are costed by applying a unit cost to the total tank peripheral area, defined as that area in a 3.05-m (10-ft) wide perimeter around each tank.

Each tank is assumed to have a hot and a warm valve sized according to the maximum heat collection rate. Costs applied to the valves account for their sizing and material (316 stainless steel for the hot side and carbon steel for the warm side).

The cost of the tank foundation is obtained by applying a unit cost to the sum of the gross tank bases, defined as a 6.10-m (20-ft) larger diameter than the tank diameter.

Each tank is assumed to have 15.2 m (50 ft) of piping on the hot and warm sides connecting the tank to the respective supply and return lines. This piping is sized according to the maximum heat collection rate and costed accordingly with different unit costs for hot and warm sides. The pipes are insulated and costed in the same way as the tanks.

The final element of cost in the account is the draw salt inventory, calculated by applying a 30.9 cents/kg (14 cents/lb) cost to the bulk draw salt [33.1 cents/kg (15 cents/lb) was used in the final plant cost estimate].

In summary, the cost and performance model describes the tank costs as a function of the electrical energy storage requirement and the differential temperature across the hot and warm sides. In the range considered, the storage cost was less than 3% of the total plant cost, so the total system cost is little affected by the simplicity of the cost and performance model and assumptions therein.

Electric Power Generating System Model. It was initially intended to use the SCAP code to provide the modeling of the electric power generating system (EPGS) in the system design code SS2. Originally designed for the high-temperature gas-cooled reactor (HTGR), SCAP was modified for general application here by substituting temperature-dependent fluid properties* for the existing helium properties. The code was then employed to prepare a few point EPGS heat balances in the selection of a suitable heat transport fluid, discussed in Section 2.1. However, the fluid selection study showed that SCAP was too complex to be conveniently integrated into the system code. The complexity lies in the number of independent quantities that must be fed into the code which require judicious selection by a designer. These quantities include initial guesses of state point temperatures to ensure solution convergence, selection of the number of feed-water heaters, bleed flow coupling paths, and condenser backpressure, etc. It was therefore decided to employ a simple model of the EPGS which was normalized to the much more precise calculation of SCAP or the equivalent code at Bechtel National where the EPGS design was actually performed.

* Draw salt, Therminol 88, and sodium.

The EPGS efficiency model then was taken as

$$\eta_G = K(E) \frac{(T_1 - T_2)}{T_1},$$

where η_G = gross EPGS efficiency,

$K(E)$ = a normalization factor.

T_1 and T_2 are the maximum and minimum absolute temperatures in the cycle.

T_2 , the condenser temperature, is taken as fixed at 311 K (560°R). T_1 is taken as the hot salt temperature less a fixed approach of 28 K (50°F). In essence, a normalized Carnot cycle is employed. Such a model is clearly limited, particularly since it does not properly account for the pinch point at the preheater input or the approach temperatures of the superheater and reheater units. As a consequence of the severe limitation of this EPGS model, the SS2 code cannot be used to predict the most economic salt return temperature. However, its prediction of the most economic salt delivery temperature is considered satisfactory. The normalization factor $K(E)$ is given by 3413 divided by the gross turbine heat rate HR in Btu/kW-hr. HR is in turn a parabolic fit to the computer-generated data given below:

Gross Turbine Rating [MW(e)]	Heat Rate [kJ/kW-hr (Btu/kW-hr)]
105	8516 (8072)
315	8313 (7880)
525	8237 (7808)

The turbine auxiliary power is taken as the reference value of 4.334 MW for a 100-MW(e) net plant and then scaled up for large turbine ratings according to output to the 0.7 power.

The EPGS cost model is based on early estimates of each element of the subsystem for a 100-MW(e) plant that is then scaled according to gross electric output according to a suitable exponent. The elements are as follows:

Turbine: \$9.828 million, 0.75 exponent.

Heat rejection equipment: \$2.562 million, 0.8 exponent.

Condensing system: \$0.497 million, 0.8 exponent.

Feed heater: \$0.967 million, 0.8 exponent.

Working fluid treatment system: \$0.970 million, 1.0 exponent.

Design and Engineering: 17.18% of the sum of the above costs.

3.3 PARAMETER SELECTION

3.3.1 System Analysis Results

Initial point runs made with the SS2 cost/performance code in combination with results from the SUNPOW code (see Section 2.2) led to the conclusion that minimizing the receiver cost and increasing its performance would be the predominant means of improving overall system cost effectiveness. The primary step in the minimization has been to increase the solar concentration factor, which reduces the relative thermal losses and requires less receiver per square foot of collector. Based on this objective and support from results of SUNPOW studies, a revised reference receiver was adopted. The results reported here involve optimization and sensitivity studies performed using the reference receiver. The optimization and sensitivity studies reported in this section were completed prior to some of the latest performance model improvements and final performance calculations. No significant change in the trends found in the earlier analyses should result from the more precise calculation, and the studies have therefore not been repeated.

The design parameters allowed to be varied in the optimization performed were:

1. Included arc of the panel.
2. Panel length.
3. Mirror strip width.
4. Width of panel aperture.*
5. Mirror strip glass thickness.
6. Width of receiver surface.*
7. Receiver pipe diameter.*
8. Depth of secondary concentrator.
9. Receiver insulation thickness.

*This parameter is directly coupled to other parameters being varied and was optimized by the use of a coupling coefficient which the optimizer could vary.

10. Fluid outlet temperature.*
11. Reference warm pipe diameter.
12. Reference hot pipe diameter.
13. Storage size.**
14. Collector size (relative to rating).**

The parameters selected from the optimization studies are presented in Table 3.3-1. Certain parameters continued to vary in the optimization after the effect of their change had become relatively small. Where handling, potential problems, uncertainty, or other qualitative evaluation so indicated, these particular parameters were restricted from further change.

Initial optimization calculations were started with the objective of minimizing the BBCOE. It became obvious that designing to give 4 hr of direct operation on the worst insolation day, December 21, in order to obtain minimum capital cost while providing 420 MW(e)-hr of storage and optimizing on the BBCOE presented a dichotomy of goals. The EPGS and other elements such as buildings and switchgear represent essentially fixed costs for a given rating and are approximately 20% of the total plant cost. Reducing the unit cost of this fixed cost equipment by greater utilization will reduce the BBCOE for any given heat collection system design. Optimization on the BBCOE will therefore make the plant sized for the design day have a net increase in capacity factor, as long as the benefit of increased capacity factor outweighs the increase in collection system cost.

The optimization objective was therefore changed to "minimize the cost of thermal energy" so as to provide the most cost effective plant with the reference plant sized to fully utilize the 420-MW(e) storage. Subsequently, investigations of resizing the plant for different capacity factors and storage capacities to achieve a reduced BBCOE were conducted.

*Set at 842 K (1054.8°F) in the final optimization owing to practical constraints already mentioned.

**Storage size and collection size were fixed during the reference optimization and evaluated in later sensitivity studies.

TABLE 3.3-1
DESIGN PARAMETERS SELECTED FROM OPTIMIZATION DATA

Parameter	Value
Panel arc	96 deg
Panel length	11 m (36 ft)
Mirror strip width	50.8 mm (2.0 in.)
Panel aperture	3.61 m (11.833 ft)
Mirror strip glass thickness	2.29 mm (0.09 in.)
Receiver width	53.3 mm (2.1 in.)
Receiver pipe diameter	31.1 mm (1.225 in.)
Depth of secondary concentrator	38.1 mm (1.50 in.)
Receiver insulation thickness	34.9 mm (1.375 in.)
Collector row length	274 m (900 ft)
Reference warm pipe diameter	140 mm (5.5 in.)
Reference hot pipe diameter	140 mm (5.5 in.)

Sensitivity studies were performed for selected major plant parameters that were varied about the reference design values. All sensitivity studies involved variation of the selected variables with all others at the reference value except for those cases where a direct relation necessarily exists. For example, the panel length cannot be varied with a fixed number of panels without changing the total length, causing a mismatch of fluid velocity, and distorting the result. Panel length was therefore adjusted along with the number of panels to retain essentially a fixed total length. When the number of panels was changed (for total length study), the receiver pipe diameter was adjusted to maintain essentially a fixed heat transfer efficiency.

Parameters exhibiting large effects on cost (high-sensitivity set) are shown in Figure 3.3-1. Data are plotted for the change in BBCOE (with 420 MW(e)-hr storage) as a function of the parameter being varied. Parameters plotted include panel width (obtained by adding even numbers of reference width mirror strips), mirror strip width (accompanied by equivalent receiver width change), outlet temperature, and panel length. Increasing the panel width reduces the BBCOE despite the increase in unit panel costs because the relative cost of the receiver is reduced and because the receiver performance improves with the higher solar concentration factor.

The selected panel width was 3.61 m (142 in.), since further increases would engender potential handling problems and additional reductions in the BBCOE would be small.

Mirror strip width was limited to 50.8 mm (2 in.), the high side of the apparent minimum at 45.7 mm (1.8 in.), owing to the lack of detailed evaluation of receiver position loss. Narrower strips are desirable despite increased edge loss because a narrower receiver results and a higher concentration is achieved. However, the smaller the receiver, the greater are the miss loss and secondary concentrator loss resulting from receiver position deviation.

Increasing the receiver fluid outlet temperature results in a lower BBCOE, as shown in Figure 3.3-1, due primarily to the increasing EPGS efficiency,

Fig. 3.3-1. Sensitivity of bus bar cost of electricity (BBCOE) to selected parameters (high-sensitivity set) (data from first plant capital costs)

since the collector system efficiency suffers only a slow decline. The selected fluid temperature of 839 K (1050°F) (actual steam generator input temperature) is somewhat above the minimum cost of thermal energy and less than the minimum BBCOE. This temperature was selected because gains from higher values appear small, while uncertainties about decomposition rates and material corrosion rates become larger. Practical experience with molten salt systems above a 839 K (1050°F) operating temperature is quite limited. Thus, it is inappropriate to advocate higher temperatures and expect to achieve a reasonably credible design. The 839 K (1050°F) temperature is also the temperature selected by Martin Marietta (Ref. 3.3-1) for their central receiver solar energy system employing draw salt.

Selection of the warm salt temperature was not made on the basis of SS2 optimization results because that temperature does not feature in the simplified EPGS model in that code. The impact of lowered warm salt temperature somewhat increases collector field efficiency and modestly reduces ESS and HTS cost. However, these effects are small compared with the variation of the EPGS efficiency due to the permissible feedwater temperature. These results which are given in Table 3.3-2 and shown graphically in Figure 3.3-2, clearly indicate the sensitivity. The warm salt temperature must be high enough to permit a feedwater temperature as high as is reasonable without the pinchpoint at the boiler inlet becoming excessively small. A further criterion on the warm salt was that it have a reasonably wide margin over its melting point of 494 K (429°F) to ensure that no blockage occurs during any startup/shutdown transient or during intermittent cloudiness. The selected temperature of 561 K (550°F) (steam generator exit) appears to be a reasonable compromise among these requirements and is also the same as that adopted by Martin Marietta (Ref. 3.3-1) for their central receiver solar energy system.

Panel length was varied between 6.4 and 13.0 m (21 and 42.5 ft). The selected length of 11.0 m (36 ft), while short of the system minimum, is considered to be a reasonable upper limit based on the receiver span design criterion that the center sag not exceed 6.35 mm (1/4 in.). A further factor that limits the receiver length is that the length must be consonant

TABLE 3.3-2
SUMMARY OF PARAMETRIC DATA ON EPCS PERFORMANCE

Turbine Throttle Pressure [MPa (psig)]	Superheat/ Plant Reheat Temp [K (°F)]	Feedwater Temp K (°F)	Hot Salt Temp [K (°F)]	Warm Salt Temp [K (°F)]	Salt Flow [10 ⁶ kg/hr (10 ⁶ lb/hr)]	Gross Turbine Heat Rate [kW(e)/kW-hr (Btu/kW-hr)]	Feedwater Flow [10 ⁶ kg/hr (10 ⁶ lb/hr)]
13.79 (2000)	839 (1050)	450 (350)	894 (1150)	529 (493)	2.71 (5.98)	2.394 (8170)	282.9 (623.7)
		478 (400)	894 (1150)	543 (518)	2.80 (6.17)	2.373 (8098)	291.4 (642.4)
		506 (450)	894 (1150)	557 (543)	2.88 (6.36)	2.352 (8026)	302.5 (666.8)
		533 (500)	894 (1150)	568 (563)	2.98 (6.58)	2.354 (8033)	333.2 (734.6)
13.79 (2000)	783 (950)	450 (350)	839 (1050)	544 (520)	3.48 (7.67)	2.474 (8445)	307.6 (678.2)
		478 (400)	839 (1050)	556 (541)	3.59 (7.91)	2.451 (8364)	317.2 (699.4)
		506 (450)	839 (1050)	568 (562)	3.71 (8.17)	2.427 (8284)	330.0 (727.5)
		533 (500)	839 (1050)	577 (579)	3.83 (8.45)	2.424 (8272)	363.9 (802.2)
13.79 (2000)	728 (350)	450 (350)	783 (950)	562 (551)	4.79 (10.57)	2.568 (8766)	337.2 (743.4)
		478 (400)	783 (950)	571 (568)	4.96 (10.93)	2.544 (8681)	348.5 (768.3)
		506 (450)	783 (950)	581 (585)	5.14 (11.33)	2.517 (8592)	363.4 (801.2)
		533 (500)	783 (950)	588 (598)	5.30 (11.69)	2.506 (8554)	401.2 (884.5)
13.79 (2000)	672 (750)	450 (350)	728 (850)	582 (588)	7.64 (16.84)	2.677 (9168)	374.2 (825.0)
		478 (400)	728 (850)	589 (600)	7.92 (17.45)	2.658 (9071)	387.5 (854.3)
		506 (450)	728 (850)	596 (612)	8.22 (18.13)	2.628 (8971)	405.6 (894.1)
		533 (500)	728 (850)	601 (621)	8.49 (18.72)	2.611 (8910)	449.9 (991.8)
16.55 (2400)	819 (1050)	450 (350)	894 (1150)	531 (496)	2.69 (5.94)	2.367 (8079)	278.1 (613.1)
		478 (400)	894 (1150)	545 (521)	2.78 (6.12)	2.345 (8004)	286.2 (631.0)
		506 (450)	894 (1150)	558 (545)	2.86 (6.31)	2.323 (7928)	296.9 (654.6)
		533 (500)	894 (1150)	568 (563)	2.93 (6.45)	2.314 (7896)	324.8 (716.1)

TABLE 3.3-2 (Continued)

Turbine Throttle Pressure [MPa (PSIG)]	Superheat/ Plant Reheat Temp [K (°F)]	Feedwater Temp K (°F)	Hot Salt Temp [K (°F)]	Warm Salt Temp [K (°F)]	Salt Flow [10 ⁶ kg/hr (10 ⁶ lb/hr)]	Gross Turbine Heat Rate [kW(e)/kW-hr (Btu/kW-hr)]	Feedwater Flow [10 ⁶ kg/hr (10 ⁶ lb/hr)]
16.55 (2400)	783 (950)	450 (350)	839 (1050)	548 (527)	3.45 (7.67)	2.449 (8360)	303.2 (668.4)
		478 (400)	839 (1050)	560 (548)	3.59 (7.91)	2.425 (8276)	312.4 (688.7)
		506 (450)	839 (1050)	572 (570)	3.71 (8.19)	2.400 (8192)	324.9 (716.2)
		533 (500)	839 (1050)	580 (584)	3.80 (8.37)	2.384 (8137)	355.6 (784.0)
16.55 (2400)	728 (850)	450 (350)	783 (950)	569 (565)	4.91 (10.82)	2.549 (8701)	333.9 (736.1)
		478 (400)	783 (950)	578 (581)	5.07 (11.17)	2.522 (8607)	344.6 (759.7)
		506 (450)	783 (950)	587 (597)	5.23 (11.54)	2.494 (8513)	359.3 (792.2)
		533 (500)	783 (950)	593 (608)	5.35 (11.79)	2.470 (8429)	394.0 (868.7)
16.55 (2400)	672 (750)	450 (350)	728 (850)	594 (610)	8.30 (18.30)	2.674 (9127)	372.7 (821.7)
		478 (400)	728 (850)	601 (621)	8.60 (18.95)	2.643 (9021)	385.4 (849.7)
		506 (450)	728 (850)	606 (631)	8.88 (19.58)	2.612 (8915)	403.7 (890.0)
		533 (500)	728 (850)	610 (638)	9.07 (19.99)	2.581 (8808)	445.8 (982.9)

Fig. 3.3-2. Gross turbine heat rate

with the length of the concentrator panel. Panel lengths of much greater than 11.0 m (36 ft) are extremely unwieldy and costly to handle. In fact, this was the reason that two 5.49-m (18-ft) long panels were employed instead in the selected design.

Figure 3.3-3 presents data for low-sensitivity parameters; the scale on this plot is 10 times that of Figure 3.3-1. The first data are for the number of 11.0-m (36-ft) long panels (or total length). In generating these data, the receiver pipe diameter was varied proportionally with length to maintain equivalent heat transfer and pressure drop, while the remainder of the system was maintained at the reference values. It is believed that longer lengths might continue to show some economic benefit if field piping sizes and/or field layout changes were made to correspond to longer collector row lengths. The total effect is, however, relatively small. The selected length of 274 m (900 ft) (25 panels) was kept short of the apparent minimum to minimize potential difficulties with transport delay, drainage, and thermal expansion.

The panel included arc was varied from approximately 92 to 102 deg. The observed small dependence is attributed to the trade-off between the larger optical dispersion (greater radius) of the small arc and the increased incident energy on the secondary concentrator and smaller apparent (projected) secondary concentrator aperture for the outer mirror strips of the larger arc. The 96-deg arc chosen is near the minimum of this shallow function.

The reference field pipe diameters (hot and warm) were varied in consort, since the optimization program had changed both from an initially larger value to equal 140-mm (5.5-in.) values [actual standard size of 128 mm (5.047 in.) was used in calculations]. The "reference diameter" actually refers to the piping diameter on the supply and return manifolds. The other pipes are sized from the "reference" on a "constant velocity" basis to the nearest standard pipe size. The system cost performance is affected by heat losses in the piping, the cost and size of the pipe, and the pressure effects on the cost of the pipe. As shown in Figure 3.3-3, the

Fig. 3.3-3. Sensitivity of bus bar cost of electricity (BBCOE) to selected parameters (low-sensitivity set) (data from first plant capital costs)

minimum in this cost is quite small, approximately 142-mm (5.6-in.) diameter. The resulting standard pipe sizes are given in Table 3.3-3.

TABLE 3.3-3
OPTIMUM PIPE DIAMETERS PREDICTED BY SS2 (see Fig. 3.3-3)

Warm side:	128 mm (5.047 in.) (supply manifold)
	203 mm (7.981 in.) (header)
	255 mm (10.02 in.) (trunk)
Hot side:	128 mm (5.047 in.) (return manifold)
	203 mm (7.981 in.) (trunk)

The preceding pipe dimensions were not adopted because they produced a high system pressure level [~ 2758 kPa (400 psi) in the receiver]. The SS2 code had no penalty provisions for this pressure level, but a practical limit of 1172 kPa (170 psi) on the flexible downcomers and uprisers had been quoted by the supplier of such units for the solar salt test rig at GA. The solution, of course, was to use more pumps distributed throughout the HTS. In the meantime, Bechtel National had concluded that a non-pressure-balanced piping arrangement with stepped pipes on the supply and return manifolds of much larger diameter than those given in Table 3.3-3 would be optimum. Such a system had not been modeled within SS2, and time limitations prevented an assessment within the code of this alternate arrangement which had the advantage of meeting downcomer/upriser pressure limitations. The Bechtel-designed piping was therefore adopted. However, the resulting cost of the HTS was found to be relatively high, and the substantial difference in pipe dimensions between the adopted values and those of Table 3.3-3 suggests that further optimization of the system could be beneficial.

The glass thickness for the mirror strips was varied from 0.762 to 3.30 mm (0.03 to 0.13 in.). While edge losses decrease with thinner glass, the increase in purchase and handling costs rapidly offsets this advantage. The less than 1 mill/kW-hr BBCOE difference between 3.18- and 1.52-mm (0.125- and 0.06-in.) standard glass thicknesses represents a fairly small impact. Selection of the mirror strip glass is therefore based on handling

and breakage considerations. (A discussion of glass thickness with respect to hail damage is given in Section 3.5.)

While certain other design parameters have some impact on the BBCOE, those considered in this study were evaluated as the most significant "non-constrained" parameters of the heat collection subsystem. A second area requiring investigation is the relation of the heat collection subsystem and the storage/EPGS subsystems to the optimum plant size. The study of this area is discussed below.

As noted earlier, the EPGS and some other portions of the plant are essentially a fixed cost for a given plant rating and specific "warm" and "hot" fluid conditions. A BBCOE benefit potentially exists in providing larger collection systems and storage subsystems to more fully utilize the "fixed cost" portions of the system, which constitute approximately 20% of the cost of a plant designed to meet the minimum "4 hr of direct rated operation on December 21." More detailed studies of the impact on BBCOE of thermal storage capacity are described in Section 5.

As noted in Section 2.2.3, the more detailed receiver performance model, developed under GA funding, has been incorporated into the SUNPOW and SS2 codes. The new receiver model and several other minor code improvements provide an accurate calculation of performance and predict an increased collection efficiency for a given set of optical and mechanical properties. The final selected reference optical properties (see Sections 4.2 and 4.3) and EPGS-heat transport performance data from Bechtel National were incorporated in the SS2 code. Performance calculations with the revised data show that the collector system is slightly larger than required to "just fill" the 420 MW(e)-hr of storage.

An analysis was performed to assess the impact on BBCOE of varying the number of collector rows with storage fixed at the reference value. An assumption of the analyses was that energy greater than that which could be used directly at 100% load or placed in the reference storage would not be collected. Figures 3.3-4(a) and (b) present the BBCOE and percent loss of

Fig. 3.3-4(a). Bus bar cost of electricity versus number of collector rows

Fig. 3.3-4(b) Percent annual loss of collectable energy versus number of collector rows

annual collectable energy, respectively. As may be seen from these figures, 932 rows would just fill the reference storage from energy collection at rates in excess of that required for 100% electrical load. However, the BBCOE decreases until 970 rows are provided. This decrease is due to the greater utilization of fixed cost equipment throughout the year while only providing "overcollection capacity" for a few minutes of a few days of the year. As more rows are added, the increasing cost of collection equipment and increasing portion of uncollected energy result in a BBCOE increase.

The field size described previously (Ref. 3.3-2) was based on a collector field sized to deliver rated electrical load at 10 a.m. on December 21. General Atomic was subsequently directed to size the field to fully utilize the 420-MW(e) storage. The specified field size is 972 rows. The latest performance calculations with improved modeling show the field to be slightly larger than required to achieve the minimum BBCOE with the fixed reference storage. The difference is small and does not warrant changing the field, especially since it would perturb the then ongoing field layout and plant cost estimate.

The data of Figures 3.3-4(a) and (b) assume that excess collectable energy would be discarded. In fact, at least two potential methods exist for using all or most of the energy. The EPGS may be operated at 105%, thus reducing the amount going to storage. Alternatively, some of the hot fluid may be routed to the warm tank, raising its temperature such that late afternoon flow is higher and design outlet temperature is maintained longer. Such operating strategies would result in a minimum BBCOE for a larger field.

3.3.2 Optical/Thermal Data

This section discusses the various optical and thermal data pertinent to collector and receiver elements that have a significant impact on plant performance. A complete listing of all optical parameters from the SUNPOW code used in the final calculation is given in Table 3.3-4, and Figure 2.2-1 illustrates the energy loss train.

TABLE 3.3-4
REFERENCE OPTICAL PROPERTIES

Concentrator Mirror Reflectivity	0.94
Cover Glass Refraction Coefficient	1.515
Cover Glass Thermal Conductivity	0.0724 J/s·m·K (0.450 Btu/hr·ft·°F)
Cover Glass Solar Transmissivity	0.900
Cover Glass Solar Absorptivity	0.030
Cover Glass Thermal Absorptivity	0.880
Cover Glass Emissivity	0.88
Absorber Plate Solar Absorptivity	0.940
Absorber Plate Thermal Absorptivity	0.300
Absorber Plate Solar Reflectivity	0.06
Absorber Plate Thermal Reflectivity	0.70
Absorber Plate Thermal Emissivity	0.30
Secondary Concentrator Solar Reflectivity	0.88
Secondary Concentrator Thermal Emissivity	0.25
Insulation Casing Thermal Reflectivity	0.60
Insulation Casing Thermal Emissivity	0.40

The reflectivity of the back-silvered mirrors (FREFL) on the primary concentrator is taken as 0.94. Such a value has been achieved with 2.29-mm (90-mil) thick glass of low reduced iron content.

The dirt loss factor (FDIRT) is taken as zero. While this is clearly incorrect for a practical design, the loss is both site specific and dependent on the frequency of mirror cleaning. Therefore, the zero loss was selected to facilitate DOE's comparison with other line focus concentrator studies.

The receiver shadow loss factor (FSHADO) has been taken as 5% averaged over the daily interval. The loss is intended to encompass the shadowing of the primary concentrator by the receiver and its support structure. It was originally intended to calculate the shadowing over the day in the SUNPOW code, but the programming was not implemented owing to shortage of time. The receiver nominal shadowing loss is its width of 124 mm (4.90 in.) divided by the concentrator aperture of 3.61 m (11 ft 10 in.), i.e., 3.43%. This loss occurs only when the sun lies within the concentrator aperture angle divided by 8 (12 deg for the reference design), assuming a centrally located tangent facet. However, much light is collected when the sun lies outside this angle. Consequently, the allocation of a 5% loss for shadowing by the receiver and its support structure is considered conservative.

The receiver position loss factor (FPOS) of 4% refers to the loss that occurs because the receiver is incorrectly positioned. The malpositioning arises from dead band in the receiver motor drive, positioning accuracy [± 3.18 mm (1/8 in.) tolerance is the objective], and receiver sag between supports [maximum allowable of 6.35 mm (1/4 in.)]. The 4% value taken for FPOS is considered reasonable.

A receiver coupling loss (FCOUP) of 1% is taken. Since a continuous heat absorber duct is now utilized, there should be no such loss and the presence of a 1% factor now represents an unnecessary conservatism. The 1% allowance was inserted when it was thought that a welded or brazed heat absorber surface would be used instead of the mechanically attached one of

the reference design. Since welding or brazing can impair the selective absorptive coating in the vicinity of the weld or the braze, a modest loss was assumed.

The reflectivity of the secondary concentrator wings was taken as 0.88, the quoted value for Kinglux, a polished anodized aluminum. This corresponds to an absorptivity (FSECON) of 0.12. The use of a back-silvered mirror which has higher reflectivity (0.94) but higher cost is an additional choice, but it has not been used in the reference design. Polished electroplated or sputtered gold is a further possibility offering potentially greater cost effectiveness.

The receiver cover glass properties have a significant impact on system performance and were therefore selected carefully. A borosilicate glass is used, since the glass temperature may go up to 589 K (600°F) and strength at temperature is required. In addition, borosilicate glass has a higher thermal conductivity and lower coefficient of thermal expansion, so it is more resistant to cracking than ordinary glass when subjected to temperature transients. The plant cost estimate is based on Tempex, a borosilicate glass manufactured by the Schott Glass Company; however, Corning Pyrex 7740 has a similar composition (it is made by a different process) and is believed to have similar optical properties. The normal spectral reflectance, R , of Pyrex 7740 in the optical spectrum is reported in Ref. 3.3-3, and an average value of 0.07 appears conservative. The normal transmission, T , of light through 12.7 mm (0.50 in.) of Pyrex 7740 is given as a function of wavelength in Ref. 3.3-4. The transmission, T' , for the reference thickness of receiver glass, 2.29 mm (90 mil), is determined by the following formula:

$$T' = (T + R) 2.28/12.7 - R .$$

The resulting transmission ranges from 0.87 to 0.92 over the optical frequency range. A value of 0.90 appears quite reasonable and has been used in the SUNPOW code. The solar absorption (AGS) is then simply determined by difference as 0.03.

While the above values pertain to the normal transmission and reflectivity, the conservative value of R employed mitigates against the fact that the concentrated light on the receiver cover glass departs somewhat (up to 24 deg) from normal incidence.

The normal transmission of infrared radiation through several thicknesses of Pyrex 7740 was measured at GA, and the data are shown in Figure 3.3-5. The frequency-dependent transmission for 2.29-mm (0.090-in.) thick glass was obtained by interpretation and was then weighted over various black body spectra* to give the following results:

Temperature, K (°F)	589 (600)	728 (850)	867 (1100)
Transmission	0.012	0.032	0.057

Since the bulk of the emissive heat loss comes from the hot end of the receiver and since SUNPOW is only set up to use a single value, the average thermal transmission (TCL) of the cover glass was taken as 0.050. The normal spectral reflectance of Pyrex 7740 in the infrared range averages at about 0.07 (Ref. 3.3-3). Hence, by difference, the thermal absorption (AGL) of the glass was taken as 0.88.

The thermal conductivity (TCGI) of the calcium silicate insulation was taken from Ref. 3.3-6 as an average of 7.24×10^{-5} kJ/m·K·s (0.045 Btu/hr-ft-°F). This value is intended to reflect an average over the entire collector row in which the fluid temperature varies from 568 to 843 K (562° to 1057°F) while the duct skin temperature is 33 K to 22 K (60° to 40°F) hotter (respectively.) The 7.24×10^{-5} kJ/m·K·s (0.045 Btu/hr-ft-°F) value corresponds to the point value at about 506 K (450°F). Although this may be a reasonable average temperature of the receiver insulation, the averaging process is crude and the thermal conductive losses are likely to be under-predicted. Use of a lower-conductivity insulation innermost to the receiver at the hot end of the collector row provides a backup solution should the

* $\phi(\lambda) = C_1 \lambda^{-5} / (e^{C_2/\lambda T} - 1)$, where $C_1 = 3.74 \times 10^{-5}$ ergs-cm²/sec and $C_2 = 1.4387$ cm-K, per Ref. 3.3-5. T is in Kelvin and λ is in cm.

Fig. 3.3-5. Normal transmission through Pyrex 7740

observed thermal losses be significantly different from those predicted for this design.

The most significant parameters affecting the collector field performance are the solar absorptivity (α = APS) and thermal emissivity, ϵ [taken as equal to the thermal absorptivity (APL)]. The values corresponding to black chrome were initially utilized, although it was recognized that black chrome would degrade (increase in ϵ mostly) at the elevated temperatures of this receiver design. General Atomic believes that a suitable selective coating will be developed; indeed, the performance of the central receiver system can benefit from such a coating. Thus, the black chrome properties α = 0.93 and ϵ = 0.13 provided initial results reported to DOE (Ref. 3.3-2) concerning the field size and plant performance. In addition, the coating question and its development encouraged the notion of a replaceable absorber surface, a concept which is retained in the reference design. Nevertheless, it was felt necessary to survey the coating development field to determine what data sources should be utilized to provide the reference values for the selective absorbing surface in the receiver.

A brief literature search (Refs. 3.3-7 and 3.3-8) was conducted, and the results together with data on file at GA were used to ascertain the most promising leads for solar selective surfaces available or in development which are stable in air up to 867 K (1100°F). From an initial list of about 60 surfaces, a much shorter list (Table 3.3-5) was prepared which also includes the GA surface and Pyromark paint for comparative purposes. The table itself shows a broad-based development effort with results which are quite promising. General Atomic has adopted the Cornell University surface properties in Ref. 3.3-9 for its receiver design because it appears to be best qualified at this juncture, having been exposed in air at the desired temperature range, and because its properties were measured at close to the proposed receiver operating temperatures. This selection is not intended to reflect a critical judgment on the other coatings, since it is clear that development progress could readily change the relative status. However, based on the published information, the Cornell University

TABLE 3.3-5
COMPARISON OF REPORTED PROPERTIES OF SOLAR SELECTIVE ABSORBING SURFACES

Surface/Description	Qualifications	α	ϵ	Exposure Temperature	Reference
Dornier, W. Germany Solartex electro-deposited	Not believed to be tested in air at this temperature.	0.93-0.96	0.14-0.18	700°C	3.3-8
Englehard, CuO/Au Englehard, CuO/Au	This coating believed no longer in use.	0.9 0.8	0.1-0.2 0.06	<500°C ~600°C	3.3-7 3.3-8
Englehard, CuO/Ag/Rh ₂ O ₂		0.9	0.1	<400°C	3.3-8
Englehard, CuO/Rh ₂ O ₂ /Ag organic coating	Coating deposited only on quartz thus far.	0.91	0.07-0.08	500°C in air for 1000 hr	Private communication to W. Walker (GA)
University of Arizona, Si ₃ N ₄ /Si/Cr ₂ O ₃ /Ag/oxide on stainless steel		0.8	0.1	500°C	3.3-7
University of Arizona, Si/Ag bilayer		0.8	0.07	500°C in vacuum	3.3-8
University of Arizona, black molybdenum		0.9	0.12	500°C in vacuum	Private communication to W. Walker (GA)
Colorado School of Mines, Electrodeposited black cobalt		0.95	0.02	Unknown	<u>Energy Abstracts</u>
Cornell University, Pt/Al ₂ O ₃	Optical properties measured at 500°C.	0.94	<0.3	600°C in air for 300 hr	3.3-8
General Atomic, CoCu/Au on stainless steel		0.88	0.35	1000°F in air	Private communication to W. Walker (GA)
Pyromark paint	Commercially available. Properties at temperature.	0.96	0.75	1100°F in air	Company literature

surface appears to be a conservative choice among the surfaces of Table 3.3-3.

It should be noted that Pyromark paint is an established durable surface which is easy to apply and maintain. Its high absorptivity is negated by its high emissivity. However, it is quite possible that the use of Pyromark paint in the low-temperature portion of the receiver, say less than 644 K (700°F), with the selective coating used on the hottest portion of the receiver could lead to either an improved performance or a more cost-effective system since paint costs less than the specialized selective surface coatings.

A study of the impact of selective coating properties on collector performance was conducted, and the results are shown in Figure 3.3-6. The ordinates are the average daily collector output per square foot of mirror. The study shows that three other surface properties give performance approximately 10% better than the reference one adopted for this plant design. These are the black chrome (Ref. 3.3-10), the Dornier surface (Ref. 3.3-7), and the Engelhard surface (Ref. 3.3-7). The analysis suggests that a significant performance improvement in the system will be obtained as the selective surfaces are further developed. This improvement has not been used for the first plant cost estimate reported in Section 5.

REFERENCES

- 3.3-1 "Conceptual Design of an Advanced Central Receiver Power System," Martin Marietta Company, DOE Contract EG-77-C-03-1742, January 1979.
- 3.3-2 Schuster, J. R., et al., "Line Focus Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.

Fig. 3.3-6. Effect of absorber coating on heat collection

- 3.3-3 Touloukian, Y. S., Thermophysical Properties of High Temperature Materials, Vol. 4, Part 2, Macmillan, New York, 1967, p. 1709.
- 3.3-4 Touloukian, Y. S., Thermal Radiative Properties, Vol. 8, 1972, pp. 1593-1594.
- 3.3-5 Baumeister, T., Standard Handbook for Mechanical Engineers, McGraw-Hill, New York, 1958, pp. 4-108.
- 3.3-6 PabCo Data Sheet, Industrial Insulation Division, San Francisco, California.
- 3.3-7 "Thermal Power Systems Workshop on Selective Absorber Coatings," December 1977, Golden Colorado, SERI 6-8.
- 3.3-8 Call, Patrick J., "National Program Plan for Absorber Surfaces R&D," SERI/TR-31-103, January 1979.
- 3.3-9 Solar Energy Materials 1 (1), 105 (Feb. 1979).
- 3.3-10 Pettit, R. B., and P. R. Sowell, "Solar Absorptance and Emittance Properties of Several Solar Coatings," J. Vac. Sci. Technol. 13, (2), 596 (1976).

3.4 UTILITY DATA

3.4.1 Introduction

The economic utilization of solar-generated electric power will depend strongly on the characteristics of the particular electric company utility grid to which it is connected. Southwestern Public Service Company (SPS) has supplied load and cost data specific to its grid to provide a data base for investigation of the impact of coupling to a solar plant. The characteristics of SPS are therefore presented briefly below.

Southwestern Public Service Company is an integrated electric generation, transmission, and distribution system serving a 116,500-square-kilometer (45,000-square-mile) area comprising the Panhandle and South Plains of Texas, the Panhandle of Oklahoma, Southeastern New Mexico, and one county in Southwestern Kansas. Because of rising fuel costs and increasing regulations governing the use of natural gas, SPS made the decision in 1972 that no more gas-fired generating units would be built. To meet the growing electric power demands of the area, SPS completed its first two coal-fired 356.7-MW units, one in 1976 and one in 1978. An identical unit will come on-stream in 1980. A 565-MW unit will be completed in 1979 as a co-generation project with a major chemical plant customer. The construction program from 1974 through 1982 represents a 76.6% increase in generation capacity.

The characteristics of the SPS system load are established primarily by five types of customers: residential, commercial, industrial, rural, and governmental or municipal. Each type of load represents a fairly equal portion of the total load, and each has been growing at approximately the same rate for the past several years.

3.4.2 Load Profiles

Figure 3.4-1 shows the daily and hourly load distribution of the total SPS system generation for the week of July 24-30, 1977. Figure 3.4-2 is a

Fig. 3.4-1. Daily load profile for SPS system generation for July 24-30, 1977

Fig. 3.4-2. Daily load profile for SPS system generation for December 18-24, 1977

similar plot of the hourly loads for the week of December 18-24. Comparison of these two figures shows some aspects of interest for a solar thermal electric power plant design:

1. The peak summer demand is approximately 50% greater than the peak winter demand.
2. The summer peak occurs approximately 5 hr after noon.
3. The summer peak fall-off at the weekend appears small.
4. The winter peak is flat, and any maximum usually occurs in the morning.

The duration and timing of the peak summer load are such that the proposed 4-hr storage for the LFSCPS is insufficient to shift supply to meet demand. Thus, the 4-hr storage would not permit reduction of the amount of spinning reserves necessary to assure firm power to SPS customers. It is important to stress that any solar plant must have a high degree of availability to a utility. Otherwise, the utility must maintain additional power reserves at additional capital cost. Therefore, a standby fossil-fired salt heater is necessary if the solar plant is to receive full capacity credit. Calculations presented in Section 5 of this report also show that storage will improve availability as well as BBCOE.

3.4.3 Incremental Cost Data

Typical cost curves as represented by the hourly incremental cost of generation for the corresponding weeks are shown in Figures 3.4-3 and 3.4-4. These figures show that there is a slight correlation of the generating cost and incremental cost with the variations in the peak load, but this is not a very significant influence. The incremental cost ranges from approximately 13 to 15 mills per kilowatt-hour for most of the time, so there is not a large variation in this cost. Table 3.4-1 gives the average fuel cost in dollars per million Btu, the generating cost for fuel only in mills

Fig. 3.4-3. Hourly incremental cost of SPS system generation for July 24-30, 1977

Fig. 3.4-4. Hourly incremental cost of SPS system generation for December 18-24, 1977

TABLE 3.4-1
SOUTHWESTERN PUBLIC SERVICE COMPANY GENERATING COST DATA
1977 MONTHLY AVERAGES

Month	Fuel Cost [\$/ 10^6 kJ (\$/MM Btu)]	Fuel Generating Costs (mills/kWh)	Total Operating Cost (mills/kWh)
January	1.42 (1.35)	14.29	15.32
February	1.36 (1.29)	13.80	14.97
March	1.45 (1.37)	14.18	15.21
April	1.48 (1.40)	14.92	16.03
May	1.50 (1.42)	15.09	16.14
June	1.48 (1.40)	15.07	15.98
July	1.46 (1.38)	14.98	15.83
August	1.42 (1.35)	14.45	15.38
September	1.42 (1.35)	14.71	15.72
October	1.53 (1.45)	15.24	16.33
November	1.66 (1.57)	16.30	17.62
December	1.57 (1.49)	15.82	17.10

per kilowatt-hour, and the total operating cost in mills per kilowatt-hour for each of the months of 1977. These cost data do not include investment charges. The data assist in establishing the value of replacement solar generation.

The above costs, of course, will be different for other utility companies depending on their mix of generation, fired by gas, by oil, or by coal. Therefore, these figures, though valid for SPS, must be evaluated for each particular utility company that is considering the application of solar power.

3.5 OPERATIONAL AND ENVIRONMENTAL CONSIDERATIONS

This section describes the various operational factors that impacted the design approach. Included are a discussion of startup and shutdown aspects, maintenance considerations, and environmental factors and a discussion of the resistance of the FMSC to hail.

3.5.1 Collector Row Length Limitations

A major question in the program has been determining the design criteria for the collector row length. Ideally, it should be as long as possible, since this reduces optical end losses and amortizes valve, downcomer, and receiver drive motor costs over a large mirror surface. However, two practical limitations are the need for convenient vehicular access and the restriction imposed by the topography. Land is rarely totally flat. A further criterion is the fact that as the row length increases, the flow must also increase, so a larger receiver heat transfer duct is required. However, the receiver dimensions generally center around the diameter of the heat receiver duct, and the bigger the receiver is, the greater its costs are, the bigger its shadowing loss on the concentrator panel is, and the larger its heat loss is. Based on these general considerations, a row length of 274 m (900 ft) was initially set as an upper limit for the reference design. Cost optimization studies subsequently showed that a longer length would be more economical.

However, the most significant criteria pertinent to row length arise from plant startup requirements. It has been shown that the molten salt cannot stay in the receiver duct overnight without solidifying; consequently, the receiver is drained at night into the header pipes. The salt would not solidify overnight in some of the header pipes, but it is not desirable to leave the hot salt there for two reasons:

1. The heat loss of the pipes would be increased from 3% (averaged over the isolation interval) to about 8% (averaged over the 24-hr day).

2. Since the pipes have to be the repository for salt from the receivers, they must be drained in part. If they must be drained in part, they can readily be drained completely.

The above considerations led to the philosophy that both the receivers and pipe lines should be drained overnight and during periods of continued low insolation. Criteria for gravity drainage were developed (Ref. 3.5-1), and it was shown that modest slopes would be sufficient to drain the molten salt into sumps where it could stay overnight or be transferred to the thermal storage system. If necessary, some assistance could be provided from a nitrogen purge supply.

Electrical trace heating is used to heat the inlet and outlet pipes to 561 K (550°F) before any fluid is pumped through them from the thermal storage system. Electrical trace heating is utilized rather than steam trace heating even though the latter is thermodynamically more efficient. The reason is that electrical trace heating is operationally easier: it may be turned on or off faster, it requires no auxiliary steam boiler, and it does not have condensate return lines which could be frozen and might need to be electrically trace heated themselves.

The next step in this operational logic is that startup occurs with both hot and warm pipes heated to 561 K (550°F), at which point fluid is discharged from the ESS. When the molten salt enters the cold receiver, it cools off rapidly owing to the thermal capacity of the receiver elements. Calculations (reported in Section 3.6) show that for a 274-m (900-ft) long row, the hot salt would not traverse the entire receiver length without freezing if the receiver's initial condition was ambient temperature. Clearly, this is as unsatisfactory as would be the thermal shocking of the receiver pipe caused by such a startup strategy. Therefore, it is necessary to heat the receiver pipe both to keep it from being thermally shocked and to ensure that it can be readily filled with warm salt without freezing it. It was thought undesirable to provide trace heating of the receivers because of added costs, although the cost increment has not been assessed. The philosophy has been to place the empty receiver in focus so that it

may be solar heated. A calculation of this startup transient, described in Section 3.6, was performed and showed that such a startup was feasible. However, the calculation also showed that during the initial traverse of the warm salt, the receiver had to be brought out of and into focus twice to prevent overheating of the receiver by keeping it within ± 56 K (100°F) of the salt temperature. The necessary defocussing leads to a loss of solar energy and to delays in plant startup which can only be accentuated if the collector row length is made longer. Thus, the selection of 274 m (900 ft) as the collector row length seems to be a reasonable compromise between the above conflicting requirements.

It is emphasized that the purpose of the electrical trace heating is not to heat cold fluid in the pipes but to temperate the pipes before warm [561 K (550°F)] fluid is passed through them. Thus, the power utilized by the trace heating is a modest fraction of that produced by the solar electric power plant. At the same time, the electric trace heating has the capacity to melt solidified salt in the pipes, because it is unacceptable to depend on drainage being successfully implemented 100% of the time.

If the receiver were inadvertently left full of molten salt which solidified, this would not be catastrophic. Information in the literature (Ref. 3.5-2) indicates that the salt does not expand significantly upon solidification, so it does not pose a problem of pipe cracking. However, salt solidification in the receiver duct could impose a difficult mechanical requirement upon any expansion bellows between receiver sections. The postulated failure mechanism is that as the receiver cools, the salt solidifies in the bellows and tends to restrain them from further expanding. This is one reason why a decision was made not to take up expansion along the receiver duct with in-line bellows but to accommodate it by an expansion box at each end. Thus, the philosophy is taken that occasions will arise where molten salt is inadvertently left in the receiver to solidify, that it does not harm the receiver, and that its occurrence is infrequent. The pertinence of the latter statement is that it is unnecessary to install electric trace heating in the receiver to prevent such an occurrence.

When the receiver contains solid salt, it is intended to use concentrated insolation to thaw out the system. Since the whole length of the receiver is fully illuminated only at noon, trace heating of the first 4.57 m (15 ft) of each receiver end is used to complete the thawing of salt in the receiver pipe.

While the technical approach described above appears to meet all the requirements, it may present operational difficulties and thus needs to be assessed with experimentation. It is appropriate to consider alternatives should the experimentation indicate unforeseen difficulties. These alternatives include the following:

1. Further shortening of the row length.
2. Changing from draw salt to Hitec, which has a lower melting point [416 K (288°F) instead of 494 K (429°F)]. [General Atomic has verified that there is an industrial plant which has successfully used Hitec at 867 (1100°F) for over a year.*]
3. Placing the receiver in periods of low insolation in a parking position on the concentrator which is thermally well insulated such that the heat losses accompanying continued but slow circulation are tolerable and become acceptable due to the improved operational simplicity.

*The possibility that the salt composition may be modulated by varying the cover gas composition also merits investigation. Draw salt was chosen over Hitec because Hitec is believed to have greater chemical instability at high temperature and because of the higher cost of Hitec. The latter is a consequence of bulk sodium nitrite (in Hitec) being substantially more expensive than bulk sodium nitrate. However, the operational advantages offered by the lower melting point of Hitec suggest that the decision to use draw salt could be re-examined.

Some of the factors in the preceding discussion have motivated various mechanical design decisions. One such decision is the use of a detachable, stainless-steel-clad copper absorber plate in the receiver. This plate carries the selectively absorptive coating needed to ensure good system performance. Should the coating be destroyed by any malfunction accompanying startup or shutdown, it can be readily replaced. Another decision is to accommodate the receiver expansion entirely at the ends of the collector rows. This is a further factor that limits row length and necessitates the use of steel rather than aluminum in the receiver structural material because steel has a lower thermal coefficient of expansion. In addition, the expansion box at the receiver ends is cheaper than in-line bellows, permits simplification of receiver arm attachments, and allows higher optical efficiencies to be achieved owing to the elimination of gaps between otherwise separate receiver segments.

3.5.2 Initial Plant Startup

The operational aspects of initial startup of the plant have been considered briefly. The major questions are how to temperate components in the system before passing hot or warm draw salt through them and how to charge the salt into the system. These questions are not believed to have a significant cost impact on the plant and could be deferred until a subsequent (preliminary design) phase. However, some tentative approaches are discussed here and in Section 4.4* to indicate that there are enough alternative methods to accomplish these objectives.

*There are some differences between the means suggested here and the methods proposed in Section 4.4. It is too early to conclude which is best. The Section 4.4 method is the basis utilized for costing. However, the concepts discussed here are reported for completeness and to provide a basis for future investigations.

The attemperation of the piping and valves is accomplished by electric trace heating, which must be present to prevent salt freezing and blockage in the pipes. The attemperation of the heat receiver is accomplished by solar heat except for the end 4.57 m (15 ft) and flexible downcomers, which are electrically trace heated. This leaves the thermal storage tanks, the drainage sumps, and the steam generator to be addressed. It seems uneconomical to trace heat the storage tanks, because their thermal loss rate is so low that once heated and filled with hot/warm draw salt, they should never cool down below the salt freezing point during the lifetime of the plant including extended maintenance periods. Thus, the trace heating would be used only once in the plant lifetime. A possible approach is to heat the tanks with low-pressure superheated steam from an auxiliary boiler and then to drain out the condensate. Another approach, typically used in the industry, is to pass a solution of water and draw salt throughout the entire system including the tanks, which gradually heats the salt solution so that the water of solution and crystallization is gradually driven off with a consequent reduction in fluid vapor pressure and increase in its melting point. Such equipment is sold by American Hydrotherm Company and may possibly be available on a rental basis as well. The solution approach may also be employed to attemperate the sumps and the steam generator. The latter may also be attemperated by passing through it steam from an auxiliary boiler whose steam conditions are raised slowly so as to avoid excessive temperature transients in the steam generator elements.

The initial draw salt charging of the plant may utilize either the solution method described above or molten draw salt if the components have previously been attemperated. There is even the possibility of using molten Hitec to secure an initial partial filling should this be shown to have operational advantages. In any event, it seems appropriate in the initial filling to utilize the small heater in the purification loop to maintain the salt temperature at desired levels. The draw salt may be melted by introducing the cold solid material into either a sump or an auxiliary melt tank which in turn contains heating coils through which steam or some other suitable fluid passes. A question arises here of whether to perform this operation in a continuous or batch manner; the latter would facilitate the control of

the core gas composition, which can affect impurity buildup in the molten salt. This question has not yet been investigated.

3.5.3 Collector Field Maintenance

A brief study of the operating and maintenance (O/M) costs of the collector field has been performed by GA.

Table 3.5-1 is a list of identified collector maintenance items for the concentrator and receiver subsystems, their maintenance frequency, and the proposed correction scheme. Most of these items are combined into a maintenance protocol that makes most efficient use of the maintenance crew as they move across the collector field. This maintenance protocol, consisting of a frequency and maintenance direction for both scheduled and unscheduled events, is presented in Section 5.3.3, since it forms the basis for collector field maintenance labor and material costs.

The O/M costs presented in Appendix II are only tentative since there is no experience base for them. However, the exercise to develop the O/M cost estimate is useful since it identifies components of concern (the receiver and its drive train require about 80% of the total collector maintenance), shows that corrective maintenance labor of 4420 hr/yr is only ~25% of the total of 19,270 hr/yr, and forms a basis for initial plant availability estimates.

The mirror cleaning costs depend very strongly on the cleaning frequency necessary, which in turn depends on the acceptable level of mirror reflectivity degradation, on the dustiness of the site as well as atmospheric chemical and particulate content, and on the frequency of rain sufficient to wash the mirrors. These items are quite site specific, so generalizations on mirror cleaning requirements are not easy to make. It is evident that the FMSC is less favorable than steerable mirror systems in taking advantage of rainfall to clean the mirror surface because the water drains

TABLE 3.5-1
CONCENTRATOR AND RECEIVER SUBSYSTEM MAINTENANCE

Maintenance Item	Frequency of Maintenance	Correction Scheme
Mirror modules		
Cracking	1 month	Replace
Settling - visual check	1 month	Realign
Settling - instrumented check	1 yr	Realign
HRA mounting - wear or failures	1 month (initially more frequent)	Repair or replace worn or failed parts
Alignment		Realign
Expansion joints		Replace
Leaks		Seal
Cover glass cracking		Replace
HRA support structure	1 month (initially more frequent)	
General condition - corrosion		Clean and paint or replace
External damage		Straighten, dress, repair
Visual inspection of bolt-ups, etc.		Tighten, replace
Visual inspection of geometry and alignment		Realign
Instrumented (laser) check of geometry and alignment	1 yr	Realign
Drive shafts and pulleys		
Mountings	1 month	Secure
Freedom to expand		Free
Universal connectors		Secure
Pulley alignment		Realign
Pulley belt condition		Replace
Pulley and fittings security (keys coming out)		Secure
Bearings	1 yr	Grease
Drive motors		
Electrical connections	1 month	Secure
Mounting		Secure
Drive shaft couplings		Secure
Jacks		
Grease joints	1 month	Clean and lubricate
Visual alignment check	1 month	Realign
Instrument alignment check	1 yr	Realign, check worm gear, replace if worn
Remove screwjack covers, inspect, clean, and lubricate	6 months	Lubricate

TABLE 3.5-1 (Continued)

Maintenance Item	Frequency of Maintenance	Correction Scheme
Piping		
Joint leaks	1 month	Seal
Expansion joint mounting	1 month	Secure
Cracks	1 month	Replace
Pump		
Seal leakage	1 month	Seal
Pipe connection	1 month	Secure
Performance check	1 yr	Replace
Flex hose		
External damage	1 month	Repair
Cracks	1 month	Weld or replace
Joint leaks	1 month	Tighten
Valves		
Packing glands	1 month	Tighten or replace
Pipe connections	1 month	Seal
Seating	2 yr	Remove and test

off much more slowly and removes accumulated grime less vigorously. (Note that the FMSC collector rows are graded so that there is axial drainage, unlike the array at Sandia Laboratories.)

Studies of mirror reflectivity loss with time reported in Ref. 3.5-3 were examined. However, in view of the uncertainty, a mirror cleaning frequency of once per month was assumed. A primary concentrator cleaning rate of 0.05 m/s (10 ft/min) per man was estimated. This estimate is based on manual cleaning in which detergent and water are sprayed onto the mirrors and allowed to drain. No mechanical brushing is employed except on a very infrequent basis which has not been assessed.

The normal cleaning approach then involves spraying the mirrors with high-pressure jets of detergent/water mix followed by a demineralized water spray rinse. A suitable configuration has pressurized water faucets at each 91 m (300 ft) of the 274-m (900-ft) collector row length and 45.7-m (150-ft) long hoses stowed on drums mounted on carts. The detergents are introduced by venturi injection. This manual system has the advantage that it can be used to clean optical surfaces in the heat receiver as well.

The above mirror cleaning approach is different from that previously stated by GA in which a mobile mechanical scrubber and cleaner traveled down the mirror along the continuous FMSC panels. Such an approach has been precluded by the present design, which has thin concrete panels to achieve desired cost and performance goals that are not designed to support a motorized scrubber. Also, the concentrator panel design employs a vertically mounted receiver support post to achieve fabrication/installation economies which prevents the free traffic for a vehicular cleaner.

The resulting estimate of time to clean the mirrors and receiver is 18,598 hr, or about the same as the total maintenance labor time.

3.5.4 FMSC Resistance to Hail Damage

On the night of August 9, 1978, a hailstorm occurred at the Midtemperature Solar Systems Test Facility (MSSTF) at Sandia Laboratories in Albuquerque, New Mexico, and resulted in damage to the mirror surface of the FMSC array. During the midterm review under the LFSCPS program, GA was asked to assess the vulnerability of the FMSC to hailstorm damage and to recommend changes to avoid such damage in future systems.

The subject hailstorm was the most severe observed at the MSSTF since its inception in 1975 and involved hailstones varying from about 12.7 to 19.1 mm (0.50 to 0.75 in.) in diameter driven by winds up to 15.2 m/s (34 mph). Reference 3.5-4 describes the storm and its effects on components at the facility. The following is quoted from that reference:

"There were three thicknesses of glass reflectors in the MSSTF -- 3 mm (0.125 in.), 2.3 mm (0.090 in.) and 1.5 mm (0.060 in.). With one exception, only the 1.5 mm glass was damaged." "The 260-m² General Atomic FMSC lost up to 5% of the mirror surface." "The glass mirrors were Corning 0317 fusion glass 1.5 mm thick." "The damage was concentrated at or near the tangent slat (center facet). The facets in this area underwent the most damage during construction and, therefore, provide the poorest support to the glass. A close examination of the cracked areas showed that there was indeed poor support for the glass at these points."

In order to interpret the hail damage to the FMSC and develop a basis for recommendations on preventing such damage to future FMSC installations, GA conducted screening tests to obtain qualitative information on the relevant variables influencing mirror breakage from simulated hail impact. These tests were a part of a company-funded development program. The qualitative variables considered in the tests included mirror strip thickness, unsupported length, and types of adhesive.

Qualitative conclusions resulting from the study of mirror breakage from simulated hail impact are:

1. Mirror strips well bonded on a hard, smooth, flat surface are very resistant to hail damage regardless of the adhesive used.
2. The impact resistance of well-mounted mirror strips is a weak function of mirror thickness.
3. Mirror thickness is an important parameter in hail resistance when mirrors are not solidly bonded to the concrete substrate.
4. Mirror strips mounted on certain elastic foundations (e.g., 3M 4262 adhesive on concrete) offer more resistance to hail damage than mirror strips mounted inelastically.

Recommendations for preventing hail damage to future FMSC installations are:

1. Use thicker glass mirrors, e.g., 2.29 to 3.18 mm (90 to 125 mils) without a requirement for solid bonding to the concrete substrate.

OR

2. Solidly bond the thinner mirrors to a smooth, undamaged concrete substrate. The use of an elastic bonding medium will further increase impact resistance.

3.5.5 Effect of Wind Loads

The main effect of wind on the FMSC system is to increase the effective static load on the receiver and its support structure due to the lift and drag induced on the receiver structure. The wind is not expected to impair system operability up to 13.4 to 17.9 m/s (30 to 40 mph) unless resonant excitation of some system elements occurs.

Dynamic loads due to structural vibration resulting from vortex shedding as the wind blows across the structure could occur if the system is not properly designed. An estimate of the increased static loads and resulting deflections of the receiver structure and its associated support system can be readily calculated assuming unity lift and drag coefficients. The actual lift and drag coefficients would be determined by experiment. Further study, experimental testing, and verification are required before the effect of vortex shedding can be determined, since it is very difficult to calculate shedding frequencies accurately, particularly in this case where the structure can be oriented in any direction with respect to the free stream and where there are several degrees of freedom and possible modes involved.

Once the fundamental mode frequencies of the structure are calculated and the shedding frequencies are experimentally determined, the problem becomes one of adjusting the design to detune the structure so that the modal and shedding frequencies do not match in any regime where a significant amount of energy can be deposited in the structure. This can be accomplished by adjusting the structure stiffness, providing spoilers, or adding point masses at selected locations to shift certain modal frequencies and avoid the problem area.

Deflections due to stagnation pressure increase the deflection of the receiver aperture 8% to 12% over the maximum normal static condition of 6.35 mm (1/4 in.) when the velocity is 13.4 m/s (30 mph). This results in a very minor loss in collection efficiency. The wind loading is too small at 13.4 m/s (30 mph) to appreciably increase the torque on the receiver control motor.

Significant collection losses should not be encountered until the wind velocity reaches approximately 26.8 m/s (60 mph), at which point the wind-induced deflection of the receiver structure could increase 50% over the maximum static values. At 40.2 m/s (90 mph), the static deflections could increase an additional 100% of the maximum static values, doubling the stresses in the structure.

Assuming that the dynamic effects resulting from the wind can be dealt with effectively as discussed above, then the plant should only show a minor loss in efficiency at 13.4 to 17.9 m/s (30 to 40 mph) and should be able to survive in 40.2 m/s (90-mph) conditions without structural failure.

REFERENCES

- 3.5-1. Schuster, J. R., et al., "Line Focus Solar Central Power Systems, Phase 1. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.
- 3.5-2. Fried, J. R., "Heat Transfer Agents for High Temperature Systems," Chem. Eng. 80 (12), 89 (1973).
- 3.5-3. Freeze, J. M., "Effects of Outdoor Exposure in the Solar Reflectance Properties of Silvered Glass Mirrors," DOE Report SAND-78-1649, Sandia Laboratories, September 20, 1978.
- 3.5-4. Harrison, Thomas D., "Midtemperature Solar Systems Test Facility Results: Effects of Severe Hailstorm on August 9, 1978," DOE Report SAND-78-2182, Sandia Laboratories, 1978.

3.6 SYSTEM DYNAMIC BEHAVIOR

3.6.1 Control Philosophy

The primary control philosophy adopted for the LFSCPS control is to normally collect all "collectable" energy and to deliver that energy to the hot or the warm storage tank as dictated by the return (hot tank) temperature. The EPGS is operated at demanded load (except for startup/shutdown transients) and overcollection and undercollection are accommodated by the ESS. The control objective function is to maintain collection system design outlet temperature for the maximum time consistent with startup and shutdown transient requirements. Normally the load-storage strategy would be managed to meet the user requirements within the "collect maximum energy" strategy. In the event of a dispatch condition which would result in collection of less than the maximum energy, the plant operators will start the plant or initiate conditions to isolate an appropriate portion of the collection field.

With the ESS acting as a buffer to heat collection versus heat utilization rates, the collector and electrical generator subsystems may operate in a semiautonomous manner except for certain startup and shutdown timing sequences. Operation of the steam generator from a "constant temperature" fast response "firing" system (pumping from the hot tank) leads to a mixed mode control of the plant with the more stable "turbine follow boiler" mode dominant. Load response should be rapid and stable, and the (short-term) fast full range response will provide a flexible operating plant. Load following could easily be provided within the constraint matching the desired total average load/storage relation over the total collection period.

To implement this philosophy, the individual collectors are controlled to bring their outlet temperatures to values consistent with design temperature delivery at the hot tank. Each local process control (PC) unit

(controlling seven or eight rows) has a demand based on the total half-field outlet (hot tank) temperature plus an offset which adjusts the inlet valve of each row to adjust local flow to achieve the desired temperature. The total return temperature is used to adjust the half-field pump flow to meet the design temperature. The "adjusted" bias of the local temperature control provides nearly the required row temperatures at all rows to result in the design total outlet temperature. The integral control on the total outlet temperature, with proportional control only on individual rows, allows sufficient offset to achieve the desired system level temperature adjustment while retaining local temperature control. The long time delays (transport delay) of the system and the mix of many parallel outputs prevent exclusive use of system temperature for local control. On the other hand, variable collection rates and heat losses preclude achieving the desired system outlet temperature by exclusive use of local temperatures.

Local response and protection provided by the collector system PC units, under a supervisory control mode, provide rapid responses in an autonomous fashion, effectively decoupling the collector controls in terms of rapid response. The collector PC units effectively follow four rules:

1. Avoid receiver overtemperature so as to prevent degradation of the selective coating.
2. Avoid fluid overcooling so as to prevent coolant solidification.
3. Maximize operating time at the demanded fluid outlet temperature.
4. Minimize thermal transients to the receiver.

Except for encountering conditions that preclude meeting these rules, acceptance of supervisory input of mode and setpoint data and transmission

of intermittent routine data are the only communications required between the collector PC units and the central unit. Abnormal conditions are alarmed and related data transmitted (exception data) at higher rates.

3.6.2 Design Transient Events

The distributed digital control approach effectively decouples the collector system controls from the rest of the plant. Since the transient performance of the EPGS and the ESS is generally well known, the LFSCPS transient studies have concentrated on the collector system.

Collector system transients are divisible into five groups: normal operation, startup, shutdown, temporary cloud cover, and emergency responses. Only the first four groups can have a design impact. The primary concerns were control system instability caused by the significant fluid transport time delays and possible overheating or overcooling of the receiver. Table 3.6-1 summarizes the objectives and results of the analytical evaluations of these transients.

The following sections describe the CNTRL computer code used in the collector system transient evaluation, the startup and shutdown transients, and one response to a severe cloud cover event.

3.6.3 Dynamic Analysis Model

The CNTRL computer code is designed to follow "slugs" of fluid from the warm tank, through the appropriate piping into the collector, and back to the hot tank. Because the piping transients on either end are not design limiting, the CNTRL model was reduced to a single collector with a given salt inlet temperature. With the control system developed for the LFSCPS, CNTRL can automatically pass from startup into normal control, or from normal operation to shutdown, or through various other permutations. CNTRL also has a single cloud cover model that causes a variable length and velocity cloud to traverse the collector from end to end in either direction.

TABLE 3.6-1
LFSCPS TRANSIENT ANALYTICAL EVALUATION

1. Event:	Normal operation.
Objectives:	Stable operation at design temperature with the normal daily solar variation.
Result:	Feasible.
2. Event:	Startup from a cold, purged condition.
Objectives:	Establish feasibility for anticipated row lengths, evaluate alternate startup methods, and determine thermal transient rates.
Results:	Several startup methods are feasible. The preferred startup method preheats the empty receiver to near inlet temperature before beginning flow.
3. Event:	Shutdown from near design point by receiver system drainage from hot toward cold end.
Objectives:	Establish feasibility for anticipated row lengths and determine thermal transient rates.
Results:	Drainage consistent with a 0.1% grade is feasible. Smaller grades were not investigated but may be possible if local grade variations are not too severe. Normal shutdown commences when the receiver temperature rise becomes too small.
4. Event:	Transient cloud cover.
Objectives:	Establish an acceptable control system response.
Results:	Acceptable response is feasible in all cases. Considerable response optimization should be undertaken. Cloud cover can have so many possible variations that any simple control response tends to degrade outlet temperature more than necessary. Extended cloud cover leads to automatic shutdown.

CNTRL has four major subroutines. The PIPE subroutine for insulated pipes was not used in this study. The HRA subroutine, for heat receiver assemblies, contains a fully implicit simplified model of the collector. The model is nearly identical to the "original receiver performance model" of Figure 2.2-7. The important differences are the implicit temperature solution achievable by mirror simplifications in the radiative heat transfer model and reformulation as a set of unsteady (time-varying) equations. The third subroutine is the control system subroutine. The fourth major subroutine is an earlier version of the SUNPOW mirror model.

The CNTRL model was specifically formulated for systems with substantial transport lags. The collector is divided axially into as many as 70 axial "nodes." For each integration, the control system determines the desired flow based on prior conditions. The new flow, new insolation, new mirror losses, and new cloud cover are combined by the unsteady-state heat transfer equations into new fluid, glass, insulation, and receiver pipe temperatures. At the end of the step, each slug is advanced axially downstream by one node. Therefore, the collector outlet temperature cannot be affected by changes in the inlet temperature until the inlet fluid travels the length of the receiver.

The expected, and observed, effect of the transportation lag is reduced control system stability. Several stable controls were developed. The model normally uses the preferred control system, but switches can be set to enable certain interesting controls that have been discarded.

The control system involves substantial logic. Thus, distributed digital control is well suited to this controller. The major functional groups are defined in Figure 3.6-1. Five measured temperatures and the measured insolation are used to set a flow demand. Local flow control around the pump is used to ensure agreement between measured flow and demand flow.

Fig. 3.6-1. Collector control system functional groups

The five measured temperatures are the fluid inlet and exit temperatures and three receiver pipe temperatures at roughly 109 m (300 ft), 183 m (600 ft), and 262 m (860 ft) from the inlet. The pipe temperatures are used for fast proportional flow control. The fluid exit temperature is used in a slow reset flow control. The fluid temperature is expected to vary linearly from inlet to exit. Therefore, measured deviations from the expected linear profile cause the controls to attempt linearization.

Severe events may lead to a "bang-bang" defocus control. (Partial defocus may be acceptable pending analysis of an asymmetrically heated receiver assembly.) Except for startup, while the empty receiver is being filled, and severe cloud cover, the defocus command has not been initiated by any transient. Therefore, the control system will collect nearly all available solar energy.

The control system is discussed as part of the following discussions of transient events.

3.6.4 Startup and Shutdown Sequences

The startup and shutdown sequences have the greatest likelihood of leading to a frozen salt slug in the receiver.

Startup. The first question regarding startup is whether the warm salt [~ 561 K ($\sim 550^\circ\text{F}$)] can be pumped into an ambient temperature collector without forming a frozen slug [~ 506 K ($\sim 450^\circ\text{F}$)]. The answer is no. Even at design maximum flow, the salt will solidify within the first 30.5 m (100 ft). Of course, even if solidification did not occur, the receiver pipe would see step changes in temperature from ambient to ~ 533 K ($\sim 500^\circ\text{F}$). Reverse fill from the hot [839 K (1050°F)] end toward the warm end was not considered.

The second question regarding startup is whether the salt can be pumped into the receiver some other way without refreezing. The answer is yes. The empty receiver can be preheated in sunlight until the pipe temperature measures 533 K (500°F). Salt flow is then begun. Normally, the far end of the receiver will heat so fast that it could exceed its design limit of ~ 867 K ($\sim 1100^\circ\text{F}$). Therefore, the measured pipe temperatures are fed into a "bang-bang" receiver defocus control that keeps the receiver between 533 and 617 K (500° and 650°F).

When the fluid reaches the exit for the first time, the axial temperature profile is highly nonlinear. Therefore, the startup controller advances to a new mode that conditions the collector with a specified volume of flow, typically twice the receiver pipe volume. After the conditioning mode, a modified "normal control" begins. The fluid outlet temperature is controlled to follow an exponential-rise-to-a-limit demand temperature curve. A 0.19 K/s (20°F/min) time constant was found to be the maximum rise rate achievable under control in the morning on winter equinox. The same rate understates the possible rise time at more favorable times of the year.

The complete startup sequence is illustrated in Figures 3.6-2 and 3.6-3. Note that the controlled rise in demand temperature is necessary to prevent overshoot on the design receiver pipe temperature caused by transport lag effects.

Shutdown. The first question regarding shutdown is whether warm salt can drain without plugging for reasonable grades. The answer is yes for a 0.1% grade and a 0.366-m (1.2-ft) head. Therefore, drainage from either end is acceptable at normal operating conditions.

The second question regarding shutdown is what the preferred method of shutdown should be. The preferred method would deliver the most collectable energy to the ESS. That leads to type types of shutdown: (1) shutdown by operator or master control system (MCS) command, and (2) shutdown due to insufficient insolation (such as "Day is Done . . . Sun is Gone . . .").

Fig. 3.6-2. Startup transient flows beginning at 7 a.m., December 21

Fig. 3.6-3. Startup transient salt temperatures beginning at 7 a.m., December 21

Both types of shutdown are illustrated in Figures 3.6-4 and 3.6-5. Once again, note that the controlled rise in reduction temperature is necessary to prevent undershoot of the exit fluid temperature that could lead to salt solidification.

3.6.5 Mode Changing Sequences

As shown in Figure 3.6-1, a significant portion of the control system is devoted to mode changing logic. The startup controller begins by heating an empty collector, followed by rapid collector fill with "bang-bang" defocus, then collector thermal conditioning, then rise to demanded temperatures, and finally normal control. The shutdown controller, depending on its initiator, begins in normal control. As the sun sets, the outlet temperature drops along a controlled path, eventually dropping below a level leading to receiver defocus and drainage. These mode changes are automatic.

A substantially more complex path could follow various cloud cover events. Figures 3.6-6 and 3.6-7 show the "unknowledgeable" and "knowledgeable" control system responses to a 16.1-km (10-mile) wide opaque cloud traveling from inlet to exit at 20 m/sec (393.7 ft/min). The cloud width (of 13 min) was selected to just miss causing shutdown in the "unknowledgeable" case. The difference between "unknowledgeable" and "knowledgeable" response is that the latter knows by insolation measurement that a cloud was present, whereas the former assumes that the cloud misses the sun sensor.

Note that the cloud response logic drops the demand temperature with the measured temperature and then follows the same control logic used in startup to raise the temperature back to design point.

Clearly, an extra minute or two in the "unknowledgeable" case would initiate automatic shutdown. If the sun were then to reappear, should drainage

Fig. 3,6-4. Fluid temperatures following operator-initiated defocus at 10 min

Fig. 3.6-5. Pipe temperatures as sun drops over horizon starting from 2 p.m., December 21

Fig. 3.6-6. Control system response to a 20 m/sec cloud, 10 miles (or 13 min) wide, traveling inlet to exit, unseen by insolation detector

Fig. 3.6-7. Control system response to a 20 m/sec cloud, 10 miles (or 13 min) wide, traveling inlet to exit, seen by insolation detector

be stopped and startup commenced? Should drainage be prevented by extra circulation of warm fluid to keep the outlet above the drainage setpoint? Should reverse flow from the hot to the warm end be used to keep the receiver at near design temperatures, leading to a faster return to design temperatures? All of these modes are feasible, but each has its own drawbacks. The current approach may drain too often. The other approaches waste energy, and there is no sure way of knowing when the attempt to stay at temperature is futile.

The startup sequence relies on continued sunlight while the receiver is being filled. Appropriate logic for sunlight interruption has not been developed.

In summary, the mode-changing logic used in this study is adequate. Further consideration should be given to more sophisticated logic tailored to the issues raised above in an effort to raise the total energy collection and also raise the quality (temperature) of that energy.

4. CONCEPTUAL DESIGN

4.1 INTRODUCTION

4.1.1 System Description and Requirements

This section describes the overall system design, pertinent system requirements, and the corresponding requirements imposed on the various subsystems. A site description is also presented. The overall system requirements, as requested by DOE, are listed in Appendix I. Table 4.1-1 gives the resulting overall parameters of the LFSCPS.

The key top level requirements are that the LFSCPS, after startup, be capable of operating directly from the collector field (i.e., without drawing from energy storage) at 100 MW(e) net for at least 4 hr on the day of worst heat collection and that on the best day it fully charge an energy storage capacity equivalent to 420 MW(e) of net power generation while simultaneously producing 100 MW(e) net. The latter requirements were also employed in the first generation design of the central receiver solar electric power system.

Clear day insolation levels at Barstow, also given in Appendix I, were used in sizing the collector field. It is important to note that the average insolation levels at that site are about 85% of the clear day values; consequently, the design is somewhat optimistic. In the marketing analysis of Section 7.3, this optimism has been partly corrected by use of an average availability factor of 80%.

The site was assumed to be generally flat desert land containing modest brush additions which must be removed. Some grading of the land is required for roads, parking, and buildings. The resulting total cost of the site

TABLE 4.1-1
KEY PARAMETERS FOR 100-MW(e) LFSCPS

Net design output	100.5 MW(e)
Mirror surface area	$9.66 \times 10^5 \text{ m}^2$ ($10.4 \times 10^6 \text{ ft}^2$)
Heat transport/storage fluid	Draw salt
Receiver fluid inlet/outlet temperatures	565/843 K (557°F/1057°F)
Collector type	FMSC modules
Module size	5.49 m long/3.61 m wide (18 ft/11 ft 10 in.)
Geometric concentration factor	71
Receiver surface properties	0.94 absorptivity 0.30 emissivity
Receiver length	36 ft
Module arrangement	976 rows 274 m (900 ft) each at 7.62-m (25-ft) pitch
Land required	232 hectares (573 acres)
Energy storage capacity	420 MW(e)
Plant capacity factor	45.6%
Turbine type	Tandem-compound, 3600-rpm reheat unit
Turbine throttle steam conditions	16.5 MPa/811 K (2400 psig/1000°F)
Number of feedwater stages	7
Net EPGS efficiency	39.4%
Net collector field efficiency (including piping losses)	62.0% at equinox noon 32.7% year average
Net plant efficiency	23.8% at equinox noon 12.9% year average
Cost estimate for first plant	\$287.0 million
BBCOE for first plant at 45.6% capacity	151 mills/kW-hr

preparation is \$25,204 per hectare (\$10,200 per acre), which is considerably more than the raw land cost of \$2,199/hectare (\$890/acre), but is consistent with values predicted by Sandia-Livermore.

Figure 4.4-3 illustrates the overall plot plan of the LFSCPS and Figure 4.4-2 provides details of the piping layout. The collector field sizing was based on selecting a row length of 274 m (900 ft), use of the concentrator panel dimensions selected from the parametric studies of Section 2, and a row pitch of 7.62 m (25 ft) selected to keep panel-to-panel shadowing at an acceptably low level. The row pitch is uniform because this permits vehicular access to both sides of the collector row. The resulting collector field size together with allowance for roads and fencing amounts to 232 hectares (573 acres). The collector rows are east-west oriented because for the FMSC this maximizes the heat collection and reduces the speed with which the receiver must move, thus minimizing the receiver drive motor power requirements. Overall site dimensions are 1191 m (3910 ft) wide by 1945 m (6384 ft) deep.

The axial spacing between collector rows which feed a common hot salt manifold has been taken as 4.57 m (15 ft), an amount considered sufficient to permit vehicular access without encroaching on the isolation valves or the flexible downcomers which move up to about 711 mm (28 in.) on each end to accommodate half the thermal expansion of the receiver tube in each row. The spacing between the north and south sectors of the field is determined by the aggregate sum of the energy storage tanks, their surrounding dikes and berms, and a 12.2-m (40-ft) wide asphalted road. The spacing between the east and west sectors of the field is established by the turbine and steam generator buildings together with two 6.1-m (20-ft) wide asphalted roads that permit access to the input ends of all collector rows. Located at the input end are the control/isolation valves, motor drives and their power supplies, digital processors for control of collector row groups, and various instrumentation and control junctions. Thus, the asphalted roads provide well-paved corridors to the "busy" portions of the collector field.

In addition, 6.1-m (20-ft) wide north-south paved roads at each end of the collector field serve a similar purpose.

The EPGS is located at the center of the field, since this reduces the length of the piping runs and hence the cost of the HTS and the HTS pumping power and thermal losses. However, in order to prevent cooling tower drift and blowdown from fouling the mirror surfaces, the cooling tower is located at that edge of the field which is downstream of prevailing winds. In practice, this requirement would need to be examined with respect to actual site topography and the source of makeup cooling water. Buried concrete conduit transports water between the centrally located condenser and the cooling tower.

The administration and personnel access control buildings are located on the edge of the field by the main road through the site that is closest to the nearest main public highway. The site is fenced for plant security and has a perimeter ditch to provide for control of run-off water. All land used for vehicular traffic which is not paved is oiled or otherwise treated to prevent dust from being raised. This includes the northerly and southerly portions of the perimeter road and the roads between collector rows. It is anticipated that these roads will not be used very frequently so the more expensive treatment is not required.

The ESS comprises multiple tanks because they are operationally simpler than thermoclines and are only modestly more costly at worst, and because they permit decentralization of the storage system. The decentralization is illustrated in Figure 4.4-2 and shows the hot storage tanks located at the end of each hot manifold. The warm tanks are located near the hot tanks for the following reasons:

1. To act as a depository for any salt from the hot manifold that is substantially below design temperature,
2. To provide the hot tank with a source of blended salt of varying temperature for conditioning the steam generator.

4.1.2 System Performance

The performance of the LFSCPS is detailed in Section 5 for the days and times requested by DOE. Staircase efficiency data are also given in that section. This section summarizes the performance and provides supplemental information.

The overall year-round efficiency of the plant over the insolation interval is 12.9%. The peak efficiency is 23.8%, which occurs at equinox noon. The total plant output during the year is 4.26×10^8 kW(e)-hr based on the given insolation conditions. This corresponds to a 48.6% capacity factor. The collector field output varies during the day as the insolation level varies. How this output translates into a variation in daily electric output depends on the energy storage management strategy and whether the backup fossil-fired heater is used. If a follow-the-sun mode of operation is utilized, the performance given in Table 4.1-2 for June 21 (as an example) is realized. In the example of Table 4.1-2, the plant begins EPGS operation at the time that the receiver output fluid temperature has reached the normal preset level of 843 K (1057°F). All energy gathered at lesser temperatures is considered lost to the system, although this is not truly the case. Such energy is, in practice, used to heat up the receiver subsystem during the startup, and energy gathered in excess of that amount is transported to the warm storage tanks. The calculations of Table 4.1-2 further assume that no net power is produced until power to drive the auxiliaries is generated. In addition, the energy collected at temperatures below 843 K (1057°F) as the collector field is shut down is treated as lost to the system, although this is not the case in practice. The performance calculations are conservative for the June 21 example of Table 4.1-2. However, they will be much less conservative for December 21, when the time between sunrise and the attainment of the required fluid outlet temperature is a minimum, i.e., only 0.78 hr instead of 2.70 hr. Because the system performance depends so much on establishing a suitable fluid management strategy, which has not yet been defined, performance data for other months analogous to that in Table 4.1-2 have not been generated.

TABLE 4.1-2
LFSCPS DAILY PERFORMANCE FOR JUNE 21

Time (hr)	Net Electric Output [MW(e)]	Comments
4.76	--	Sunrise
7.46	--	843 K (1057°F) fluid outlet temperature attained
7.53	0	
8	30.2	
9	92.1	Fluid commences into storage at 9.18 hr
10	100.1	
11	100.5	
12	100.6	
		Day is symmetric about solar noon

The performance of the LFSCPS also varies throughout the seasons. Figure 4.1-1 illustrates this season variation for the reference FMSC design in which the symmetrically shaped panel is normal to the latitude declination. This is the orientation for maximizing the yearly electric output. However, tilting the panel toward the north permits more energy to be collected during the summer, albeit at a lower yearly average. Such an FMSC panel may be more useful to a summer-peaking utility. Hence, the GA design of the plant is readily adaptable to utilities which have high summer loads or which have extensive cloud coverage in the winter.

Fig. 4.1-1. Seasonal heat collection characteristics of the FMSC

4.2 CONCENTRATOR SUBSYSTEM

4.2.1 Concentrator Subsystem Requirements

The concentrator subsystem reflects incident insolation and focuses it on the moving heat receiver. The subsystem comprises the FMSC primary concentrator panel, its support, and its foundations.

The subsystem is sized to meet the overall energy interception requirements associated with the desired electrical output stated in Section 4.1. The sizing is performed by means of the SS2 code through use of the SUNPOW code as a subroutine.

The primary concentrator performs its function in the following manner. It comprises an array of narrow reflecting strips which are arranged in a semicylinder and oriented in a specific manner as illustrated in Figures 1.3-1 and 1.3-2. Detailed information on the principles of the FMSC is given in Ref. 4.2-1.

The reflecting strips must be carefully positioned and oriented in order to perform their function properly. The requirement that the strip deviate less than 3.18 mm (1/8 in.) vertically along its normal from its reference position and less than 15 min of an arc from its reference slope was adopted. This requirement produces an acceptable minimization of receiver miss losses.

The reflecting strips and their supports must be durable and have long-term resistance to the environment in order to meet the lifetime requirement for the plant of 30 yr. If the long-term resistance is not sufficient for the above purpose, the strips and/or their supports should be of lower cost and be readily replaceable. The primary concentrator is also required to survive wind speeds up to 40.2 m/s (90 mph) and the seismic specification given in Section 4.3.10 of Appendix I.

The preceding requirements have been met in the reference plant design by means of a concrete panel for the primary concentrator to which are glued back-silvered glass mirror strips. Other methods of meeting the above design requirements were explored and are reported in Section 4.2.5.

The width and thickness of the mirror strips were established by optimization studies with the SS2 code. Values of 50.8 mm (2 in.) and 2.29 mm (0.090 in.) were chosen for the mirror strip width and thickness, respectively. The length of the concentrator panel must be equal to the receiver length when multiplied by an integer. Since the receiver length was 11 m (36 ft), a panel length of 5.49 m (18 ft) was adopted in view of the unknown with respect to handling of the longer modules. The concentrator must be supported by means of adjustable supports so that it can be readily lined up upon installation and realigned should any foundation sag occur.

4.2.2 Concentrator Subsystem Design

The primary concentrator physical specifications are given in Table 4.2-1, and the corresponding equipment list upon which the plant cost is based is given in Table 4.2-2.

The collector design used as the reference for the LFSCPS study is shown in cross section in Figure 4.2-1. The main structure of the concentrator is a concrete shell with a nominal thickness of 11.7 cm (4.6 in.). The inner radius of the shell is about 2.44 m (96 in.) and the aperture is 3.6 m (142 in.) wide. The concentrator shell contains 71 facets on its inner surface, each of which is 5.08 cm (2 in.) wide. The center of each facet is located on the tangent circle.

The concrete shell is reinforced with eight 12.7-mm (0.5-in.) diameter steel cables in the longitudinal direction. These cables are pretensioned prior to concrete placement. Transverse reinforcement is provided by bent rebars spaced approximately 30.5 cm (12 in.) apart.

TABLE 4.2-1
CONCENTRATOR SUBSYSTEM SPECIFICATIONS

General

Type	FMSC
Number of mirror facets	71
Mirror reflectivity	0.94
Geometric concentration factor	71

Physical

Concentrator aperture	3.6 m (142 in.)
Concentrator aperture angle	96 deg
Concentrator radius	2.44 m (96 in.)
Panel thickness	117 mm (4.6 in.)
Panel width	4.1 m (160 in.)
Panel length (2 panels/receiver)	5.5 m (18 ft)
Panel tilt	35 deg
Facet mirror width	51 mm (2 in.)
Facet mirror thickness	2.3 mm (0.090 in.)
Facet tolerance (3σ)	± 15 min
Panel material	Prestressed, reinforced concrete
Panel support	Adjustable steel posts
Foundations	52 per collector row
Diameter	0.61 m (2 ft)
Depth	0.61 m (2 ft)

TABLE 4.2-2
CONCENTRATOR SUBSYSTEM EQUIPMENT LIST
Quantities per 274-m (900-ft) Collector Row
(972 Collector Rows)

<u>Item</u>	<u>Quantity</u>
Concentrator panels	50
Panel concrete	135 m ³ (177 yd ³)
Steel fiber reinforcing	8.05 Mg (8.87 tons)
Prestressing strand	2226 m (7301 ft)
Reinforcing bar	803 kg (0.89 ton)
Mirror strip length	1.83 m (6 ft)
Mirror strip number	10,650
Foundation concrete	22.7 m ³ (29.7 yd ³)

Fig. 4.2-1. Concentrator module for LFSCPS

The concrete shells are made by a continuous casting technique in 121-m (396-ft) lengths. A vibrating screed is used to consolidate the concrete after it is placed, as shown in Figure 4.2-2. A steam cure is used to produce a 20.7- to 27.6-MPa (3000- to 4000-psi) concrete within 24 hr.

Following the cure periods, the 121-m (396-ft) long castings are cut into 5.5-m (18-ft) lengths. This is done to facilitate handling the panels, each of which weighs about 6804 kg (15,000 lb).

The mirror strips are 5.08 cm (2 in.) wide, 2.3 mm (0.09 in.) thick, and 1.38 m (54 in.) long. The glass is water-white and is back-silvered after cutting to size. The silvered strip is subsequently sealed on the back side with epoxy.

The mirror strips are placed on the facets of the concrete shell by a machine. A rubber-based adhesive which is sprayed onto the back of the mirror prior to placement is used to hold the mirror strip in place on the facet. Once the mirrors are placed, the shells can be stacked for storage until they are transported to the concentrator erection site.

The molds used to cast the concrete shells are fiberglass-coated concrete. They are made from a perfect male master mold about 7.6 m (25 ft) long. Twelve such casting molds would be required to produce the concentrators for a 100-MW(e) plant in a period of 2 yr.

Each panel is supported at four locations at the ends of the panel. The steel support posts are 12.7 cm (5 in.) in diameter with a 6.4-mm (0.25-in.) thick wall. The support of adjacent interior panels is shared by a single post.

The rear panel support posts are all identical, as shown in Figure 4.2-1. The front posts alternate between a long actuator support column which passes through a slot between panels for that purpose and a short column. Adjustment capability is provided at each post so that the concentrator can be aligned with a minimum amount of field labor.

Fig. 4.2-2. Forms for precast concrete panels

The concentrator support posts are set on cylindrical concrete footings. The footings, which are 61 cm (24 in.) in diameter and 61 cm (24 in.) deep, are poured in holes previously drilled by a power auger. Approximately 50,000 footings are required for the 100-MW(e) LFSCPS.

The parallel rows of concentrators are placed on 7.6-m (25-ft) centers as shown in Figure 4.2-3. This provides sufficient room for a 3.7-m (12-ft) wide roadway between each row for transportation access. All rows are nominally on the same level. A 1:500 slope is provided along the concentrator row to allow the hot salt to drain from the system and also to allow any water collected on the mirror to run off. A drawing of a single collector row is shown in Figure 4.2-4.

4.2.3 Concentrator Design Studies

A number of rather extensive studies (Refs. 4.2-2 through 4.2-4) have shown that concrete should be a cost effective material for an FMSC concentrator panel. General Atomic has employed concrete in previously fabricated panels, and any difficulties encountered with the concentrator subsystem to date are attributed to the method used to cast the concentrators rather than the use of concrete per se.

During the first portion of the LFSCPS design study, time was spent investigating new manufacturing methods which would both reduce the fabrication cost of the concrete concentrator panel and improve its optical accuracy. Some time was also spent looking at alternate concentrator manufacturing concepts which used materials or techniques of construction that had not been investigated previously. The two additional concepts which evolved from this investigation involved the use of a rigid urethane foam structure in one concept and the use of rectangular steel tubes for the mirror strip supports in the other. All three concepts studied are reported in detail in Ref. 4.2-5.

Fig. 4.2-3. Typical spacing of concentrators

Fig. 4.2-4. General arrangement of LFSCPS

Concrete Fabrication. To the present time, concrete primary concentrators have been cast in individual forms. Their design has included a concentrator support or buttress at each end. The process is labor intensive, uses excessive material, and has produced a concentrator of marginal accuracy.

A simplified panel without buttresses and a more accurate and repeatable manufacturing method have been selected. The design, which substantially reduces the panel cost, requires a precasting factory at or near the site. The factory will contain several 121-m (396-ft) long, fiberglass-coated concrete forms cast from a 5.49-m (18-ft) long, precision-made master mold. Pretensioned 121-m (396-ft) panels will be cast as a slab without top forms. The concrete will then be consolidated with a rail-mounted vibrating screed and steam-cured to obtain 20.7- to 27.6-MPa (3000- to 4000-psi) strength in 1 day. The panels will be cut into appropriate lengths and moved to the construction site. The use of steam-curing will allow reuse of each mold every other working day.

This method is flexible enough to accommodate steel forms, forms of shorter length than 121 m (396 ft), and panels of various sizes and configurations. Mirrors can be attached either at the precast yard or after panel installation at the site. The method is practical for 100-MW(e) plants and does not pose any major technical problems. Preliminary direct costs for various size panels sufficient for a 100-MW(e) plant are given in Table 4.2-3. These costs were used in the parametric analyses and in the modeling in the SS2 code. They are less than the final data given in Section 5.

The outline of a typical simplified panel is shown in Figure 4.2-5. This panel has an aperture of 3.05 m (10 ft) with a mirror strip width of 63.5 mm (2.5 in.). The current reference design reported in Section 4.2.2 has a 3.61-m (11.83-ft) aperture and a 50.8-mm (2-in.) mirror strip width; however, construction will be very similar. The legs or buttresses were eliminated both to allow casting the panels as a slab and to facilitate storage and transport. The ends have been modified to ease stacking. Elimination of the buttresses also allows the use of various foundation schemes which will be more cost effective.

TABLE 4.2-3
PARAMETRIC COST ESTIMATES OF CONCENTRATOR PANELS FOR ONE 100-MW(e) PLANT^(a)

	2.13-m (7-ft) Panels				3.05-m (10-ft) Panels			
	Total Cost (\$ million)	\$ Per Panel	Cost Per Unit Area of Mirror		Total Cost (\$ million)	\$ Per Panel	Cost Per Unit Area of Mirror	
			\$/m ²	\$/ft ²			\$/m ²	\$/ft ²
Precast panels								
Personnel	12.0	257	15.39	1.43	10.2	316	13.02	1.21
Materials	5.8	124	7.43	0.69	8.9	275	11.41	1.06
Forms	1.4	30	1.83	0.17	1.2	37	1.51	0.14
Batch plant	0.3	6	0.43	0.04	0.4	12	0.54	0.05
Equipment and buildings	0.5	11	0.65	0.06	0.6	19	0.75	0.07
Subtotal	20.0	428	25.73	2.39	21.3	659	27.23	2.53
Substructure and installation								
Excavation	2.3	49	2.91	0.27	2.0	62	2.58	0.24
Concrete supports	12.3	263	15.72	1.46	11.0	341	14.10	1.31
Transport (under 10 miles)	0.47	10	0.65	0.06	0.65	20	0.86	0.08
Installation	2.3	49	2.91	0.27	2.0	62	2.58	0.24
Subtotal	17.4	371	22.17	2.06	15.7	485	20.13	1.87
Grand total	37.4	799	47.90	4.45	37.0	1144	47.36	4.40
Additional cost for:								
100-mile transport	2.3	50	2.91	0.27	3.2	100	4.09	0.38
300-mile transport	7.0	150	8.93	0.83	9.7	300	12.39	1.15

(a) Assumptions:

2-yr construction time.
46,700 7-ft panels; 32,300 10-ft panels.
Mirror area: 8,400,000 ft².
Productive labor at precast factory

Transportation by 22-ton-capacity trucks at \$2.00 per mile.
Contingencies for major breakdowns, strikes, risk, and profit are not included.

Fig. 4.2-5. Typical concrete primary concentrator

The individual concentrator panels will be installed on the support columns using a mobile crane. Alignment bolts will be provided between the support and the panels to facilitate final alignment. A 12.7-mm (1/2-in.) gap will be provided between panels for thermal expansion.

The design of the panel support structure will depend to some extent on the construction site and type of heat receiver support structure selected. The support for the panels will utilize columns. A power auger will be used to excavate the cylindrical footings for the support columns. The selection of this concept was based in part on the conclusions reached in a foundation cost study by Auld and Lodde (Ref. 4.2-5). While Auld and Lodde's study was for single-axis tracking systems (it was directed specifically to parabolic troughs and did not consider the fixed mirror concept), the conclusions reached are believed valid.

Foamed Plastic Fabrication. One of the alternate methods of fabricating the primary concentrator which looks promising utilizes structural polyurethane foam. Such a material is used as main flooring in mobile homes. The advantages of such a material are speedier fabrication (form turnaround of 1 hr) and factory fabrication and quality control. The main drawback to using a construction of this type has been that the base material exhibits a high thermal coefficient of expansion, usually in the range of $54 \text{ to } 72 \times 10^{-6} \text{ K}^{-1}$ ($30 \text{ to } 40 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1}$). However, GA has identified at least one product which may have the thermal expansion properties suitable for a stable primary concentrator.

The product suggested is Glasspak, which is currently being manufactured by Conolite Division of Woodall, Inc., a subsidiary of Libbey-Owens-Ford Company. Glasspak is a fiberglass-reinforced foam. Polymer Development Laboratories (PDL) has recently obtained the rights to the Glasspak process and is now building a plant in California to make the product.

Glasspak is essentially made by impregnating fiberglass roving mat with a slow-rise (10-min) polyurethane foam material. This "prepeg" is then placed in a warm closed mold, where it is allowed to expand into a rigid form producing a pressure of about 276 kPa (40 psi) on the mold. Table 4.2-4 gives some of the physical properties of the material.

The cost of Glasspak is substantially higher than that of nonreinforced foam [$\$353/\text{m}^3$ ($\$10/\text{ft}^3$) versus $\$208/\text{m}^3$ ($\$5.88/\text{ft}^3$) for the same density material]. Therefore, in order to obtain maximum use of its properties, the following procedure has been suggested:

1. Make a male and female mold set of the FMSC surface.
2. Place the Glasspak prepreg in this mold and allow it to expand against the surface.
3. Remove the female mold, leaving the cured Glasspak on the male mold.
4. Replace the first female mold with a new female mold representing the dimensions of the desired outside contour of the structure. This mold may also contain a precured piece of Glasspak or possibly a thin layer of aluminum or steel for the back of the structure.
5. Place regular urethane foam inside the mold with probes to form a core of about $40.1 \text{ kg}/\text{m}^3$ ($2.5 \text{ lb}/\text{ft}^3$).
6. Allow the core to cure and remove the entire structure from the mold.

The finished panel would consist of a 12.7-mm (0.5-in.) thick fiberglass-reinforced skin on the outer and inner surfaces and a low-density core about 102 mm (4 in.) thick. A finished structure 7.62 m (25 ft) long by 3.05 m (10 ft) wide would weigh about 204 kg (450 lb).

TABLE 4.2-4
PROPERTIES OF GLASSPAK REINFORCED FOAM

	Standard Grade MA-415	Heavy Duty Grade MA-720
Core		
Foam density, kg/m^3 (lb/ft ³)	64.1 (4.0)	112 (7.0)
Glass mat reinforcement, kg/m^2 (oz/ft ²)	0.458 (1.5)	0.610 (2.0)
Combined density, flexscored, kg/m^3 (lb/ft ³)	96.1 (6.0)	135 (8.4)
Composite		
Flexural strength, MPa (psi)	33.1 (4,800)	39.3 (5,700)
Flexural modulus, MPa (psi)	1,186 (172,000)	3,172 (460,000)
Shear strength, kPa (psi)	483 (70)	1,276 (185)
Shear modulus, MPa (psi)	5.65 (820)	17.2 (2,500)

In order to obtain the desired thermal expansion properties, the Glasspak fiber-to-foam ratio may have to be altered. Another possibility is that some or all of the glass fiber could be replaced by carbon fiber in the mat production.

A drawing of a typical foamed plastic FMSC primary concentrator is shown in Figure 4.2-6. The foundations for this system are much lighter than those for a concrete system, since their sizes are dictated by the maximum wind load on the structure and not the weight of the concentrator.

Preliminary cost estimates indicate that a foamed concentrator panel could be manufactured for about $\$21.53/\text{m}^2$ ($\$2/\text{ft}^2$). While the foamed plastic concept appears to offer several possible advantages, including lower cost, a development program would have to be completed to obtain more information on both the physical properties of the glass-stabilized material and its reaction to the environment. This concept therefore could not be selected at this time as the reference for the LFSCPS concentrator design.

Steel Fabrication. There is some prior experience in fabricating FMSC panels from both steel and aluminum. For the most part, these panels were not successful, either because of high cost or because the resulting structure was not sufficiently rigid. A short study was conducted to see if some of these problems could be overcome by improving the design or assembly techniques. The tubes are assembled on a jig to provide the necessary alignment accuracy.

The concept pursued for the steel fabrication design involved using thin-walled square steel tubing to form the concentrator facets. The tubing was 50.8 mm (2 in.) on a side, which corresponded to the 50.8-mm (2-in.) glass strip width.

The proposed assembly method involved holding a number of these tubes (40 or more depending on aperture) between two sturdy steel end plates. The ends of each square tube contained a square steel plug which was turned to form a cylinder at the end. The cylinders were placed in jig-drilled holes

Fig. 4.2-6. Typical foamed plastic primary concentrator

located in the end plates and held in place by a single shoulder bolt. The cross section of such an assembly is shown in Figure 4.2-7.

Single mirror modules could be made by this method to lengths of up to about 6.1 m (20 ft). One or more intermediate supports for the tubes would be required to prevent excessive sag and also to reduce wind-induced vibration of the tubes due to vortex shedding.

Correct alignment of the individual mirrors would be done by assembly against master templates, following which the correctly aligned tubes would be locked in place by the shoulder bolt.

One advantage of this type of construction is that the capital investment for dedicated production facilities would be low. Components could be fabricated in existing facilities, with final assembly and alignment taking place at or near the power plant site in a minimal facility requiring little in the way of high-cost specialized equipment.

At the time the cost comparisons were made, indications were that the cost of producing steel primary concentrators will be somewhat greater than that for the reference concrete concentrator. Also, this method of construction will limit the size of concentrators. For these reasons, the concept was not pursued further.

4.2.4 Collector Deflection Analysis

A finite element analysis of the concrete primary concentrator for the 100-MW(e) LFSCPS subjected to both dead weight and unequal settlement of supports was performed. The purpose of the analysis was to determine whether the selected dimensions of the concentrator panel permitted the angular and positional tolerances to be met. It was found that the maximum surface rotation about the longitudinal axis under dead weight is less than 0.25 deg and thus is within the allowable. The distortion due to a unit

Fig. 4.2-7. Steel primary concentrator assembly

support settlement at one support post provides valuable information for designing a proper foundation so that the facet accuracy of the primary concentrator can be secured.

The particular panel investigated was an 11-m (36-ft) long, 114-mm (4-1/2-in.) thick lightweight concrete shell with longitudinal prestress and embedded steel fibers. The detailed geometry of the concentrator is shown in Figure 4.2-8. The MODSAP computer code was employed as the analytical tool in performing the analysis.

A MODSAP (Ref. 4.2-7) model which consists of 72 plate-shell elements and 91 nodal points was developed for this investigation. Figure 4.2-8 depicts the side view of the mathematical model, while its developed view is shown by Figure 4.2-9. The boundary conditions at the four support posts (nodes 3, 6, 87, and 90) are all simply supported. The two load cases considered here are gravitational load in the Z direction and 25.4-mm (1-in.) vertical support settlement at nodal point 90.

With the presence of prestress and embedded steel fibers, the structure is assumed to be homogeneous, isotropic, and linearly elastic. The following physical properties are used to describe the material model:

Lightweight concrete weight density	= 1938 kg/m ³ (0.07 lb/in. ³).
Young's modulus of elasticity	= 15,859 MPa (2.3 x 10 ⁶ psi).
Poisson's ratio	= 0.3.

The vertical displacement is the most significant component. The vertical displacement shapes are plotted for the gravitational load case in Figure 4.2-10, and the unit settlement case is plotted in Figure 4.2-11 at selected cross sections. The displacements are non-dimensionalized by a maximum displacement d_0 to facilitate the presentation.

Fig. 4.2-8. Side view of mathematical model for primary concentrator

Fig. 4.2-9. Developed view of mathematical model for primary concentrator

Fig. 4.2-10. Vertical displacement plots of primary concentrator due to dead weight

Fig. 4.2-11. Vertical displacement plots of primary concentrator due to 1-in. vertical settlement at node 90

The most significant measurement of the facet distortion of the concentrator is the rotation about the longitudinal (Y) axis. The magnitude and location of the maximum rotations for the two load cases are summarized in Table 4.2-5.

TABLE 4.2-5
MAXIMUM FACET DISTORTION

Load Case	Maximum Rotation (deg)	Location
Dead weight	0.157	Node 49
25.4-mm (1-in.) settlement	2.0	Node 90

The results show that the maximum rotation of the primary concentrator due to dead weight is within the allowable of ± 0.25 deg. To meet this limit, the allowable foundation settlement should be limited to less than 3.18 mm (1/8 in.). While this limit is modest, most settlement occurs during the early portion of plant life and may be accommodated with the adjustable supports. Thus, adjustment at the end of the construction period is expected to ensure that alignment is adequate throughout the plant lifetime.

4.2.5 Concentrator Performance

Concentrator performance for the FMSC cannot effectively be separated from the receiver configuration and performance. The geometric relationships of the concentrator described in Section 2.2 determine the behavior of the system. The relation to the receiver, however, determines the numerical performance achieved by a given concentrator design. Some general aspects of the concentrator performance are therefore discussed here, and the combined concentrator/receiver performance is discussed following the discussion of the receiver detailed performance in Section 4.3.6.

Evaluation of mirror strip aiming tolerance showed that adequate performance could be achieved with a ± 0.5 -deg three sigma normal distribution. Since the expected tolerance for the planned panel concrete fabrication technique is ± 0.25 deg, the satisfactory performance at greater tolerance is reassuring. The ability to quantify the effects of aiming tolerance permits performance/cost trade-offs to set tolerance values.

Mirror strip glass thickness had previously been allocated as a fractional loss. The quantification of the time (angular) dependent glass edge losses indicates that the penalty associated with "thicker" glass, within the 1.52- to 3.18-mm (0.06- to 0.125-in.) range considered, does not significantly impact the overall performance. Thinner glass does reduce losses, but other characteristics such as optical properties, mounting, degradation, impact resistance, and handling are overriding considerations. The effect of the mirror strip glass thickness varies between winter and summer and is particularly significant in early and later hours. This aspect must be considered if system requirements are weighted toward these times.

Changes in included arc of the collector in the range of 92 to 98 deg do not significantly affect the performance. For a given aperture, a greater arc reduces the collector radius and therefore reduces the average solar speed and aiming loss. However, a greater arc results in a smaller apparent receiver aperture for the outermost facets and hence an increased miss loss as well as a greater fraction of energy impinging on the secondary concentrator with attendant losses.

While increased mirror strip width decreases glass edge losses, the wider image results in more miss loss. The strip width is therefore tied to the receiver configuration.

Studies were conducted for mirror strips with slight curvature to reduce focal line width. The results of the study showed a potential increase in overall efficiency of approximately 3.5% to 4.5% for a given receiver/concentrator configuration. Alternatively, the curvature could be used to:

1. Further increase concentration by reduction of the receiver width.
2. Reduce edge loss and decrease fabrication cost by using wider mirror strips.

The uncertainties in cost, handling, etc., of curved strips and their added "novelty" were such that this feature was not adopted for the reference design. It is, however, a potential means of future improvement.

REFERENCES

- 4.2-1 Russell, J. L., Jr., E. P. DePlomb, and R. K. Bansal, "Principles of the Fixed Mirror Solar Concentrator," General Atomic Report GA-A12902 (Rev.), February 1, 1977.
- 4.2-2 Potthoff, R., et al., "Investigation of a Central Station Solar Power Plant, Final Report," ERDA Report GA-A13821, General Atomic Company, February 26, 1979.
- 4.2-3 Russell, J., and R. Potthoff, "Demonstration Model of Solar Power Concentrator - Final Report for Period January 1, 1974 through December 31, 1974," General Atomic Report GA-A13352, May 15, 1975.
- 4.2-4 Potthoff, R., General Atomic Company, unpublished data.
- 4.2-5 Schuster, J. R., et al., "Line Focus Solar Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.
- 4.2-6 Auld, H. E., and P. F. Lodde, "Study of Low Cost Foundation/Anchor Designs for Single Axis Tracking Solar Collector Systems," DOE Report SAND 78-7048, University of New Mexico, January 1979.

4.2-7 Johnson, J. J., "MODSAP, A Modified Version of the Structural Analysis Program SAP IV for the Static and Dynamic Response of Linear and Localized Non-Linear Structures," DOE Report GA-A14006 (Rev.), General Atomic Company, June 1978.

4.3 RECEIVER SUBSYSTEM DESIGN

The receiver subsystem consists of the heat receiver and truss, the receiver support system, the receiver aiming system, the receiver positioning mechanism, and the receiver instrumentation system. The isolation valves (one of which controls flow through the receiver) for each collector row are included in the heat transport subsystem.

4.3.1 Receiver Design Requirements

The heat receiver in an FMSC system must be accurately positioned at the focus of the primary concentrator. For best results the focal line should coincide with the center of the secondary concentrator aperture.

With an FMSC, the line of focus is always located on the imaginary surface described by a cylinder whose sides are parallel to the longitudinal axis of the concentrator with a cross section which is formed by the focal circle which passes through the centerline of each mirror facet as shown in Figure 1.3-2. In addition, the secondary concentrator should remain aimed at the tangent facet at all times.

Deviations from this ideal situation will cause minor losses in energy collection efficiency. The function of the heat receiver subsystem design is to ensure that the desired geometric relationships are maintained at all times within a reasonable set of limits.

For the purposes of the LFSCPS design study, the heat receiver is designed to operate between 561 K (550°F) inlet and 839 K (1050°F) nominal fluid outlet temperatures. The nominal fluid flow rate in a particular heat receiver is a function of the collector area associated with a single heat transfer pass and other variables which are discussed in Section 4.3.6.

The environmental requirements for the entire plant are given in Appendix I. Of particular importance to the heat receiver subsystem are the operational wind velocity spectrum, the 40 m/s (90-mph) wind survivability requirements, and the 0.20- to 0.25-g horizontal and vertical ground acceleration. This latter point is discussed in detail in Section 4.3.4.

The general design requirements for a power plant of the type studied here are covered in applicable portions of Section I and Section VIII of the ASME Pressure Vessel Code.

In order to maintain a high energy collection efficiency consistent with the cost goals of the program, certain design tolerances have been specified for positioning the heat receiver. These are given in Table 4.3-1.

4.3.2 Receiver Reference Design

This section describes the components which were selected as a reference for the LFSCPS study. These components were used as a basis for estimating the plant cost. Table 4.3-2 provides the specification of the reference receiver design, and Table 4.3-3 gives the corresponding equipment list.

Heat Receiver. A cross section of the heat receiver used in the LFSCPS plant is shown in Figure 4.3-1. The heat receiver tube in this design is 3.81 cm (1.5 in.) in diameter with a 0.125-cm (0.049-in.) thick wall. The tube material is carbon steel for the first 12 receiver sections and type 316 stainless steel for the remaining 13 receiver sections. The selection of these particular materials and their expected performance with draw salt are discussed in Section 2.1.6.

The heat receiver tubes of the various receiver sections are welded together without expansion joints. An expansion stop or anchor for the tube is provided between the 16th and 17th receiver sections; this allows an equal thermal expansion in either direction. The total expansion in

TABLE 4.3-1
HEAT RECEIVER DESIGN TOLERANCE

Radial deviation from reference circle	0.64 cm (0.25 in.)
Lateral deviation from focal line	0.32 cm (0.125 in.)
Angular twist of individual heat receiver with respect to tangent facet	0.25 deg
Angular misalignment of receiver row	0.125 deg
Heat receiver positioning tolerance	0.32 cm (0.125 in.)

TABLE 4.3-2
HEAT RECEIVER SUBSYSTEM SPECIFICATIONS

General

Absorbing element	Nickel-clad copper plate mechanically fastened to circular heat transport duct.
Absorbing surface	Pt/Al ₂ O ₃ coating with solar absorptivity of 0.94 and emissivity of 0.3.
Insulation	Incoming concentrated insolation passes through a borosilicate glass cover acting as a convective barrier; rear and sides of receiver insulated with canned calcium silicate.
Secondary concentrator	Functions to narrow incoming insolation and to increase effective absorption; reflecting surface of polished aluminum (reflectivity of 0.88).

Physical

Collector row length	274 m (900 ft)
Receiver length	11.0 m (36 ft)
Receiver width (shell)	107 mm (4.2 in.)
Receiver depth (shell)	135 mm (5.3 in.)
Receiver radius of travel	2.44 m (96.0 in.)
Receiver trussing width	813 mm (32.0 in.)
Receiver trussing depth	704 mm (27.7 in.)
Shell and trussing materials	Carbon steel
Heat transport duct material	Carbon steel < 800°F SS-316 > 800°F

Mechanical

Receiver central sag (maximum)	6.4 mm (0.25 in.)
Receiver twist (maximum)	0.25 deg
Receiver expansion per collector row	142 mm (56 in.)
Expansion accommodation	At ends through flexible downcomers and uprisers.
Receiver pivot	Virtual pivot at inlet to secondary concentrator.
Receiver support	Single post
Actuation means	Partially tracked articulated machine screwjack driving an overcenter link.
Angular range of travel	240 deg
Aiming mechanism (receiver must look at tangent facet)	Link-actuated and cog-belt-driven pulleys.
Drive mechanism	Jacks driven by cog belt from a single shaft for each collector row; driven by end-mounted geared 5-hp electric motor (stowing) or by further geared and clutched 0.5-hp electric motor for control.
Stowing time (covers max. range of travel)	15 min
Survival wind speed	90 mph
Max. operational wind speed	48 to 64 kph (30 to 40 mph) (limited by wind excitation at resonant frequencies).

TABLE 4.3-3
HEAT RECEIVER SUBSYSTEM EQUIPMENT LIST

Quantities per 274-m (900-ft) Collector Row
(972 Collector Rows)

<u>Item</u>	<u>Quantity</u>
Concentrator long support column assembly	26
Concentrator short support column assembly	26
Receiver support post assembly	26
5-hp electric motor	1
0.5-hp clutched electric motor	1
Motor support assembly	1
Shaft/bearing assembly	25
Jack input drive assembly	26
Jack	26
Jack actuator assembly	26
Receiver arm and link	26
Receiver tilt mechanism and link	26
Receiver pivot assembly	26
Receiver shell assembly	25
Receiver trussing assembly	25
Flexible piping unit	2
Receiver expansion assembly	2

Fig. 4.3-1. Receiver truss assembly for LFSCPS

either direction from a nominal 296 K (72°F) to the operating temperature will be about 71 cm (28 in.). From 296 K (72°F) to 256 K (0°F), the maximum contraction will be about 9.1 cm (3.6 in.). These numbers are absolute expansions and are not calculated relative to the receiver shell.

The end receiver sections are designed to allow for the above thermal expansion, as shown in Figure 4.3-2. The reinforced flexible metal hoses at the ends take up the expansion between the heat receiver tube and the remainder of the field piping as well as bending from the receiver tube moving on the focal circle. The interface between the receiver and the field piping is shown in Figure 4.3-3. The concentrated solar energy striking the receiver is transferred to the heat receiver tube through a nickel-clad copper surface mechanically connected to the tube. This surface, shown in Figure 4.3-4, is plated with a Pt/Al₂O₃ solar selective coating.

The mechanical and thermal problems associated with mechanically attaching the heat-absorbing surface shown in Figure 4.3-4 to the heat transfer tube are recognized. This method was selected primarily to permit changing the selective surface without changing the heat transfer tube.

As more information and experience are obtained about the operating characteristics of high-temperature selective surfaces, the design may be changed to one in which the heat absorber is brazed to the heat receiver tube. At that time, the absorber may take the form of two separate wings rather than being at a single element.

The heat receiver tube is supported within the heat receiver structure by steel rails brazed to the side of the heat receiver tube. These rails slide in runners connected to the receiver structure. The rails are the same material as the heat receiver tube, and the runners are fabricated

Fig. 4.3-2. Receiver truss assembly for LFSCPS

Fig. 4.3-3. Piping interface for LFSCPS

4.3-10

NOTE:

1. MATL IS .10 COPPER
WITH .010 NICKEL CLADDING
ON EA. SIDE

Fig. 4.3-4. Surface for transferring solar energy to heat receiver tube

from Nitronic 60.* No lubrication is used between the surfaces. The braze used is NiCuMn 37.**

The previous design, shown in Ref. 4.3-1 and in Figure 4.3-16, had the heat receiver tube mounted on rollers. The design was changed to a friction type joint to provide damping between the tube and the structure. In the previous design, the entire heat receiver tube load would be carried back to the anchor point during a seismic event. In the new design, the load will be distributed to a certain extent over the entire receiver structure. The coefficient of friction reported for the Nitronic 60 couple varies between 0.3 and 0.5 as shown in Table 4.3-4.

A thin stainless steel liner is placed around the heat receiver tube about 1 cm (0.39 in.) from the tube. This liner provides an insulating function because of the stagnant air gap which exists and because the polished surface reflects long-wavelength radiation. In addition, this structure is used to seal in the insulation to prevent absorption of moisture and to prevent formation of insulation dust on the receiver window. The brackets carrying the heat receiver support runners are spot welded to the inside of the receiver at intervals of about 1.2 m (4 ft). The extreme edges of the liner are welded to the outer receiver shell. A partition is welded in each end of each receiver section to maintain a continuous insulation barrier.

The insulation between the receiver liners and the receiver shell is calcium silicate. It is placed in the shell in preformed sections prior to the insertion of the liner. The section joints are designed to eliminate direct radiation and conduction paths between the liner and the shell.

*Nitronic 60 is an Armco alloy (Fe-17%, Cr-9%, Ni-8%, Mn-4%, Si-0.13%, N-0.08%C).

**NiCuMn 37 is a Wesgo alloy.

TABLE 4.3-4
COEFFICIENT OF FRICTION FOR NITRONIC 60 COUPLES

Contact Stress [MPa (psi)]	Coefficient of Friction
0.800 (116)	0.50
5.60 (812)	0.35
14.0 (2,031)	0.38
28.0 (4,062)	0.44
56.0 (8,123)	0.44
112.0 (16,246)	0.44

The secondary concentrator for the heat receiver is in the form of a "V" trough with a half angle of 18.4 deg. The inner surfaces of the trough are lined with 0.3-mm (0.012-in.) thick Kinglux C4/P12 Solar Reflective Sheet* to reflect any incident energy onto the heat-absorbing surface of the heat receiver tube. The reflectivity factor of C4/P12 is 87.6%.

Spring clips are provided on the outside of the "V" trough structure. These clips, which are made of 30-gauge type 304 stainless steel, are designed to support the borosilicate window with a minimum thermal conduction to the structure. The outer edge of each half of the trough is attached to the receiver shell with screw fasteners.

The borosilicate glass receiver window is 0.23 cm (0.090 in.) thick. Each piece is 1.8 m (6 ft) long by 5.8 cm (2.3 in.) wide and is supported by four spring clips. When assembled, the window is located about 0.952 cm (0.375 in.) from the majority of the heat-absorbing surface to minimize heat conduction between the heat-absorbing surface and the window. The purpose of the window is to minimize convection and entrap long-wavelength radiation.

The receiver shell is constructed of thin carbon steel and is formed into a deep channel 10.7 cm (4.2 in.) wide by 12.7 cm (5 in.) deep. Steel was selected for the shell because its thermal coefficient of expansion is much lower than the most likely alternative material, aluminum. Each 11.0-m (36-ft) long heat receiver section is connected to the adjacent section with a 4.8-mm (0.188-in.) thick splice joint as shown in Figure 4.3-5.

*A product of Sherwood Solar Products Division, Kingston Industries.

Fig. 4.3-5. Receiver splice and support for LFSCPS

The splice joint is designed with two mounting brackets on the side which support two 9.53-mm (0.375-in.) diameter, 32.5-cm (12.8-in.) long rods. The support rods attached to the brackets slide in bushings mounted on the end of the parallel link system, which provides the pivoting capability for the heat receiver.

The sliding joint between the receiver and its support allows the receiver shell to expand as it heats up during operation. In addition, the joint allows the receiver to move relative to the remainder of the system during seismic excursions.

A set of low-rate [35.7 kg/m (2 lb/in.)] springs is provided at each sliding joint. The spring rate was selected to provide sufficient force to center the receiver and overcome the gravity forces caused by the 1:500 slope built into the system to provide fluid drainage. The rate is low enough to keep the receiver fundamental frequency well below the earthquake frequencies.

Heat Receiver Truss. The heat receivers designed prior to the LFSCPS study had a maximum unsupported length of 7.62 m (25 ft). They incorporated a thick aluminum shell as a structural member and derived structural stiffness from the configuration of the shell. The deflections thus obtained were higher than desired in some instances and required some external wire bracing to augment the stiffness of the main structure.

During the preliminary cost analysis of the LFSCPS study, it became evident that the system cost was affected significantly by the unsupported heat receiver length. This is because the cost of a heat receiver actuation system is associated with each heat receiver support in the FMSC concept.

Preliminary investigations of heat receiver length and deflection indicated that the unsupported length could be increased from 7.62 m (25 ft) to 11.0 m (36 ft) either by using a high-specific-stiffness material, such as

a carbon-carbon composite, for the heat receiver shell or by augmenting the receiver structure with a truss system. The truss system was selected for the reference design because of the current high costs of carbon-carbon composites. These costs are decreasing relative to the costs of other materials with time, and therefore this decision should be re-evaluated at a later date.

After studying several alternate truss configurations, a final selection was made to use a triangular cross section truss mounted on the back of the heat receiver structure as shown in Figure 4.3-6. The members are tubular steel to provide as high a stiffness-to-weight ratio as possible at a minimum cost.

Analysis of the truss structure was carried out using the MODSAP code (Ref. 4.3-2), which is a finite element analysis code. This code provides the forces and moments in each member and the deflection and rotation of each nodal point. Three loading conditions were used in the analysis to provide information on the stresses and the deflection of the truss: (1) when the support was vertical, (2) when it was lying on its side, and (3) when it was at 45 deg to the vertical.

The three apex members which run the entire length of the truss structure are 12.7 mm (0.50 in.) in diameter. These members also serve as conduits for control and instrumentation wiring. The remaining members are 8.9 mm (0.375 in.) in diameter.

A system consisting of stamped sheet metal is used to make the joints between the members. The sheet metal half shells are attached to the associated members using epoxy backed up by a metal fastener. This construction is used to minimize truss assembly cost.

The maximum receiver deflections and twist were calculated for each loading condition. It should be noted that the average deflection and twist are only half the values given in Table 4.3-5.

3/8 OD X .022 WALL
STL TUBE
TYP ALL TUBES EXCEPT APEX TUBE

BRACING IN PLANE OF TOP OF
RECEIVER (SHOWN DOTTED)

1/2 O.D X .028 WALL STL
TUBE-APEX TUBE ONLY

DETAIL A
DETAIL B

ELEVATION OF RECEIVER & TRUSS - INTERMEDIATE
1/8" = 1"

Fig. 4.3-6. Receiver truss assembly for LFSCPS

TABLE 4.3-5
RECEIVER TRUSS MAXIMUM DEFLECTION AND TWIST ^(a)

Load Direction	Z Axis Deflection [mm (in.)]	Y Axis Deflection [mm (in.)]	X Axis Twist (deg)
Vertical	2.6 (0.103)	0.15 (0.006)	0.07
Horizontal	0.15 (0.006)	4.0 (0.159)	0.10
45 deg	1.7 (0.068)	2.5 (0.100)	0.38

(a) Axis orientation as shown below:

The maximum stress in an apex member occurs at the center of the truss with a stress of 36.1 MPa (5230 psi). The highest stress in a diagonal member is about 7.93 MPa (1150 psi). The strength of the members is set more by buckling criteria than by stress. In the design shown, all members are safely below the buckling limit.

The structure presented here is believed to be adequate for the application. It is felt that some additional optimization of the structure could be made which would reduce the number of members and therefore the cost.

Heat Receiver Support and Actuation. Several alternate methods of supporting and positioning the heat receiver were investigated during the first part of the study. These alternates are reported in detail in Section 4.3.3. A preliminary cost estimate was made for each system for comparative purposes. The system selected as the reference design incorporates the single post support, articulated link drive, parallel link receiver support, and pulley and rod concept for aiming the receiver at the tangent facet of the FMSC.

One of the primary reasons for the selection of this particular concept over the other possible alternatives is that this entire support system can be assembled and adjusted in the factory. The assembled system can then be shipped to the field and erected with a minimum of field labor. The anticipated lower factory labor rates more than compensate for the somewhat higher material and manufacturing costs associated with certain portions of this particular system as compared with some systems studied which showed potentially lower material and construction costs.

This particular combination of support, positioning, and aiming meets all the design improvement goals established in the beginning of the study. These include:

1. Simplified installation and alignment.
2. Uninterrupted 240-deg receiver travel.

3. Clear mirror surface between supports.
4. Continuous heat receiver aperture.
5. Low total cost.

The major components which make up the heat receiver support and positioning system reference design are discussed below. The designs were used as the basis for the final system cost estimate.

Main support column. The main support column, shown in Figure 4.3-7, consists of a single 127-mm (5-in.) schedule 40 pipe. These columns are placed at 11.0-m (36-ft) intervals. The column has a rectangular plate welded to one end, which is attached to a cylindrical concrete footing by means of four 12.7-mm (0.50-in.) diameter studs cast into the footing. Adjustment nuts are provided between the base and the studs for rapid alignment of the structure.

A bracket welded to the column above the base plate supports the end of the jackscrew, the drive shaft which supplies the jackscrew power, and the cog belt system which transfers the power between the two units (see Figure 4.3-8). A second rectangular plate located above the jackscrew bracket is used to support one corner of the concrete shell which forms the primary concentrator for the FMSC. Bolts embedded in the concrete concentrator forms pass through holes in the support plate and allow rapid alignment of the concentrator using adjustment nuts.

A double track is welded on one side of the support column. This track is used to guide the four rollers attached to the jackscrew during the first portion of the jackscrew travel.

The end of the support column terminates in two channel members which are welded to the cylindrical portion of the column. These channels support the bolt which passes through the end of the receiver support arm to form the radial arm pivot point. The small cog belt pulley, which is used in the heat receiver aiming system, is fixed to one of the support channels

Fig. 4.3-7. Long concentrator support post for LFSCPS

Fig. 4.3-8. Intermediate concentrator post for LFSCPS

When properly aligned, the receiver support arm pivot point is coincident with the tangent circle of the primary concentrator.

Articulated link positioning device. The simple jackscrew linkage, like those used previously on FMSC systems, can only position the heat receiver over an angular range of about 140 deg without physically changing the pivot points. A range of up to 240 deg is desired for the LFSCPS, both to allow efficient collection of solar energy throughout the year and to provide a safe stowing position for the heat receiver assembly during high-wind conditions.

In the assembled articulated link system shown in Figure 4.3-9, the jackscrew shown in Figure 4.3-10 is connected to the radial heat receiver support arm by a 660-mm (26-in.) long link pivoted at each end. The jackscrew extension tube extends from the traveling nut to one end of the link. The other end of the link is attached to the receiver support arm.

During the first portion of jackscrew travel while the receiver support arm is traversing about 175 deg of rotation, the extension tube on the jackscrew is restrained to move in a straight line by four roller bearings being guided in a rail system. A hook-like extension is welded to the pivoting end of the receiver support arm. After the receiver support arm has rotated through about 175 deg of travel, the hook engages the connecting pin between the end of the jackscrew and the link. This allows the radial arm to continue its movement through the desired 240 deg of travel. As the hook engages the pin, the four rollers disengage the track. This allows the jackscrew to pivot about 3 deg about its lower support point.

The jackscrew used is of the traveling nut type. A machine thread was selected rather than a ball nut as had been used previously both to decrease cost and to prevent the nut from "running down" in the event of a failure in the input power train. A 4536-kg (5-ton) jackscrew was selected since this is the nearest standard size available.

Fig. 4.3-9. Support column and screwjack assembly for LFSCPS

Fig. 4.3-10. Screwjack-link assembly for LFSCPS

Figure 4.3-11 is a plot of the angular position of the receiver support arm as a function of the jackscrew lift. It also shows the relationship between the rate of change of the receiver support arm position and the jackscrew lift. It can be seen that the positioning is reasonably uniform and that for most of the control range (0 to 210 deg), the rate of change of position is also uniform, with the worst condition occurring at the 0-deg position, which corresponds to mid-summer, either early in the morning or late in the afternoon, when the rate reaches 0.315 deg/mm (8 deg/in.). The maximum rate of change in position occurs when the heat receiver is approaching its stow position. At this time the movement rate is about 0.472 deg/mm (12 deg/in.) of jackscrew lift.

Receiver support system. The proper radial position of the heat receiver is maintained by the receiver support system. This system consists of the receiver support arm, which is attached to the main support arm at one end, and the heat receiver mounting system on the other end.

The basic structure of the receiver support arm is 76.2 x 102 x 4.76 mm (3 x 4 x 0.188 in.) rectangular steel tube approximately 2.44 m (8 ft) long. Bushings are provided at the main support column end to allow the receiver support arm to move in an arc about the support column. The hook device which supports the articulated link during the last portion of support arm travel is welded to the support arm at a point near the pivot point.

The main support column end of the receiver support arm also carries the cog wheel and cog belt tension device used in the heat receiver aiming system. The primary attachment point for the articulated link positioning system is welded on the side of the receiver support arm at a point 457 mm (18 in.) from the pivot point. The outer end of the receiver support arm terminates in four brackets which provide the pivot locations for the heat receiver support system.

Fig. 4.3-11. Radial arm angular position and rate of change as a function of jackscrew travel

The method selected for supporting the heat receiver on the end of the receiver support arm uses the parallel link system. In this system, a virtual pivot point is created at the center of the receiver aperture by a system of parallel links as shown in Figure 4.3-12.

The main advantage of the parallel link system is that it allows the use of an uninterrupted heat receiver aperture over the entire 274-m (900-ft) receiver length. In addition, with the receiver pivot point and the focal arc coincident, the optical efficiency of the FMSC is maximized.

The members which form the parallel link system of support are each formed from two pieces of stamped sheet steel. The two pieces are spot welded together to form a member with cross section similar to an I-beam. This configuration provides the stiffness required at a minimum of weight and cost.

Each of the pivot points of the articulated link support system is bushed with a self-lubricated bearing to provide a low-friction, low-wear bearing surface. The other running surface at the bearing points is provided by 9.53-mm (0.375-in.) diameter bolts.

There are seven members in the parallel link system, three attached to the receiver support arm and four attached to the receiver. The members attached to the receiver support arm are arranged with two members on the side closest to the receiver and a single member on the side away from the receiver. The four members attached to the receiver are all identical and are arranged in pairs.

The heat receiver end of the parallel link system terminates in bushings. These are connected to two slide rods mounted on the side of the heat receiver. This slide joint allows differential thermal expansion to occur between the receiver and its support system. The length of the slide rods is sized to allow relative movement between the receiver and the supports during earthquake. More details on this aspect of the design are given in Section 4.3.4.

4.3-29

Fig. 4.3-12. Receiver support arm and tilt mechanism for LFSCPS

Receiver aiming system. The FMSC system requires that the heat receiver be aimed at the tangent facet at all times to maximize performance. The system previously used to accomplish this was a slider rod which was attached to the primary concentrator at the midspan on the tangent facet. The attachment point was allowed to pivot in the direction normal to the longitudinal axis of the primary concentrator. The other end of the rod was allowed to slide through a bushing mounted on the side of the receiver. The rod was designed with an offset equal to the half-width of the receiver so that the receiver aperture always pointed at the tangent facet.

This approach was abandoned for several reasons. First, the increased radius of the primary concentrator required a much longer guide rod than the one previously used. Second, the change of the receiver pivot point from near the receiver center of gravity to a point which maximized the optical efficiency increased the bending moment applied to the guide rod. Third, the incorporation of a truss structure on the top of the receiver increases the rod bending moment further because it shifts the center of gravity further away from the pivot point.

In view of the above considerations, a guide rod required for the new design would have had to be about 63.5 mm (2-1/2 in.) in diameter to limit the slope angle to an acceptable value. The weight of the rod became a significant factor and would have increased the deflection of the heat receiver when the receiver was in the summer position. This would have necessitated a stiffer truss system to meet the design criteria.

After studying several alternate approaches to the aiming problem, the system shown in Figure 4.3-13 was selected. This system uses a pulley and cog belt system located near the pivot end of the radial arm, which provides the proper 2:1 angular relationship between the receiver and the radial arm, and a single rod connected between the large pulley and a lever which is part of the parallel link receiver support system.

Fig. 4.3-13. Receiver support arm and tilt mechanism for LFSCPS

The small pulley is fixed to the top of the support column side. The larger pulley, which is mounted on the radial arm, is allowed to rotate. The two pulleys are connected by a cog belt so that the larger pulley rotates as the radial arm moves about its pivot point on the end of the support column.

A lever arm rigidly attached to the large pulley by a shaft is pinned to one end of an adjustable tubular rod with a clevis pin. The other end of the tubular rod, which is 25.4 mm (1 in.) in diameter, is attached on the other end to a lever on the heat receiver support mechanism with another clevis pin.

Movement of the receiver support arm about its pivot will thus cause the members of the heat receiver support system to move in such a way that the heat receiver aperture always remains aimed at the FMSC tangent facet. One major advantage of this system is that it can be assembled and adjusted in the shop and should require a minimum of field adjustment.

Receiver drive system. The receivers are arranged in 274-m (900-ft) units. Each of these units contains 26 jackscrew devices to position the heat receiver over the primary concentrator. These 26 jackscrews are driven from a single 274-m (900-ft) long drive shaft.

The drive motor system for positioning the heat receiver was selected based on two main requirements. The first requirement is that the heat receiver must be positioned within ± 0.075 deg during normal operation. The second is the requirement for rapidly stowing the heat receiver in a high-wind condition.

The minimum rate of heat receiver position change during normal operation is dictated by the maximum rate of change in the focal line during the day.

This can be obtained by looking at the typical diurnal change in focal line movement during December and June as shown below:

Time	Angle (deg)		Δ Angle (deg)	
	Dec.	June	Dec.	June
7 a.m.	118	119	36	37
8 a.m.	82	82	19	19
9 a.m.	63	63	10	10
10 a.m.	53	53	5	5
11 a.m.	48	49	1	1
12 noon	47	47		
			Σ 71	Σ 72

Thus, the maximum movement averaged over 1 hr is 37 deg/hr or 0.62 deg/min or 0.01 deg/sec. Over most of the day, the required movement is much less.

The angular movement as a function of jackscrew height is shown in Figure 4.3-11. The jackscrew selected has an input/output ratio of 0.63 turn/mm (16 turns/in.). During the normal control portion of the receiver travel, the rate of angular change varies from 0.118 to 0.315 deg/mm (3 to 8 deg/in.) of jackscrew travel.

At the minimum angular movement of 0.118 deg/mm (3 deg/in.) of jackscrew travel, the minimum input shaft speed required to track the focal line would be 3.3 rpm. If one assumes a minimum nominal drive motor running time of about 2 sec to make the 0.075-deg error correction, the shaft speed required is about 4.5 rpm. The actual speed selected was between these two values at 3.8 rpm, which provides a maximum drive motor running time of about 48 sec and a minimum drive motor running time of 2.36 sec. The actual maximum input power to the jackscrew drives required under normal steady-state control conditions is about 0.142 hp for each row.

The stowing maximum time required during high-wind conditions was selected as 15 min after separate discussions with representatives of the U.S. Weather Bureau, Sandia Laboratories, DOE, and the Martin Marietta Corporation. This represented something of a compromise among the several opinions obtained, but was consistent with the time used to stow heliostats under similar conditions.

A 240-deg rotation of the heat receiver is required to move the heat receiver from the extreme summer position to the stow position in 15 sec. The average shaft speed required to accomplish this is 55.4 rpm. The nominal shaft speed selected is 58.3 rpm to allow for acceleration and deceleration. The maximum nominal input shaft power required is 2.17 hp for each row.

Consideration was given to several types and configurations of motors, including electric, hydraulic, and pneumatic motors. A schematic of the system selected is shown in Figure 4.3-14. In this system a 1/2-hp, 1725-rpm a.c. motor with a 15:1 gear reducer drives the larger stowing motor shaft through an electrically actuated clutch. The a.c. stowing motor, which is rated at 5 hp and 1750 rpm, is connected through a 30:1 gear reducer to the 274-m (900-ft) long input drive shaft.

During normal control operations, the clutch is engaged and the 1/2-hp motor drives the larger motor at 115 rpm and the drive shaft at 3.83 rpm. During stowing, the clutch is disengaged and no power is supplied to the control motor. The 5-hp motor is used for power, turning the drive shaft at 58.3 rpm.

The total system power required to drive the heat receivers in the control mode for the entire 100-MW(e) plant varies through the day and over the year. The power required is a maximum in December and June and is near zero in March and September. An estimate of the power required during a day in December or June is shown in Table 4.3-6. It is felt that this estimate is on the conservative side.

TYPICAL ANGULAR VELOCITIES

0.019°/S MINIMUM IN CONTROL MODE

0.291°/S MINIMUM IN STOWING MODE

Fig. 4.3-14. LFSCPS HRA drive train (diagrammatic)

TABLE 4.3-6
ESTIMATE OF TOTAL CONTROL DRIVE MOTOR POWER CONSUMED IN
DECEMBER OR JUNE ON THE LFSCPS

Time of Day	HRA Angular Position (deg)	Δ Angle (deg)	Horsepower Averaged Over the Hour
7:00 a.m.	119		
8:00 a.m.	82	37	267 (0.2% of 100 MW)
9:00 a.m.	63	19	137
10:00 a.m.	53	10	72
11:00 a.m.	49	4	29
12:00 noon	47	2	14
1:00 p.m.	49	2	14
2:00 p.m.	53	4	29
3:00 p.m.	53	10	72
4:00 p.m.	82	19	137
5:00 p.m.	119	37	267

The jackscrews used in the articulated link drive assembly are attached to a jackscrew support bracket as shown in Figure 4.3-15. The support bracket is pivoted from an attachment point on the main support column. The jackscrew support bracket also carries a stub shaft mounted in bearings located in the end opposite the jackscrew attachment point. The stub shaft, which is 25.4 mm (1 in.) in diameter and about 305 mm (12 in.) long, carries a cog pulley at one end. The end of the stub shaft contains collars which are in turn bolted to the adjacent drive shaft sections. Each collar is pinned to the stub shaft through a slot, which allows the collar and adjacent drive shaft to move about 9.53 mm (3/8 in.) to prevent the buildup of excessive thermal expansion loads within the drive shafts.

The stub shaft cog pulley is connected to a similar pulley located on the jackscrew input shaft by a steel-reinforced rubber cog belt. An adjustable belt tensioning device is also located on the jackscrew support bracket.

Twenty-five of the jackscrew support brackets in a single 274-m (900-ft) row are identical. The support bracket used at one end of the row is similar but carries an additional bracket for mounting the drive motor system.

During the last 55 deg of heat receiver movement, the jackscrew rotates approximately 3 deg about the support bracket pivot point. At this time all the components attached to the bracket also move, maintaining a fixed relative position to each other. The stub shaft, and therefore the end of the drive shaft, moves approximately 19.1 mm (3/4 in.).

There are 25 drive shaft sections in a single 274-m (900-ft) long collector row. The drive shafts are 50.8 mm (2 in.) in diameter and about 10.7 m (35 ft) long. A support bearing is provided at the midspan to prevent excess shaft deflection and shaft oscillations. The bearing support is rigidly attached to the front intermediate support column for the primary concentrator as shown in Figure 4.3-15. Since the end points of the drive shaft move with the jackscrew and its midpoint is rigidly supported, the

Fig. 4.3-15. Driveshaft and motor installation for LFSCPS

drive shaft must deflect 19.1 mm (3/4 in.) during the total heat receiver travel. The forces and stresses involved in this deflection are all very small. The shaft bending stress is minimized through the use of a flexible coupling at each end of the shaft, which permits some shaft slope misalignment without creating a bending moment in the end of the shafts.

Since the torque is applied at one end of the drive line, there will be some shaft windup. Under the worst normal loads this can amount to approximately 180 deg from end to end, which corresponds to a difference in jack-screw height of about 0.813 mm (0.032 in.), or about 0.096 deg in heat receiver angular misalignment from end to end.

4.3.3 Receiver Subsystem Studies

The first part of the LFSCPS study was directed to the investigation of designs which could reduce the cost of hardware for a large-scale plant. In addition, the design of some components which had been built prior to this study needed to be improved to provide the better operational flexibility required of a large plant. This section presents the results of some of those studies. Much of the information reported here was reported in more detail in Ref. 4.3-1.

Heat Receiver Studies. The heat receivers built for the FMSC installation at Sandia Laboratories, Albuquerque, operate at lower pressures and temperatures than those required for the LFSCPS. In the earlier design, the heat receiver tube (HRT) was of rectangular cross section to facilitate matching heat transfer and fluid flow characteristics with the available heat flux. The concentrated solar energy absorbing surface in this system was one side of the rectangular HRT.

The system pumping requirements of the LFSCPS lead to substantial hot salt pressure levels of 1.38 to 2.07 MPa (200 to 300 psi), which would necessitate thick duct walls if a duct of rectangular cross section were retained. Therefore, a circular duct is necessary for the HRT.

To facilitate maintenance of the selective absorber coatings, a design concept has been chosen in which the primary energy-absorbing surface is separate from the heat transfer tube surface. With this concept only the absorbing surface, which could be replaceable, would require a selective coating. This design concept also allows the use of longer weld-free heat transfer tube sections, since their length would not be restricted by the limited capabilities of surface coating or plating facilities as is currently the case.

Several alternative designs for the heat receiver were studied in the first part of the program. These designs all incorporated a round heat transfer tube, and most incorporated a separate heat-absorbing surface. Both single and multiple heat transfer tubes in a single receiver were considered. The heat receiver design selected as the initial reference design is shown in Figure 4.3-16.

Receiver Truss Studies. Cost studies of FMSC systems built to date indicated that the plant cost could be reduced significantly if the receiver span could be increased from the 7.62-m (25-ft) length which is currently being used. The reason is that an increased receiver span decreases the number of receiver supports and drive systems required. These items represent a substantial fraction of the plant capital cost.

During the first phase of the study, two approaches were investigated. The first was to incorporate materials with increased specific stiffness, such as carbon-carbon composites, into the receiver structure, and the second was to examine various trussing systems. The carbon-carbon composite systems appeared structurally very favorable. However, the material cost is still very high. As a result, this approach was dropped in favor of a more cost effective system.

In the trussing investigations, cable trussing was investigated and dropped. Various "space frame" type trusses were investigated, including a box truss, a double triangular truss, and a single triangular truss such as that shown in Figure 4.3-17. A variation of this latter design was selected as the reference design and is discussed in Section 4.3.2.

Fig. 4.3-16. Initial reference design heat receiver cross section

Fig. 4.3-17. Receiver trussing

Heat Receiver Support and Positioning Studies. The initial phase of the LFSCPS study considered three alternate heat receiver positioning concepts, three basic main support structures, three receiver aiming devices, and three heat receiver support concepts. The positioning concepts include the cable-drum drive concept shown in Figure 4.3-18, the circular arch drive concept shown in Figure 4.3-19, and the articulated link drive system shown in Figure 4.3-20. The details of each system are discussed in detail in Ref. 4.3-1. As reported earlier, the articulate link drive was selected as the reference design because it appeared to be the lowest-cost method.

The main support structure design configurations investigated were the "A" frame support shown in Figure 4.3-21, the column support shown in Figure 4.3-18, and the circular arch support, which is peculiar to a particular type of drive system, shown in Figure 4.3-19. The "A" frame support was the design previously used as the main support structure of the FMSC. The column support structure was selected as the reference because of a perceived lower total system cost.

The types of heat receiver assembly (HRA) support designs studied were the offset arm shown in Figure 4.3-22, the roller track support shown in Figure 4.3-23, and the parallel link support shown in Figure 4.3-24. The offset arm was a modification of the design used previously in the FMSC array at Sandia Laboratories. The parallel link system was selected for the reference design on the basis of lowest cost and because it appeared to have some advantages with respect to seismic considerations. Both the roller track and the parallel link support systems allow a continuous HRA aperture for the heat receiver because they provide a virtual pivot at the center of the aperture.

The three types of receiver aiming systems studied included the guide rod shown in Figure 4.3-25, the wire and drum shown in Figure 4.3-26, and the pulley and rod arrangement shown in Figure 4.3-27. The guide rod is similar to the type of aiming system previously used in the FMSC array at Sandia Laboratories. It was not selected here because in order to satisfy aiming

4.3-44

Fig. 4.3-18. Receiver cable-drum drive

4.3-45

Fig. 4.3-19. Receiver circular arch drive

Fig. 4.3-20. Receiver articulated link drive

Fig. 4.3-21. Articulated link drive with "A" frame

Fig. 4.3-22. Offset arm HRA support

4.3-49

Fig. 4.3-23. Roller track HRA support

Fig. 4.3-24. Parallel linkage HRA support

Fig. 4.3-25. Guide rod aiming concept

Fig. 4,3-26. Wire and drum aiming concept

Fig. 4.3-27. Pulley and rod concept

accuracy requirements, the rod had to be too large and heavy. This is a result of the large collector and the different location for supporting the heat receiver compared with that used previously. The pulley and rod system was selected as the reference design on the basis of lowest total estimated system cost.

4.3.4 Seismic Analysis

The design of the LFSCPS embodies a field of FMSCs and receiver drive support and actuating mechanisms. The mechanisms enable the receiver to move through a 240-deg arc so as to allow it to be positioned properly throughout the year. Since the seismic response of the mechanism is complicated, the position at which a member in the mechanism experiences its maximum seismic stress needed to be pinpointed. The geometry and the connectivity of the mathematical model change as the receiver support rotates. Therefore, a special analytical procedure had to be designed for solving a problem of this type.

In order to facilitate the axial thermal expansion of the receiver tube, the supports of the tube are slip joints that allow axial displacement with very little resistance. As a result, it can be considered that the tube is effectively isolated from the seismic load, in the longitudinal direction.

Method of Analysis. A mathematical model for the receiver support and actuating mechanism (RSAM) was developed as shown in Figure 4.3-28. This model represents an interior span of the RSAM system. The model is designed in such a way that the coordinate of nodal points which are moving with the rotation of the arm can vary as a function of θ , the angle between the receiver support arm and the post.

One feature of the RSAM design which enables this mechanism to travel 240 deg is the addition of an over-center link. A hook-like appendage engages a pin (at node 5) when θ is approximately equal to or larger than 175 deg, fixing the link. At the same time, the traveling nut jackscrew rollers (at node 4) disengage from the guide track, allowing the jackscrew to pivot

Fig. 4.3-28. Mathematical model of receiver support mechanism

freely about node 2. This means that the connectivities of elements 9 and 10 in Figure 4.3-28 change location and become the over-center link (dotted) when $\theta \geq 175$ deg.

The cross-section properties and dimensions used in the model are taken from the reference design discussed in Section 4.3.2. Carbon steel is used for all members in the RSAM. Its physical properties are listed below:

Young's modulus of elasticity	=	196.5×10^3 MPa (28.5×10^6 psi)
Poisson's ratio	=	0.3
Mass density	=	7817 kg/m^3 (0.283 lb/in.^3)
Material damping	=	5% of critical

To simplify the mathematical model, the receiver truss was replaced by an equivalent beam. Its equivalent bending stiffness is obtained by equating the maximum displacements at midspan of the actual truss and equivalent beam subject to dead weight.

A special computer code named SROSCO was written to generate the nodal coordinates and proper connectivity of elements 9 and 10 given the angle θ and span length. This code also causes the MODSAP code (Ref. 4.3-2) to solve the dynamic problem accordingly. In this investigation, SROSCO was used repeatedly to calculate the seismic response for every 20-deg angle increment within the full range of 240 deg. The maximum stress in each member was then plotted against θ to locate the critical location for each member.

As specified by LFSCPS requirements, the structure is to be designed to resist the concurrent application of horizontal and vertical base response spectra of the shape given by USAEC Regulatory Guide 1.60 for 5% damping with maximum accelerations of 0.3 and 0.2 g, respectively.

Since the receiver is isolated seismically in the longitudinal direction, the critical horizontal loading condition occurs when the resultant of the two horizontal components is aligned with the X axis at a magnitude of 0.42 g.

The dynamic problem under seismic load was solved using the MODSAP code with an approximate method known as the modal superposition and response spectrum method. The input loading was in a form of acceleration response spectra at 5% damping as specified in Regulatory Guide 1.60. The stress calculated is a close approximation of the peak stress during the ground acceleration time history.

Results. Figures 4.3-29 through 4.3-35 provide a graphical representation of the stress in various RSAM components as a function of the receiver angular position. The results indicate that, with the design modeled, there could be some deformation in the receiver support arm and in the jackscrew area.

A second set of rollers was therefore included in the jackscrew roller support design to minimize the excessive bending moment which caused the high stress in the jackscrew. Time was not available during the study to change the model and rerun the problem.

The receiver support arm has not been redesigned and remains as analyzed. The section modulus of this member will have to be increased during the preliminary design either by using a larger member [76.2 x 127 mm versus 76.2 x 102 mm (3 x 5 in. versus 3 x 4 in.)] or by including gussets at the critical section to locally increase the modulus as required.

All other problem areas appear to be less critical than the two mentioned. Minor changes in design during the preliminary phase should correct all such problem areas.

The discontinuity in stress shown in Figures 4.3-29 through 4.3-35 occurs at the transition point ($\theta = 175$ deg) between two connectivities. This is due in part to the modeling used. However, it does point out that care must be taken in the system design to ensure that the transition is smooth and that shock loading does not occur as the mechanism passes through this transition region.

Fig. 4.3-29. Seismic stress in post

Fig. 4.3-30. Seismic stress in receiver support

Fig. 4.3-31. Seismic stress in jackscrew

Fig. 4.3-32. Seismic stress in link

Fig. 4.3-33. Seismic stress in tilt linkage

Fig. 4.3-34. Seismic stress in over-center link

Fig. 4.3-35. Dynamic amplification factors for receiver truss

4.3.5 Receiver Instrumentation and Electrical Equipment

Instrumentation. The receiver instrumentation per collector row consists of the following:

1. Receiver arm angular position.
2. Four proximity switches on receiver arm travel.
3. Three photocell pairs with the units of each pair mounted on opposite sides of the receiver aperture.
4. Two fluid temperature measurements, one essentially at the center of the row and one at the outlet.
5. Three receiver pipe skin temperatures.
6. Drive system failure detection circuit.

The receiver arm angular position is measured by a precision rotary position transducer. This measurement is used to allow position control for gross positioning and to provide the high-gain loop to overcome friction, stiction, and backlash in the motor gear drive system.

The four proximity switches on the receiver arm position are mounted two at each end of the travel. One of the switches automatically switches the motor drive from fast to slow (if it is on the fast or slow motor) and is located somewhat before the end of travel. The second switch at each end stops all motor drive toward the end as the actual end of travel is reached.

Three photocell pairs are provided to enable "null seeking" to precisely position the receiver in the concentrated light beam. Figure 4.3-36 illustrates the typical shape of the energy of the beam and the relation of the photocell pair to the beam. From the figure it can readily be seen that any motion of the light beam or of the receiver on which the photocells are mounted will result in a significant change in the relative light to each (i.e., one will rapidly decrease and the other will increase). Three pairs are used, and the weighted average value of the three is used for control. This method compensates for potential slight end-to-end position variation of a row and avoids the problem resulting from one or more sensors being shadowed.

Fig. 4.3-36. Photo cell pair and relation to light beam

The two fluid temperature sensors are thermowell-mounted, spring-loaded junction RTDs used for monitoring and control of the fluid temperature. The outlet unit is the row temperature difference control reference and an input to overtemperature defocus. The midpoint unit is used in control prediction and as a monitor of minimum temperature to alarm the control center for drain and isolation.

The three receiver skin temperatures are measured by iron-iron constantan thermocouples welded to the pipe surface. These measurements are used both in control and in the overtemperature defocus control.

Operating experience with the FMSC system installed at Sandia Laboratories, Albuquerque, has shown that it would be very desirable to have a system which would detect a failure in the receiver drive system. A failure in the receiver drive system could cause the receiver to become misaligned to the point that a failure of the heat transfer tube could result, creating a hazardous condition for plant personnel and possibly requiring expensive repair.

The system selected as being the most cost effective consists of a taut, two-conductor wire connecting the center of the back side of each receiver in a row. The wire is electrically isolated from the receiver. One conductor contains resistors as shown schematically in Figure 4.3-37.

A contact point, also shown in Figure 4.3-37, between the two conductors is provided at intervals equal to the receiver length. The contact points are located over the receiver support points. Adjustable stops are provided on the receiver structure so that a predetermined misalignment [6.35 mm (0.25 in.)] between the center of adjacent receivers will create a short circuit between the two conductors, which will cause a change of current in the wire.

The change of current can be used to trigger an alarm in the control room. It can also be used to indicate where the failure or misalignment has

Fig. 4.3-37. Schematic of alarm circuit

occurred. Open circuit due to wire failure can also be used to trigger the alarm to indicate a potential problem and institute operator corrective action.

Electrical Equipment. The single-line drawing for the heat receiver subsystem electrical equipment is shown in Figure 4.3-38. Electrical service must be provided for the 0.37-kW (1/2-hp) shaft and 3.7-kW (5-hp) shaft, 3-phase, 60-Hz tracking and stowing drive motors located at each of the 972 heat receiver rows. Auxiliary power at 120 V single phase is required for operation of the 972 receiver flow control valves. Eight groups of five transformers--each rated at 225 kVA, 4160/480 V, 3-phase, 60 Hz--will provide the required service. Each group of five 225-kVA transformers is sized to support 122 tracking and 122 stowing drives as well as 122 control valves. In order to limit the inrush current, group starting in groups of five motors at a time will be necessary only for the 3.7-kW (5-hp) stowing drives. For reliability, each field of 10 transformers will be supplied by three 4160-V feeders, any two of which are adequate to carry the load.

The heat receiver subsystem electrical equipment list is shown in Table 4.3-7.

4.3.6 Receiver Performance

The receiver performance is determined by the amount of energy entering the receiver aperture and by the energy radiated and convected from the receiver assembly.

Incoming energy is first reduced by the amount absorbed by the secondary concentrator. While this is not a direct loss, since the structure is heated and does re-radiate to the cover glass, the majority of this energy is lost to the air. The second loss of solar energy is by reflection from the cover glass and a portion of the reflection from the receiving surface. The solar reflection losses of direct sunlight amount to about 35% of the total receiver loss of 28.2%, or about 9.9% loss of the incoming light at equinox noon.

TABLE 4.3-7
HEAT RECEIVER SUBSYSTEM ELECTRICAL EQUIPMENT LIST

	Number	Rating
Transformers	40	225 kVA, 4160 V/480 V 3 phase, 60 V
Switchgear		
Disconnect switches	40	4160 V, 60 A, 3 phase, 60 Hz
Disconnect switches	50	4160 V, 60 A, 3 phase, 60 Hz
Motor Control Centers		
MCC	40	480 V
Wire & Cable		
		<u>Length</u>
3 c 250 MCM SKV		15,400 m (50,400 ft)
3 c 350 MCM		600 m (2000 ft)
3 c # 10		43,900 m (144,000 ft)
3 c # 12		43,900 m (144,000 ft)
Conduit		
76 mm (3 in.) rigid conduit		600 m (2000 ft)
19 mm (3/4 in.) rigid conduit		6,700 m (22000 ft)

NOTES:

1) ALL SECONDARY WIRES TO 1/2 HP TRACKING MOTORS AND SOLENOID VALVES TO BE #12 AWG MIN. WIRING TO 5 HP SLOW MOTORS TO BE #10 MINIMUM TO ASSURE MINIMIZING VOLT DROP.

- 5 - SIZE 3, FVWR GROUP STARTERS FOR STOM DRIVES.
25 - SIZE 1, FVWR INDIVIDUAL STARTERS FOR 1/2 HP TRACKING DRIVES:
5 - SIZE 1, FVWR INDIVIDUAL STARTERS FOR 5 HP STOM DRIVES.
1 - RELAY SEQUENCING AND INTERLOCKING PANEL.
1 - 45 KVA, 480 - 208 V/120 V DIST. TRANSF. & CB PANEL WITH:
27 - 1P, 15A, CB
3 - 3P, 15A, CB
3 - 3P, 15A, CB

The thermal spectrum losses occur via three basic paths: convection/radiation by and through the cover glass, conduction through the insulation and back cover, and absorption and re-radiation/convection from the secondary concentrator (in addition to the direct solar absorption noted above). The thermal losses due to radiation and convection are distributed as follows:

Secondary concentrator	15%
Radiation from receiver through glass	4%
Glass (from thermal absorption)	64%
Glass (from direct or reflected solar absorption)	11%
Loss through insulation	6%

Of the energy radiated/convected from the secondary concentrator, only about 14% is from net thermal energy absorbed from the glass or from radiation through the glass by the receiver surface; the remainder is from the directly absorbed solar energy. The loss from the secondary concentrator is about 15% of the total thermal spectrum loss.

Radiation directly from the receiver surface through the glass to either the air or the secondary concentrator is about 4% of the total thermal spectrum loss. Net radiation and conduction/convection to the glass from the collector surface, however, equal 64% of the total thermal spectrum loss. Eleven percent of the heat receiver loss is primarily due to the direct absorption of solar spectrum energy by the cover glass. The remaining loss of 6% is by conduction through the insulation and convection off the back cover.

Figure 4.3-39 presents the calculated heat transfer coefficient across the air gap between the receiving surface and the cover glass. The data are presented for various angles of inclination of the receiver assembly for this "top heated rectangular enclosure." The data presented are from a correlation based on Refs. 4.3-3 and 4.3-4 which has been implemented in the code. Gap distances of 10.2^{+} mm (0.4^{+} in.) have been adapted in the

Fig. 4.3-39. Heat transfer coefficient vs air gap at various inclinations

receiver design. As may be seen from Figure 4.3-39, the minimum feasible air gap heat transfer is being maintained for a variety of receiver positions. The fact that nearly 77% of the net heat transfer across the gap is by radiation is indicative of the minimization of conduction/convection. The large radiation is indicative of the need for a low-emissivity surface. The properties used in the current receiver design pertain to the selective coating under development at Cornell University ($\alpha = 0.94$, $\epsilon = 0.3$). When the properties published by Englehard ($\alpha = 0.91$, $\epsilon = 0.09$, see Section 3.3) are substituted, the equinox noon receiver efficiency increases from 71.8% to 76.3%. At low incident energy levels where the thermal losses stay high (nearly constant at design temperatures), the improvement is even more dramatic.

4.3.7 Concentrator/Receiver Performance

The ray tracing of the concentrator and receiver entry led to a number of design modifications/influences. Some of the more significant results are discussed below.

Analysis of the type of receiver assembly used for the FMSC array at Sandia Laboratories showed that there were significant penalties in locating the pivot point of the receiver 102 mm (4 in.) outside the reference cylinder radius. A 62% reduction in the depth of the secondary concentrator increased the net efficiency approximately 4.5% at solstice times.

Figure 4.3-40 depicts the Sandia project receiver and the truncated secondary concentrator, whose analysis showed the above performance improvements. The original configuration was analyzed with the pivot 102 mm (4 in.) outside the reference cylinder instead of at the reference cylinder. With the pivot at the reference cylinder, the 127-mm (5-in.) secondary concentrator was truncated as shown in Figure 4.3-40 and the aperture was lowered to the reference cylinder. The net efficiency increased by $\sim 2.6\%$ for the "truncated" secondary concentrator with the same ± 0.5 -deg aiming tolerance even

Fig. 4.3-40. Sandia receiver and truncated concentrator

though the "miss losses" increased slightly. The reason for the net increase may be seen from the comparison in Table 4.3-8. The longer secondary concentrator results in much more of the energy entering the aperture striking the secondary concentrator surface and being partially absorbed. The slight increase in miss loss (with larger aiming tolerance) due to the narrower aperture of the truncated design is much more than offset by the decreased absorption by the secondary concentrator. This and other evidence of the disadvantages of large secondary concentration led to selection of an alternate reference receiver design (see Section 4.3.2).

While insufficient design data have been generated to translate any sag and twist of the receiver assembly into "average displacement," provision has been made in the SUNPOW code to assess the effect on performance of such displacement. A fractional loss of 4% of the light to the receiver aperture was assigned to "receiver position loss" to account for control positioning and receiver sag/twist. The positioning should be accurate to 0.6 deg or approximately 2.54 mm (0.1 in.), and the current design criterion for combined sag/twist is 6.35 mm (0.25 in.), giving a worst case or a net average displacement of less than 7.62 mm (0.3 in.). This displacement would cause less than 5% loss in the worst condition (receiver at maximum deflection point of arc and worst case control) and much smaller "average loss."

Table 4.3-9 presents a summary output of the SUNPOW performance/loss train for a reference design parametric analysis made for December 21.* Line 11 is the "Normal Solar Flux" (i.e., normal to the direction of the solar disk). The "Aperture Flux" on line 13 is the flux at the mirror aperture (i.e., after the solar cosine loss of line 15). The "Edge Loss" on line 17 consists of light directly hitting or being reflected into the module structure plus light directly striking or being reflected into the mirror strip glass edges. "Mirror Flux" on line 19 is the resultant flux from all the mirror

*Values are in Btu/hr per square foot of mirror aperture or watts per square meter of aperture as noted.

TABLE 4.3-8
PERFORMANCE COMPARISON OF TRUNCATED SECONDARY CONCENTRATOR AND REFERENCE DESIGN CONCENTRATOR ^(a)

	Miss Loss [kW/m ² (Btu/hr/ft ²)]	Secondary Concentrator Absorption [kW/m ² (Btu/hr/ft ²)]	Heat Out [kW/m ² (Btu/hr/ft ²)]
With Zero Aiming Tolerance			
Ref. CPC	0 (0)	0.225 (71.37)	2.62 (832.82)
Truncated secondary concentrator	0 (0)	0.068 (21.57)	2.75 (874.02)
With ±0.5-deg Tolerance (2 x Expected)			
Ref. CPC	0.0011 (0.35)	0.217 (68.9)	2.63 (834.59)
Truncated secondary concentrator	0.047 (14.90)	0.0878 (27.87)	2.70 (856.05)

(a) Data for June 21; annual changes are approximately one-half those shown for June.

TABLE 4.3-9
SUNPOW PERFORMANCE DATA FOR REFERENCE DESIGN FMSC

DATE 042179

SUNPOW TEST INPUT DATA 4-13-79 REFERENCE

FMSC STEADY STATE PERFORMANCE DATA VS. TIME OF DAY

MONTH AND DAY OF YEAR		DECEMBER 21						
MONTH NUMBER (IMONTH)		12						
MIRROR TILT ANGLE (TILD)- DEGREES		35.000						
ARC OF MIRROR APERTURE (ARD)- DEGREES		95.964						
TANGENT SLAT ROTATION ANGLE (TSD)- DEGREES		.000						
FULL WIDTH OF MIRROR APERTURE (WIDTHM)- FEET		11.833						
MIRROR RADIUS (RAD)- FEET		7.852						
RECEIVER WINDOW APERTURE (WIDTHR)- INCHES		2.100						
TIME POINT NO.		4	5	6	7	8	9	10
1	SOLAR TIME OF DAY- HOURS	7.000	8.000	9.000	10.000	11.000	12.000	.000
2	SOLAR AZIMUTH- DEGREES	117.586	126.579	137.108	149.596	164.111	180.000	.000
11	NORMAL SOLAR FLUX-R/MRSF	.000	204.230	230.130	292.950	305.154	306.644	2167.341
12	(ENSF)- WATTS/SM	.000	643.865	725.519	923.570	962.043	966.741	6832.871
13	APERTURE FLUX- B/HP-SF	.000	93.788	149.391	232.865	270.523	281.426	1680.771
14	(OAPER)- WATTS/SM	.000	295.690	470.977	734.143	852.863	887.238	5298.885
15	COSINE LOSS- B/HP-SF	.000	110.442	80.739	60.085	34.631	25.218	486.570
16	(OCOS)- WATTS/SM	.000	348.184	254.543	189.427	109.179	79.503	1533.986
17	EDGE LOSS- B/HP-SF	.000	26.681	28.821	35.080	35.384	35.147	260.398
18	(OEDGE)- WATTS/SM	.000	84.115	90.862	110.594	111.554	110.806	820.943
19	MIRROR FLUX- B/HP-SF	.000	63.845	114.022	146.673	221.751	232.205	1340.943
20	(OMIPR)- WATTS/SM	.000	201.290	359.473	588.516	699.103	732.063	4227.527
21	SHADOW LOSS- B/HP-SF	.000	3.192	5.701	9.334	11.088	11.610	67.047
22	(OSHADO)- WATTS/SM	.000	10.064	17.974	29.426	34.955	36.603	211.376
23	RCVR MISS LOSS- B/HP-SF	.000	1.558	2.357	2.852	2.624	2.543	19.765
24	(OMISS)- WATTS/SM	.000	4.911	7.430	8.992	8.271	8.016	62.313
25	SEC CON LOSS- B/HP-SF	.000	1.495	2.642	4.205	4.880	5.070	30.060
26	(OSECLS)- WATTS/SM	.000	4.714	8.393	13.258	15.383	15.985	94.769
27	REC POS LOSS- B/HP-SF	.000	2.360	4.234	6.973	8.316	8.717	50.124
28	(ORPOS)- WATTS/SM	.000	7.442	13.347	21.985	26.218	27.481	158.022
29	RCVR OPT. LOSS- B/HP-SF	.000	7.891	14.155	23.341	27.854	29.202	167.792
30	(OROPT)- WATTS/SM	.000	24.876	44.625	73.586	87.814	92.065	528.990
31	RCVR THERM LOSS- B/HP-SF	.000	25.771	25.951	26.001	26.145	26.255	212.319
32	(OTHERM)- WATTS/SM	.000	83.857	81.813	81.972	82.584	82.773	669.369
33	RCVR END LOSS- B/HP-SF	.000	.290	.776	.892	.532	.000	4.588
34	(OEND)- WATTS/SM	.000	.914	2.288	2.811	1.678	.000	14.466
35	HEADER LOSS- B/HP-SF	.000	.681	.681	.681	.681	.681	5.447
36	(OHDR)- WATTS/SM	.000	2.147	2.147	2.147	2.147	2.147	17.173
37	HEAT OUTPUT- B/HP-SF	.000	16.137	56.838	111.062	137.956	146.411	774.260
38	(OOUT)- WATTS/SM	.000	50.876	179.192	350.178	434.928	461.562	2440.973
39	RCVR EFFICIENCY (E1)	.000	.313	.595	.697	.722	.728	.675
40	NORMAL EFFICIENCY (ENFF)	.000	.079	.247	.379	.452	.477	.357
41	APERTURE EFFTC. (APEFF)	.000	.172	.380	.477	.510	.520	.461

strips after the preceding losses and reflectivity loss. "Shadow Loss" (line 21), currently a fractional allocation, is then subtracted from the net energy and proportionally from the fraction of light calculated to miss the receiver (line 23) and impinge on the secondary concentrator (line 25). A fractional loss is currently allocated for "Receiver Position Loss" (line 27). The remaining energy passes to the receiver assembly where it undergoes optical (line 29) and thermal (line 31) losses. Losses from section couplings (not shown), end loss (line 33) from the unilluminated receiver portion, and loss from the header pipe to the field piping (line 35) are calculated. The remaining energy is then calculated as "Heat Output" (line 37). Receiver efficiency (heat out/energy into receiver) is calculated (line 39). The "Normal Efficiency" (line 40), heat-out/normal-flux, and "Aperture Efficiency" (line 41), heat-out/aperture-flux, are calculated. The data enclosed in boxes in the table are significant areas of difference between results for a physical design similar to that used for the FMSC array at Sandia Laboratories and for the parametric analysis reference design.

Analysis was performed of a system utilizing the basic design characteristics of the low-concentration FMSC system (SD) at Sandia Laboratories but with the fluid and temperature conditions of this LFSCPS program. The basic collector and receiver assembly were identical; however, the row length was adjusted to correspond to the revised temperature and fluid. The resulting SUNPOW performance data are presented in Table 4.3-10 in the same form as for the reference design (RD). The heat output may be seen to be 28% less than for the RD. While the receiver miss loss of the SD is essentially zero owing to the wider secondary concentrator aperture, the secondary concentrator losses are more than doubled. Thermal loss is considerably higher particularly at early hours owing to the wider [63.5 mm (2.5 in.) versus 53.3 mm (2.1 in.)] receiving surface and low incoming flux (concentration factor) per unit of receiver surface. The result of this

TABLE 4.3-10
SUNPOW PERFORMANCE DATA FOR LOW-CONCENTRATION FMSC

DATE 042179									
SUNPOW TEST INPUT DATA SANDIA DESIGN PARAMETERS									
FMSC STEADY STATE PERFORMANCE DATA VS. TIME OF DAY									
MONTH AND DAY OF YEAR			DECEMBER 21						
MONTH NUMBER (IMONTH)			12						
MIRROR TILT ANGLE (TILD)- DEGREES			35.000						
ARC OF MIRROR APERTURE (ARD)- DEGREES			90.000						
TANGENT SLAT ROTATION ANGLE (TSD)- DEGREES			.000						
FULL WIDTH OF MIRROR APERTURE (WIDTHM)- FEET			7.167						
MIRROR RADIUS (RAD)- FEET			4.950						
RECEIVER WINDOW APERTURE (WIDTHR)- INCHES			2.500						
TIME POINT NO.			4	5	6	7	8	9	10
1	SOLAR TIME OF DAY- HOURS		7.000	8.000	9.000	10.000	11.000	12.000	.000
2	SOLAR AZIMUTH- DEGREES		117.586	126.579	137.108	149.596	164.111	180.000	.000
11	NORMAL SOLAR FLUX-R/HRSF		.000	204.230	230.130	292.950	305.154	306.644	2167.341
12	(ENSEF)- WATTS/SM		.000	643.865	725.519	923.570	962.043	966.741	6832.871
13	APERTURE FLUX- R/HRSF		.000	93.788	149.391	232.865	270.523	281.426	1680.771
14	(OAPER)- WATTS/SM		.000	295.680	470.977	734.143	852.863	887.238	5298.885
15	COSINE LOSS- R/HRSF		.000	110.442	80.719	60.065	34.631	25.218	486.570
16	(OCOS)- WATTS/SM		.000	348.184	254.543	199.427	109.179	79.503	1533.986
17	EDGE LOSS- R/HRSF		.000	25.298	27.454	33.617	34.052	33.876	249.421
18	(OEDGE)- WATTS/SM		.000	79.754	86.554	105.983	107.355	106.799	786.337
19	MIRROR FLUX- R/HRSF		.000	65.200	115.362	188.115	223.071	233.468	1351.764
20	(OMIRROR)- WATTS/SM		.000	205.552	363.696	593.062	703.265	736.044	4261.642
21	SHADOW LOSS- R/HRSF		.000	3.260	5.768	9.406	11.154	11.673	67.588
22	(OSHADOW)- WATTS/SM		.000	10.278	18.185	29.653	35.163	36.802	213.082
23	RCVR MISS LOSS- R/HRSF		.000	- .000	- .000	- .000	- .000	- .000	- .000
24	(OMISS)- WATTS/SM		.000	- .000	- .000	- .000	- .000	- .000	- .000
25	SEC CON LOSS- R/HRSF		.000	3.501	5.773	9.026	10.426	10.854	64.804
26	(OSECLS)- WATTS/SM		.000	11.038	18.200	28.455	32.869	34.219	204.306
27	REC POS LOSS- R/HRSF		.000	2.478	4.384	7.148	8.477	8.872	51.367
28	(ORPOS)- WATTS/SM		.000	7.811	13.870	22.536	26.724	27.970	161.942
29	RCVR OPT. LOSS- R/HRSF		.000	.000	14.284	23.348	27.727	29.028	159.747
30	(OROPT)- WATTS/SM		.000	.000	45.033	73.610	87.413	91.514	503.626
31	RCVR THERM LOSS- R/HRSF		.000	.000	59.807	51.353	50.658	50.622	374.338
32	(OTHERM)- WATTS/SM		.000	.000	188.550	161.897	159.834	159.595	1180.157
33	RCVR END LOSS- R/HRSF		.000	.000	.297	.712	.449	.000	2.916
34	(OEND)- WATTS/SM		.000	.000	.936	2.244	1.414	.000	9.192
35	HEADER LOSS- R/HRSF		.000	.920	.920	.920	.920	.920	7.359
36	(OHDR)- WATTS/SM		.000	2.900	2.900	2.900	2.900	2.900	23.200
37	HEAT OUTPUT- F/HRSF		.000	.000	21.872	85.325	112.070	120.269	558.803
38	(COUT)- WATTS/SM		.000	.000	68.956	268.909	353.318	379.165	1761.710
39	RCVR EFFICIENCY (E1)		.000	.000	.275	.540	.594	.606	.491
40	NORMAL EFFICIENCY (NEFF)		.000	.000	.095	.291	.367	.392	.258
41	APERTURE EFFTC. (APEFF)		.000	.000	.146	.366	.414	.427	.332

"off hour" penalty is particularly notable at 8 a.m., when the low-concentration (SD) configuration does not collect sufficient energy to attain the design temperature (the code conservatively assumes no energy is credited whenever the energy collected at minimum flow is insufficient to support design temperature.)

REFERENCES

- 4.3-1. Schuster, J. R., et al., "Line Focus Solar Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.
- 4.3-2. Johnson, J. J., "MODSAP, A Modified Version of the Structural Analysis Program SAP IV for the Static and Dynamic Response of Linear and Localized Non-Linear Structures," DOE Report GA-A14006 (Rev.), General Atomic Company, June 1978.
- 4.3-3. Buchberg, H., Ivan Catton, and D. K. Edwards, "Natural Convection in Enclosed Spaces: A Review of Application to Solar Energy Collection," ASME Paper No. 74-WA-HT-12, 1974.
- 4.3-4. Arnold, J. N., Ivan Catton, and D. K. Edwards, "Experimental Investigation of Natural Convection in Inclined Rectangular Regions of Differing Aspect Ratios," ASME Paper No. 75-HT-62, 1975.

4.4 HEAT TRANSPORT SUBSYSTEM

4.4.1 HTS Requirements

The functional requirements of the HTS are to:

- Convey thermal energy from the collector field receivers to the steam generator and the ESS hot storage tanks, when insolation exceeds EPGS requirements
- Convey thermal energy from the collector field receivers and the ESS hot storage tanks to the steam generator, when insolation falls short of or is equal to the EPGS requirements
- Convey thermal energy from the ESS hot storage tanks to the EPGS steam generator, when insolation is unavailable or below the EPGS requirements.

The HTS design requirements which must be addressed in the conceptual design include:

- Selection of satisfactory materials
- Determination of optimum insulation thickness and pipe diameters
- Calculation of piping pressure loss and heat loss
- Provisions for piping support and thermal expansion, heat tracing, draining and venting
- Selection of pumps and valves
- Design of a steam generator.

The HTS conceptual design is based on the operating conditions selected in the Parametric Analysis and reported in Section 2.7.

4.4.2 HTS Design

4.4.2.1 Drawings and Data

Details of the HTS conceptual design resulting from this study are shown in the Figure 4.4-1 HTS/ESS Flow Diagram (BNI Drawing M-1), the Figure 4.4-2 HTS Piping Layout (BNI Drawing M-2) and the Figure 4.4-3 Plot Plan (BNI Drawing M-3).

Primary features of the HTS shown in the flow diagram are the sharing of stainless steel hot manifolds by adjacent receiver tiers, decentralized location of hot and warm salt storage tanks at the base of each hot manifold, location of the salt pumps in the field near the tanks, and the provision of an optional oil-fired salt heater for plant operation during prolonged periods of cloudy weather.

The HTS Equipment List and Data Lists are presented in Tables 4.4-1 and 4.4-2 respectively.

TABLE 4.4-1

HTS EQUIPMENT LIST

PIPING

106 B Carbon Steel

Diameter	mm	102	152	203	254	305	356
	in	4	6	8	10	12	14
Schedule		10S	10S	10S	10S	10S	10S
Length	m	959	1370	3000	6300	526	30
	ft	3148	4495	9842	20679	1727	100

316 Stainless Steel

Diameter	mm	152	203	254	305	356
	in	6	8	10	12	14
Schedule		10S	10S	10S	10S	10S
Length	m	547	788	1141	3156	30
	ft	1796	2586	3744	10355	100

INSULATION

Warm Salt Pipes

Pipe	mm	102	152	203	254	305	356
Diameter	in	4	6	8	10	12	14
Insulation	mm	51	51	57	57	64	64
Thickness	in	2	2	2.25	2.25	2.5	2.5
Run Length	m	959	1370	3000	6300	526	30
	ft	3148	4495	9842	20679	1727	150

Hot Salt Pipes

Pipe	mm	152	203	254	305	356
Diameter	in	6	8	10	12	14
Insulation	mm	70	102	102	102	102
Thickness	in	2.75	4	4	4	4
Run	m	547	788	1141	3156	30
Length	ft	1796	2586	3744	10355	100

HEAT TRACING

Pipe	mm	102	152	203	254	305	356
Diameter	in	4	6	8	10	12	14
Rating	w/m	144	193	220	262	302	344
	w/ft	44	59	67	80	92	105
Run	m	959	1917	3788	7444	3682	61
Length	ft	3148	6291	12428	24423	12082	200

TABLE 4.4-1

HTS EQUIPMENT LIST (continued)

VALVES

Number Required

<u>Diameter</u> <u>mm</u> <u>in</u>		<u>Warm Salt</u> <u>Ball Valves</u>	<u>Warm Salt</u> <u>Check Valves</u>	<u>Hot Salt</u> <u>Ball Valves</u>	<u>Hot Salt</u> <u>Check Valves</u>
38	1.5	976	3	976	
51	2	6			
102	4	18		10	
152	6			4	
203	8	4	4	4	4
254	10	14	6	14	6
305	12	18		24	

Construction

	<u>Warm Salt</u>	<u>Hot Salt</u>
Body	Carbon Steel	316 SS
Seats	Stellite	Stellite
Back Seals	Graphite	Graphite
Packing	Grafoil	Grafoil

HTS PUMPS

Field Salt Pumps

Pumps

Number	6
Type	Vertical Multi-Stage Centrifugal
Materials	316SS impellers & bowls carbide bearings on Colmonoy sleeves
Salt Temperature	561K (550F)
Rating	.1707 m ³ /sec @ 61 m TDH (2706 gpm @ 200 ft TDH)

Variable Speed Drives

Number	6
Type	Hydraulic Pump and Motor
Turn Down Ratio	3 to 1

Motors

Number	6
Type	Induction
Rating	261 kW shaft (350 hp)

TABLE 4.4-1

HTS EQUIPMENT LIST (continued)

Steam Generator Salt Pumps

Pumps

Number	6
Type	Vertical Multi-Stage Centrifugal
Materials	316SS impellers & bowls carbide bearings on Colmonoy sleeves
Salt Temperature	839K (1050F)
Rating	.0864 m ³ /sec @ 48 m TDH (1370 gpm @ 158 ft TDH)

Variable Speed Drives

Number	6
Type	Hydraulic Pumps and Motor
Turn Down Rates	3 to 1

Motors

Number	6
Type	Induction
Rating	112 kW shaft (150 hp)

HTS PUMPS - Continued

Drain Sump Pumps

Warm Salt Sump Pumps

Number	6
Type	Vertical Centrifugal
Materials	316SS impeller and bowl
Salt Temperature	561K (550F)
Rating	.0568 m ³ /sec @ 21 m TDH (900 gpm @ 70 ft TDH)
Drive	22 kW shaft (30 hp) Induction Motor

Hot Salt Sump Pumps

Number	6
Type	Vertical Centrifugal
Materials	316SS impeller and bowl
Salt Temperature	839K (1050F)
Rating	.0252 m ³ /sec @ 21 m TDH (400 gpm @ 70 ft TDH)
Drive	13.4 kW shaft (18 hp) Induction Motor

TABLE 4.4-1

HTS EQUIPMENT LIST (continued)

HTS DRAIN SUMPS

Warm Salt Sump

Number	2
Dimensions	3 m dia x 3 m high (10 ft dia x 10 ft high)
Materials	6 mm (.25 in) carbon steel 51 mm (2 in) calcium silicate insulation
Other Features	Salt Level Control Heat Traced

Hot Salt Sump

Number	2
Dimensions	2.4 m dia x 2.4 m high (8 ft dia x 8 ft high)
Materials	5 mm (3/16 in) 316SS 102 mm (4 in) calcium silicate insulation
Other Features	Salt Level Control Heat Traced

STEAM GENERATOR

Preheaters

Number	2
Dimensions	1.09 m dia x 12.2 m long (43 in dia x 40 ft long)
Shell (water)	carbon steel
Tubes (salt)	732 U, 19 mm (.75 in) O.D., 2.8 mm (.109 in) wall carbon steel
Duty	35,400 kWt (120.7 x 10 ⁶ Btu/hr)
Area	1043 m ² (11,225 ft ²)

Boilers

Number	4
Dimensions	1.09 m dia x 6.1 m long (43 in dia x 20 ft long)
Shell (salt)	347 SS
Tubes (water)	1000, 25 mm (1 in) O.D. 2 mm (.083 in) wall, 347 SS
Duty	15,700 kWt (53.68 x 10 ⁶ Btu/hr)
Area	406 m ² (4375 ft ²)

TABLE 4.4-1

HTS EQUIPMENT LIST (continued)

Superheaters

Number	2
Dimensions	1.27 m dia x 8.5 m long (50 in dia x 28 ft long)
Shell (salt)	347 SS
Tubes (steam)	734 U, 19 mm (.75 in) O.D., 1.7 mm (0.65 in) wall, 347 SS
Duty	45,300 kWt (154.57 x 10 ⁶ Btu/hr)
Area	835 m ² (8985 ft ²)

Reheater

Number	1
Dimensions	1.36 m dia x 7.6 m long (54 in dia x 25 ft long)
Shell	347 SS
Tubes	1000 U, 19 mm (.75 in) O.D., 1.7 mm (.065 in) wall, 347 SS
Duty	46200 kWt (157.67 x 10 ⁶ Btu/hr)
Area	907 m ² (9760 ft ²)

STEAM GENERATOR - Continued

Steam Drum

Number	1
Dimensions	1.8 m dia x 9.1 m long (6 ft dia x 30 ft long)
Shell	Carbon Steel
Internals	Moisture Separators

TRACE HEATING ELECTRICAL SUPPLY

Switch gear

	Number	
Feeder Breakers	26	4160 volt, 1200 amp
D/Gen'r. Breakers	1	4160 volt, 1200 amp
Tie Breakers	1	4160 volt, 2000 amp
Disconnect Switches	80	4160 V, 15 A fused
Disconnct Switches	80	4160 V, 200 A fused
Main Breakers	80	480 V, 3 phase, 200 A

Transformers

Transformers	80	75 kVA, 4160-480/277 volt, 3 phase, 60 Hz
--------------	----	--

TABLE 4.4-1

HTS EQUIPMENT LIST (continued)

Wire & Cable

	Length
	<u>Length</u>
3 c 250 MCM	1,220 m (4,000 ft)
4 c # 10	85,300 m (28,000 ft)
3 c 2/0 SKV	15,600 m (51,200 ft)

Conduit

.6 m (24 in) Cable Tray w/covers	7,000 m (23,000 ft)
76 mm (3 in) Rigid Conduit	1,220 m (4,000 ft)
13 mm (1/2 in) Rigid Conduit	220,000 m (720,000 ft)

TABLE 4.4-2
HTS DATA LIST

A. Design Characteristics

A.1. Pipe Layout - see Figure 4.4-2

A.2-6 HTS Piping Data

Warm Salt Pipes

O.D.	mm	114	168	219	278	324	356
	in	4.500	6.625	8.625	10.750	12.750	14.00
I.D.	mm	108	161	212	265	315	337
	in	4.260	6.357	8.329	10.420	12.390	13.624
Wall	mm	3	3.4	3.8	4.2	4.6	4.8
	in	.120	.134	.148	.165	.180	.188
Length	m	959	1370	3000	6303	526	30
	ft	3148	4495	9842	20,679	1727	100
Pipe	kg	8000	18,900	59,800	175,400	19,000	1670
Mass	lb	17,700	41,700	131,900	386,700	41,800	3670
Insulation	mm	51	51	57	57	64	64
Thickness	in	2	2	2.25	2.25	2.5	2.5
Insulation	kg	1870	3740	12,000	30,700	3400	225
Mass	lb	4120	8240	26,400	67,800	7500	500

Hot Salt Pipes

O.D.	mm	168	219	273	324	356
	in	6.625	8.625	10.75	12.75	14.00
I.D.	mm	161	212	265	315	337
	in	6.357	8.329	10.42	12.39	13.624
Wall	mm	3.4	3.8	4.2	4.6	4.8
	in	.134	.148	.165	.180	.188
Length	m	547	788	1141	3156	30
	ft	1796	2586	3744	10,355	100
Pipe	kg	7600	15,700	31,750	113,600	1670
Mass	lb	16,700	35,700	70,000	250,600	3670
Insulation	mm	70	102	102	102	102
Thickness	in	2.75	4	4	4	4
Insulation	kg	2160	6150	10,700	34,400	380
Mass	lb	4770	13,500	23,500	75,900	840

TABLE 4.4-2

HTS DATA LIST (Continued)

A.7-8 Valve Data

Materials

Body	Carbon steel for warm salt
	316 SS for hot salt
seats	stellite
back seals	graphite
packings	grafoil

Quantity

Diameter	mm	38	102	152	203	254	305
	in	1.5	4	6	8	10	12
Number		1952	28	4	8	28	42

Note - For the cost estimate, the valve count for 4 in dia and above was increased by an arbitrary 20%. This accounts for additional valves that would be defined in a preliminary (post conceptual) design
 Valve Weight - not available

A. 9 Number of Fittings - not available

A.10 Pump Data

		<u>Field Pumps</u>	<u>Steam Generator Pumps</u>
Number		6	6
Material		347 SS	347 SS
Pump Mass	kg	1137	614
	lb	3730	2015
Drive Mass	kg	309	101
	lb	1014	330

TABLE 4.4-2

HTS DATA LIST (Continued)

A.11 Pump Characteristics

	<u>Field Pumps</u>	<u>Steam Generator Pumps</u>
Number	6	6
Fluid	warm salt	hot salt
Type	vertical multi-stage	vertical multi-stage
Temperature C (F)	288 (550)	566 (1050)
Speed-rpm max	1770	1770
min	600	600
Capacity m ³ /sec	.1703	.0864
qpm	(2700)	(1370)
Design Head* max.	61	48
m	(200)	(158)
(ft) nom.	50	37.5
min.	(165)	(123)
	40	27
	(130)	(88)
NPSH Required m	4.0	3.7
(ft)	(13)	(12)
NPSH Available m	5.2	5.8
(ft)	(17)	(19)
Power kWe (drive)	248	90
hp (pump)	332	120

* Required head varies with warm and hot salt tank liquid level.

TABLE 4.4-2

HTS DATA LIST (Continued)

A.12. Steam Generator

	<u>Preheater</u>	<u>Boiler</u>	<u>Superheater</u>	<u>Reheater</u>
Number	2	4	2	1
Type	TEMA BFU U-tube & Shell	TEMA CEN St.-tube & Shell	TEMA DFU U-tube & Shell	TEMA BFU U-tube & Shell
Orientation	horizontal	vertical	horizontal	horizontal
Tube Passes	2	1	2	2
Shell Passes	2	1	2	2
Tube Fluid	salt	water	steam	steam
Shell Fluid	water	salt	salt	salt
Tube Material	carbon steel	347 SS	347 SS	347 SS
Shell Material	carbon steel	347 SS	347 SS	347 SS
Tube Number	732	1000	734	1000
Tube O.D mm	19.05	25.4	19.05	19.05
in	.750	1.00	.750	.750
I.D mm	13.51	21.18	15.75	15.75
in	.532	.834	.620	.620
Wall mm	2.77	2.11	1.65	1.65
in	.109	.083	.065	.065
Tube Pattern	Rotated Triangular	Rotated Triangular	Square	Square
Tube Pitch mm	20.64	31.75	25.4	25.4
in	13/16	1 1/4	1	1

TABLE 4.4-2

HTS DATA LIST (Continued)

A.12. Steam Generator (Continued)

		<u>Preheater</u>	<u>Boiler</u>	<u>Superheater</u>	<u>Reheater</u>
Baffle Type		none	none	none	none
Straight Length	m	12.19	6.1	8.53	7.62
	ft	40	20	28	25
Shell Diameter	m	1.09	1.09	1.27	1.37
	in	43	43	50	54
Mass per Shell	kg	36,300	15,900	27,200	27,200
	lb	80,000	35,000	60,000	60,000
Surface Area	m ²	1043	406	835	907
per Shell	ft ²	11,225	4375	8985	9760
LMTD	C	23.4	24.6	44.4	63.6
	F	42.2	44.3	80	114.4
Pinch Point	C	5.6	5.6	-	-
ΔT	F	10	10	-	-

Steam Drum

Number	1
Shell O.D. m (ft)	1.83 (6)
Straight Length m (ft)	6.03 (30)
Material	Carbon Steel
Internals	Moisture Separator
Mass kg (lb)	136,000 (300,000)

A.13. HTS Drain Tanks

<u>Tanks</u>	<u>Warm Salt</u>	<u>Hot Salt</u>
Number	2	2
Material	Carbon Steel	316 SS
Diameter m	3.05	2.44
	10	8
Height m	3.05	2.44
	10	8

Pumps:

Number	6	6
Capacity m ³ /sec	.0568	.0252
	900	400
Head m	21.3	21.3
	70	70
Power kWe	41	20
	38	18
		hp

TABLE 4.4-2

HTS DATA LIST (Continued)

B. Operational CharacteristicsB.1-3 Salt Flow Rates and Pressure LossSteam Generator Loop (100 MWe design for 1/2 field)

		<u>Hot Salt Supply</u>	<u>Warm Salt Return</u>
Diameter	mm	305	305
	in	12	12
Flow	kg/hr	1,071,000	1,071,000
	lb/hr	2,362,000	2,362,000
Pressure Loss	kPa	90	69
	psi	13	10

Field Loops (noon maximums for 1/2 field)

Diameter	mm	305	305
	in	12	12
Flow	kg/hr	2,343,000	2,343,000
	lb/hr	5,166,000	5,166,000
Pressure Loss	kPa	103	110
	psi	15	16

Field Manifolds (noon maximum)

Diameter	mm	152-305	102-254
	in	6-12	4-10
Flow	kg/hr	1,171,000 1/4	585,700 1/8
	lb/hr	2,583,000 field	1,292,000 field
Pressure Loss	kPa	145	97
	psi	21	14

B.2 Cooldown Rate - see Table 4.4-8B.4 Steam Generator Duties

	<u>MWt</u>
Preheaters (2)	65.3
Boilers (4)	64.9
Superheaters (2)	79.0
Reheater (1)	<u>42.0</u>
Total	251.2

TABLE 4.4-2

HTS DATA LIST (Continued)

B.5-6 Steam Generator Temperatures - see Figure 4.4-6

B.7-8 Steam Generator Pressures (elevation differences neglected)

		<u>Preheater</u>	<u>Boiler</u>	<u>Superheater</u>	<u>Reheater</u>
Salt Inlet	Mpa	.248	.386	.545	.545
	psig	36	56	79	79
Salt Exit	MPa	.069	.283	.421	.421
	psig	10	41	61	61
Water/Steam Inlet	MPa	18.47	18.34	18.13	2.90
	psig	2706	2660	2630	421
Water/Steam Exit	MPa	18.47	18.34	17.79	2.60
	psig	2680	2660	2580	270

B.9-10 Steam Generator Flow Rates

		<u>Preheater</u>	<u>Boiler</u>	<u>Superheater</u>	<u>Reheater</u>
Salt Flow	kg/hr	1,071,000	535,700	1,399,000	743,600
	lb/hr	2,362,000	1,181,000	3,085,000	1,640,000
Water/Steam Flow Per Unit	kg/hr	156,890	78,446	313,783	291,873
	lb/hr	345,958	172,979	691,915	621,550

B.11 Steam Generator Heat Transfer Coefficients

		<u>Preheater</u>	<u>Boiler</u>	<u>Superheater</u>	<u>Reheater</u>
Overall w/m^2-C		1420	1578	1136	767
	Btu/hr-ft ² -F	250	278	200	135
Water Side Fouling Resistance *					
	C/(w/m ²)	.00018	.00018	.00018	.00018
	F/(Btu/hr-ft ²)	.001	.001	.001	.001

* Included in overall heat transfer coefficient.

Materials. The HTS piping must transport warm salt at 561K (550F) and hot salt at 839K (1050F). A review of the literature (References 4.4-1 through 4.4-7) on materials tests with draw salt and HITEC resulted in the selection of carbon steel for containment of salt up to 672K (750F) and 316 stainless steel for temperatures up to 839K (1050F).

The selection of 316SS is conditional, however. While existing data indicate that stainless steel is most likely a satisfactory selection for service with salt at 839K (1050F) and it is presently the concensus choice (References 4.4-7 and 4.4-8), more testing is required to justify actual use in a solar central power plant. This testing should include:

- corrosion rate as a function of flow velocity up to 5 m/sec (16 ft/sec)
- intergranular corrosion resistance of 839K (1050F) samples that are stressed by a constant load
- resistance to crevice corrosion
- resistance to corrosion fatigue in a thermally cycled test loop.

Insulation. The optimum insulation thickness is that which minimizes the aggregate cost of insulation and the additional collector, receiver, and heat transport subsystem hardware required to supply the heat lost through the insulation. This total cost per foot of insulated pipe was calculated as a

function of pipe diameter for insulation thicknesses ranging from 1.5 to 5.0 inches.

The calculations were based on the use of calcium silicate insulation within a weatherproof aluminum jacket. The thermal conductivities were .00546 W/m-C (.034 Btu/hr-ft-F) for the warm salt lines and .00723 W/m-C (.045 Btu/hr-ft-F) for the hot salt lines. A film coefficient of .00113 W/cm²-C (2.0 Btu/hr-ft²-F) was assumed at the outside surface of the aluminum jacket. The incremental capital cost of the collector, receiver, and heat transport hardware required to supply the heat loss through the insulation was assumed to be \$250 per kWt.

The resulting optimal insulation thicknesses are as shown below in Table 4.4-3.

TABLE 4.4-3
OPTIMUM INSULATION THICKNESS
MILLIMETERS (INCHES)

Pipe Diameter <u>mm (in)</u>	Warm Pipe <u>567K (561F)</u>	Hot Pipe <u>839K (1050F)</u>
102 (4)	51 (2.00)	64 (2.50)
152 (6)	51 (2.00)	70 (2.75)
203 (8)	57 (2.25)	89 (3.50)
254 (10)	57 (2.25)	102 (4.00)
305 (12)	64 (2.50)	108 (4.25)
356 (14)	64 (2.50)	108 (4.25)

Pipe Schedules. Using the allowable stresses of ANSI B31.3-1976 for A358 - Grade 316 stainless steel and for A106B carbon steel, schedule 10S piping was found to be satisfactory for all pipe diameters from 25 to 406 mm (1 to 16 in). The resulting pipe wall thicknesses provide a corrosion allowance of 1.6 mm (1/16 in) for carbon steel and 0.8 mm (1/32 in) for stainless steel. The maximum operating pressures allowed for schedule 10S piping are shown below in Table 4.4-4.

TABLE 4.4-4
ALLOWABLE WORKING PRESSURE
Schedule 10S Pipe

<u>Pipe Diameter</u> <u>mm (in)</u>	<u>Carbon Steel</u> <u>ø567K (561F)</u> <u>kPa (psig)</u>	<u>316 SS</u> <u>ø839K (1050F)</u> <u>kPa (psig)</u>
102 (4)	2495 (362)	4094 (594)
152 (6)	2144 (311)	3233 (469)
203 (8)	2013 (292)	2819 (409)
254 (10)	1958 (284)	2599 (377)
305 (12)	1923 (279)	2454 (356)
356 (14)	1882 (273)	2365 (343)
406 (16)	1647 (239)	2054 (298)

Installation. The layout of HTS piping is shown in BNI Drawing M-2. In order to leave ample space for maintenance of the collectors and receivers, all salt piping runs are located below grade in trenches. The piping runs are sloped, at a 0.1 percent grade, in order to permit complete draining of all HTS

pipng. When draining the HTS, salt flows from the extremities of the warm and hot manifolds at .3 m (1 ft) below grade, to the warm and hot drain sumps at about 2.1 m (7 ft) below grade, next to the energy storage tanks. The concrete trenches extend slightly above grade and are covered with 6.3 mm (0.25 in) checkered steel plate to limit entrance of rainwater into the trenches.

The pipes are supported by steel I-beams which span the trench. The beams are located approximately 150 mm (6 in) from the bottom of the trench to prevent any water entering the trench from flooding the pipe insulation. A steel T-beam, approximately 450 mm (18 in) long and 150 mm (6 in) deep is welded to the bottom of the pipe. The T-beam, which pierces the insulation, rides on the I-beam and provides the actual pipe support. Pipe anchors have additional I-beam sections which are welded to the bottom of the T-beam sliders so as to straddle the I-beam support and thereby limit the forward and reverse movement of the pipe.

This support arrangement is used for two reasons. First, the pipe support loads are carried directly by the pipe, rather than transferred to the pipe through the mildly compressive calcium silicate insulation. Second, rubbing wear, resulting from the daily thermal expansion and contraction of the salt lines, will be considerably less for the steel I- and T-beam combination than for the alternate arrangement which sets the aluminum jacket of the pipe insulation upon an I-beam supported saddle.

The principal disadvantage of this support arrangement is that the T-beam acts as a thermal fin and increases the pipe heat loss at each support. This loss, however, has only a minor effect on the total heat loss from the HTS piping.

The thermal expansion loops shown in the HTS Piping Layout, Drawing M-2, are based on the procedures, design criteria, and allowable stress values of the ANSI B31.3 Standards.

Pipe Size Selection. The optimum pipe diameter is that which minimizes the total cost of the installed pipe, insulation, heat tracing, and power station equipment required to supply piping heat losses and pumping losses.

Calculated values of total cost per foot, presented as a function of volume flow in Figure 4.4-4, are based on a \$250 per Mwt cost for supplying heat losses and a \$1386 per kWe cost for pumping losses. The former value is simply the cost of additional collector, receiver, and heat transport equipment required to supply a megawatt of piping heat loss. The latter value, the cost of additional plant equipment required to generate an additional kilowatt, takes into account the fact that the incremental EPGS cost required to produce an additional kilowatt is less than the total EPGS cost per kilowatt for the plant. This is not the case for collector, receiver, and heat transport equipment due to the modular nature of construction.

Figure 4.4-4 OPTIMUM HTS PIPE DIAMETERS

The total pipe related costs shown in the figure provides a direct criteria for sizing lines where the flow rate is always constant. This is nearly the case for the salt lines to and from the steam generator, since the EPGS operates at maximum guaranteed load (100 MWe) most of the time. Some discretion was exercised in using the figures for the sizing of field piping, however, due to the fact that maximum design volume flow rates occur only at noon. Making due allowance for variation in flow rate with solar insolation during the day, the pipe sizes listed below in Table 4.4-5 were selected.

TABLE 4.4-5
HTS PIPE DIAMETERS

<u>Piping Runs</u>	<u>Pipe Diameter mm (in)</u>
Steam generator supply (hot)	305 (12)
Steam generator return (warm)	305 (12)
Warm salt trunk to field	305 (12)
Warm salt manifolds	254 (10) 203 (8) 152 (6) 102 (4)
Hot salt manifolds	305 (12) 254 (10) 203 (8) 152 (6)
Salt pump suction lines	356 (14)

Pressure Loss. The line friction pressure losses for the HTS Piping Layout, BNI Drawing M-2, are given below:

<u>Field Pump Piping</u>	<u>Pressure Loss</u>	
	<u>kPa</u>	<u>(psi)</u>
Pump to warm manifold	110	(16)
Warm manifold	97	(14)
Receiver run	483	(70)
Hot manifold	145	(21)
Manifold to hot tank	<u>103</u>	<u>(15)</u>
Total	938	(136)

<u>Steam Generator Pump Piping</u>		
Pump to steam generator	90	(13)
Steam generator heat exchangers	407	(59)
Heat exchanger connections	69	(10)
Steam generator to warm tank	<u>69</u>	<u>(10)</u>
Total	635	(92)

The pressure losses associated with the optimum pipe diameters discussed earlier are subject to one practical restraint. The optimum pipe diameters are acceptable choices for the HTS only if they permit operation of all receiver flex hose connections at pressure levels below their 1206 kPa (175 psig) design limit.

The maximum flex hose operating pressure corresponding to the pressure loss presented above is:

	<u>kPa</u>	<u>(psig)</u>
Salt tank static head	+ 179	(+ 26)
Pump pressure rise	+ 938	(+136)
Friction ΔP to nearest flex hose	<u>- 110</u>	<u>(- 16)</u>
Maximum Flex Hose Pressure	1007	(146)

This is below the 1206 kPa (175 psig) flex hose design limit by an ample margin and the optimum pipe diameters are clearly acceptable for use in the HTS.

Heat Loss. Heat losses were calculated for all HTS piping using the optimum thicknesses of calcium silicate insulation presented earlier in Table 4.4-3. The resulting HTS heat loss is shown below in Table 4.4-6.

TABLE 4.4-6
HTS HEAT LOSS SUMMARY

	Day Operation <u>kw</u>	Heat Trace <u>kw</u>
Field Piping		
Warm salt supply lines	258	258
Warm salt manifolds	2718	2718
Hot salt manifolds	2530	1428
Hot salt return lines	<u>538</u>	<u>317</u>
	6044	4721
Steam Generator Piping		
Hot salt supply lines	519	306
Warm salt return lines	<u>200</u>	<u>200</u>
	719	506
HTS Total Heat Loss	6763	5227

Considerable reduction in the HTS heat loss reported above may be achievable by dispensing with the calcium silicate insulation and filling the pipe trenches with a loose, high temperature insulation. One such material, Gilsulate 500, is commonly used for piping systems which operate below 533K (500F). While no previous application of this approach for piping systems in the 561K (550F) to 839K (1050F) range was identified in this study, it would appear that design analyses of piping systems using candidate loose insulation materials such as perlite might prove worthwhile. The corresponding heat loss of the HTS based on the thermal conductivity of perlite would be as shown below in Table 4.4-7.

TABLE 4.4-7

HTS HEAT LOSS WITH LOOSE FILL
INSULATION IN PIPE TRENCHES

	Day Operation <u>kWt</u>	Heat Trace <u>kWt</u>
Field piping	1596	1182
Steam generator piping	<u>190</u>	<u>127</u>
HTS total heat loss	1786	1309

Lacking sufficient field demonstration of the use of loose fill insulation in high temperature piping systems, calcium silicate insulation with aluminum jacket weatherproofing was selected for the HTS conceptual design based on its well demonstrated successful use in the process and power industries.

Heat Tracing. All HTS salt piping is equipped with electrical heat tracing sufficient to preheat the pipe prior to filling with molten salt and to prevent freezing of the salt once the lines have been filled. Total electric heating required to maintain all HTS piping at 561K (550F) was shown above in Table 4.4-4 and totals 5227 kWt. The electric heaters are installed on the exterior pipe surfaces between the pipe and the thermal insulation. This arrangement was selected for the HTS conceptual design. Subsequent experience with the 839K (1050F) draw salt loop at General Atomic Company has indicated that use of heaters within the salt pipe is equally effective or possibly more effective than the exterior heaters described here.

Primary use of the HTS heaters occurs during fill and drain operations which occur each day. The heaters can also be used as required whenever pipe temperature readings indicate a danger of salt freezing.

The heat tracing electrical supply is discussed in Section 4.4.2.5.

Drains and Vents. Provisions for easy and complete filling and draining are essential requirements of the HTS. The system must be drained for maintenance occasionally and refilled prior to subsequent operation. Since the Heat Receiver Subsystem conceptual design is based on draining the receiver of molten salt each evening and refilling it with salt each morning, the

HTS conceptual design has adopted the same drain and fill schedule.

Gravity draining of the field piping is accomplished as follows. With field pumps inoperative, the vent valves at the end of each warm and hot salt manifold are opened to admit atmospheric purge air. The warm and hot salt sump valves are opened to initiate gravity flow of salt from the HTS to the sumps. The continuous slope of the drainage flow path, from 4.3 m (14 ft) above grade in the receivers, to 0.3 m (1 ft) below grade at the manifold ends, to 2.1 m (7 ft) below grade at discharge into the salt sumps, permits continuous and complete draining of the receivers, manifolds and supply lines. Complete draining of the heat receivers and HTS field piping is accomplished in approximately 2 hours.

The heat receivers are sloped for salt drainage into the warm manifold. The drainage then flows through the supply piping and into the warm salt sump. Warm salt entering the sump is pumped into the ESS warm salt storage tank. Hot manifold salt drains into the hot salt sump and is then pumped into the ESS hot salt storage tank. If the temperature of the salt from the hot manifold falls substantially below 839K (1050F), there are provisions for pumping hot sump fluid to the warm salt storage tank so as to avoid degradation of the thermal energy in the hot storage tank.

Filling of the HTS each morning prior to start up of the Heat Receiver Subsystem is accomplished as follows.

With all heat receiver isolation valves closed, with the vent valves at the end of each empty warm salt manifold opened, the variable speed field pumps are energized and operated at part speed. Salt is thereby drawn from the warm salt tanks and delivered first to the field supply piping and then to the warm salt manifold. The continuously ascending flow paths insure complete filling of each successive section, without voids, as the molten salt progresses towards the manifold vents. As the warm salt manifolds approach the filled condition, the field pumps are reduced to minimum speed. Upon filling of the manifolds, the vent valves automatically close and the field pumps are shut down. This completes the filling of the 561K (550F) warm salt portion of the HTS and places the system in readiness for start up of the Heat Receiver Subsystem.

With the Collector/Heat Receiver Subsystem energized by insolation, and with the vent valves at the end of each empty hot manifold opened, the heat receiver isolation valves are opened and the field pumps are operated to fill the heat receivers with molten salt in the manner described in Section 4.3. Flow exiting the receivers enters the hot manifolds initially at temperatures below 839K (1050F). It is directed back to the warm salt tank until the 83 K (1050F) exit temperature is attained. Sufficient pressure to return salt to the warm salt tank cannot be generated until filling of the hot manifolds is complete and the hot

manifold vent valves automatically close. Thereupon, flow of salt to the warm salt tank is established and sustained until salt temperature exiting the heat receivers reaches 839K (1050F). At this time, HRS hot salt flow is directed to the hot salt tank.

Valves. Valve service required by the HTS includes:

- Small isolation valves for 976 heat receiver rows
- Isolation valves for HTS components
- Vent valves
- Check valves for pump discharge lines
- Control valves for steam generator flow balancing and for steam generator flow control during start up and shutdown operations.

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

A number of valve designs were considered. Globe valves were rejected for their relatively large pressure loss and adverse effect upon pipe draining. Gate valves were rejected for their translating valve stems. Two valve designs were selected as satisfactory for 561K (550F) and 839K (1050F) service in draw salt. One is the ball valve design available from Kamyr Valves Inc., and Camaron Iron Works. The other is a tight sealing

butterfly design, available from Clow Corporation, which incorporates a positively loaded conical valve seat.

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

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

Budget prices from valve suppliers indicated that the ball valve is somewhat less expensive than the butterfly valve for 356 mm (14 in) valves and smaller. Therefore, ball valves were selected for the use in the HTS conceptual design, although a more extensive evaluation of both valve types could result in selection of the butterfly valve for a portion of the HTS valves that are 254 mm (10 in) or larger.

4.4.2.3 Pumps

Pump Service. The HTS field design is based upon the use of distributed energy storage and distributed pumping. A warm salt tank and a hot salt tank are located at the center of each side

(west and east) of the collector field. Associated with each pair of tanks is a set of three 50 percent capacity warm salt pumps and a set of three 50 percent capacity hot salt pumps. The third pump in each set is a standby unit. The warm salt pumps withdraw salt from the warm tank and force it through the heat receivers and back to the hot tank. The hot salt pumps withdraw salt from the hot tank and force it through the steam generator and back to the warm tank.

Pump Driver. Both sets of pumps are driven by variable speed drives. Each drive consists of a heavy duty industrial variable displacement hydraulic piston pump, an accumulator and a fixed displacement hydraulic motor. Master Control Subsystem pump speed demand signals modulate salt pump flow rate by changing the hydraulic pump displacement.

A variable speed drive was selected for the HTS warm salt field pumps to permit the matching of field salt flow to solar insolation without the energy loss associated with throttling or the accelerated pump wear that can result from prolonged operation of a constant speed pump at low flow rates. Variable speed drive of the steam generator pumps likewise permits efficient operation at the 25 percent MGL heat balance condition without compromising pump life.

A performance and economic comparison of the variable speed hydraulic drive, selected above, with the efficient Kramerstat wound rotor motor drive showed the two drives to be approximately

equal. A variable speed synchronous motor drive presently under development promises to be superior to both.

Pump Requirements. HTS friction pressure loss calculations and static head requirements associated with salt storage tank liquid levels led to the following pumping requirements.

	<u>Field Pumps</u>	<u>Steam Generator Pumps</u>
Number Required	6	6
Salt Temperature	561K (550F)	839K (1050F)
Flow	.1703 m ³ /sec (2700 gpm)	.0864 m ³ /sec (1370 gpm)
Head-maximum	61 m (200 ft)	48 m (158 ft)
-nominal	50 m (165 ft)	37.5 m (123 ft)
-minimum	40 m (130 ft)	26.8 m (88 ft)

Head requirements vary by ± 10.7 m (± 35 ft), the salt storage tank height. It is minimum when pumping is from a nearly full tank to a nearly empty one; and maximum when the opposite case prevails.

Pump Design. The HTS pumps must be rugged and efficient. Dependable service is required to minimize plant down time. Efficient pumps and pump drives are required to minimize the expense of pump drive auxiliary power.

A number of pump designs have a proven capability for pumping molten salt. Three major types were considered

- Horizontal centrifugal pumps

- Vertical cantelever pumps
- Vertical multi-stage pumps.

Relatively inexpensive horizontal centrifugal pumps, qualified for salt service up to 728K (850F), are available from suppliers such as Dean Brothers Pumps Inc. These pumps are built for easy disassembly and maintenance in the field. Their major disadvantage is relatively low efficiency, some 5 to 10 percentage points below that available with vertical multi-stage pumps, which makes them uncompetitive for this application.

Vertical cantelever pumps are suitable for use with 839K (1050F) salt and are available on a commercial basis from suppliers such as Nagle Pumps, Inc. In these pumps the bearings and seals are separated from the impeller by a cantelever arrangement which prevents their immersion in salt.

Vertical cantelever pumps are constructed with loose clearances for abusive service. They are rugged, relatively inefficient and limited to a single pumping stage. The limitation to single stage construction eliminated this pump design from further consideration for HTS service. The required pumping heads cannot be satisfied with a single stage design.

Vertical multi-stage pumps have been used to pump molten metals and molten salts since the early 1940's. Suppliers such as Byron Jackson have built vertical multi-stage pumps for numerous applications involving molten salt, magnesium, bismuth and lead at temperatures up to 1060K (1450F). The bearings of this pump

run directly in the molten salt. The seals, however, are separated from the salt by an air cover gas system. Salt is drained from the cover gas cavity to maintain a controlled salt level. The vertical multi-stage pump design meets the head requirements of the line focus concept and provides efficient pumping. The design is rugged, easily disassembled and is flexible in handling low NPSH requirements. These advantages led to its selection for both the field pumps and steam generator pumps of the HTS.

The pumps for both warm and hot salt service are constructed of 316 stainless steel with carbide bearings on Colmonoy sleeves. Their performance is listed as follows:

	<u>Field Pumps</u>	<u>Steam Generator Pumps</u>
Capacity - m ³ /sec (gpm) (each)	.1703 (2700)	.0864 (1370)
Head-m (ft), maximum	61 (200)	48 (158)
nominal	50 (165)	37.5 (123)
minimum	40 (130)	26.8 (88)
Efficiency %	85	84
NPSH Required m (ft)	4 (13)	3.7 (12)
NPSH Available m (ft)	5.2 (17)	5.8 (19)

The steam generator conceptual design is based on the use of separate tube and shell heat exchangers to serve as preheaters, boilers, superheaters and reheaters. These heat exchangers are similar in construction to high quality process equipment. The overall arrangement of steam generator heat exchangers is shown in the HTS/ESS Flow Diagram (Figure 4.4-1) and in Figure 4.4-5. Specification sheets for the steam generator heat exchangers are shown on the four pages following Figure 4.4-5.

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

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

Figure 4.4-5 ARRANGEMENT OF STEAM GENERATOR HEAT EXCHANGERS

		DATE		NOTE: STRESS-RELIEVED PARTS MARK - SR RADIOGRAPHED PARTS MARK - XR			
				1 SERVICE: PREHEATER		EQUIPMENT NO.	
				2 (HT SUBSYSTEM)		CONSISTS OF: 2 Parallel SHELLS	
				3 PERFORMANCE: (1 SHELL)		SHELL SIDE	
				4 FLUID WATER		TUBE SIDE	
				5 FLUID Q.Entering 1.75x10 ³ kg/hr (385x10 ³ LB/HR)		SALT 1.191x10 ³ kg/hr (2.625x10 ³ LB/HR)	
				10 SPECIFIC HEAT J/kg-°K (BTU/LB-°F)		1,552.3 (0.371)	
				11 VISCOSITY 10 ⁻³ Pa-sec. 0.036 (0.024) AT 364C (686 °F)		3.13 (2.11) AT 300C (572 °F)	
				12 (10 ⁻³ lb/ft-sec) 1.197 (0.806) AT 238C (460 °F)		2.05 (1.38) AT 369C (696 °F)	
				13 GRAVITY AT °F		AT °F	
				14 NUMBER OF PASSES 2		2	
				15 VELOCITY FT/SEC. AT °F		AT °F	
				17 PRESSURE AT INLET kPag (PSIG) 18,409 (2,670)		427.5 (62)	
				18 PRESSURE DROP kPa (PSI) 48.3 (7)		179.3 (26)	
				19 FOULING FACTOR 0.00018 (0.001)*			
				20 TEMPERATURE IN °C (°F) 238 (460)		369 (696)	
				21 TEMPERATURE OUT °C (°F) 363 (686)		300 (572)	
				22 HEAT EXCHANGER (BTU/HR: 120.7x10 ⁶) 35400 kW LOG. MTD (°F 42.2) 23.4°C CORRECTED MTD °F			
				23 TRANSFER RATE: SERVICE 1,420 (250)** CLEAN		FOULING FACTOR-OVERALL:	
				24 TRANSFER AREA: GROSS 1,043 m ² (11,225 SQ.FT.): EFFECTIVE		SQ.FT.:	
				CONSTRUCTION			
				27 DESIGN PRESSURE kPag (PSIG) 19,305 (2,800)		1,304 (150)	
				28 TEST PRESSURE PSIG			
				29 DESIGN TEMPERATURE °C (°F) 388 (730)		393 (740)	
				30 CORROSION ALLOWANCE mm (in) 3.18 (1/8)		3.18 (1/8)	
				31 TUBES Q. JANT 732 0.019.05mm (3/4" B&G 12 LENGTH 12.19m (40 FT) PITCH 20.64mm (13/16")			
				32 SHELL 1,092.2mm (43") (I.D.) (O.D.)			
				33 CONNECTIONS: SHELL INLET 203.2mm (8 in.) OUTLET 203.2mm (8 in.)			
				34 (SIZE & RATING) CHANNEL INLET 355.6mm (14 in.) OUTLET 355.6mm (14 in.)			
				36 CODES AND STANDARDS APPLIED ASME I & TEMA CLASS R			
				39 HEIGHT: DRY: FLOODED: OPERATING: BUNDLE ONLY:			
				40 PART MATERIAL		PART MATERIAL	
				41 TUBES Carbon Steel		FLOAT. TUBE SHEET	
				42 SHELL " "		FIXED TUBE SHEET Carbon Steel	
				43 SHELL COVER " "		TUBE SUPPORTS	
				44 CHANNEL " "		CROSS BAFFLE Carbon Steel	
				45 CHANNEL COVER		LONG BAFFLE " "	
				46 FLOAT HEAD COVER -		GASKETS	
				47 MANUFACTURER		FOREIGN PRINT NO.	
				48 MODEL		LOCATION DWG. NO:	
				49 VENDOR		P. & I. DIAGRAM NO:	
				50 PURCHASE ORDER NO.			
				51 REMARKS: *m ² °C/W (hr-ft ² - °F/Btu)			
				52 **W/m ² -°C (Btu/hr-ft ² -°F)			
<div style="display: flex; align-items: center;"> <div style="writing-mode: vertical-rl; transform: rotate(180deg); font-size: small; margin-right: 10px;">M-235-E 11-6-98</div> <div style="text-align: center;"> SEP 20210 </div> </div>		HEAT EXCHANGER DATA SHEET		JOB No 13129		REV.	
		GA LINE FOCUS SOLAR CENTRAL POWER PLANT		SHEET 1 OF 4			

DATE		APPROVALS		MATERIAL		SUPPLY		CHECK		ENGINEER		DESCRIPTION		REVISION	
SERVICE: BOILER (HT SUBSYSTEM)										EQUIPMENT NO. CONSISTS OF: 4 Parallel SHELLS					
PERFORMANCE: (1 SHELL)				SHELL SIDE				TUBE SIDE							
FLUID				Salt				Water							
FLUID QUANT. ENTERING				595x10 ³ kg/hr (1,313x10 ³ LB/HR)				88x10 ³ kg/hr (193x10 ³ LB/HR)							
SPECIFIC HEAT J/kg-°K (BTU/LB-°F)				1,552.3 (0.671)											
VISCOSITY 10 ³ Pa-sec				1.65 (1.11) AT 430C (806 °F)				0.049 (0.033) AT 364C (686 °F)							
(10 ³ lb/ft-sec)				2.05 (1.38) AT 369C (696 °F)				AT °F							
GRAVITY				AT °F				AT °F							
NUMBER OF PASSES				1				1							
VELOCITY FT/SEC.				AT °F				AT °F							
PRESSURE AT INLET k Pag (PSIG)				399.8 (58)				18,243 (2,646)							
PRESSURE DROP. kPa (PSI)				103.5 (15)				-							
FOULING FACTOR				0.00018 (0.001)*											
TEMPERATURE IN °C (°F)				369 (696)				363 (686)							
TEMPERATURE OUT °C (°F)				430 (806)				363 (686)							
HEAT EXCHANGED (BTU/HR)				53.68x10 ⁶ 15700 kW LOG. MTD (°F 44.3) 24.6C				CORRECTED MTD °F							
TRANSFER RATE: SERVICE				1,578 (278)**				CLEAN. FOULING FACTOR-OVERALL:							
TRANSFER AREA: GROSS				406 m ² (4,375				SQ.FT.): EFFECTIVE SQ.FT.:							
CONSTRUCTION															
DESIGN PRESSURE kPag (PSIG)				1,034 (150)				19,305 (2,800)							
TEST PRESSURE PSIG															
DESIGN TEMPERATURE °C (°F)				466 (870)				388 (730)							
CORROSION ALLOWANCE mm. (in.)				-				3.18 (1/8)							
TUBES - QUANT.				1,000				O.D. 25.4mm (1") BWG 14 LENGTH 6.1m (20ft) PITCH 31.75mm (1 1/4 in.)							
SHELL - 1,092.2mm (43") (I.D.) (O.D.)															
CONNECTIONS: -SHELL: INLET				304.8mm (12 in.)				OUTLET: 304.8mm (12 in.)							
(SIZE & RATING) -CHANNEL: INLET				508.0mm (20 in.)				OUTLET: 609.6mm (24 in.)							
CODES AND STANDARDS APPLIED ASME I & TEMA CLASS R															
WEIGHT: DRY:				FLOODED:				OPERATING:				BUNDLE ONLY:			
PART				MATERIAL				PART				MATERIAL			
TUBES				347 SS				FLOAT, TUBE SHEET							
SHELL				"				FIXED TUBE SHEET				347 SS			
SHELL COVER								TUBE SUPPORTS							
CHANNEL				Carbon Steel				CROSS BAFFLE				316 SS			
CHANNEL COVER								LONG BAFFLE							
FLOAT HEAD COVER								GASKETS							
MANUFACTURER								FOREIGN PRINT NO.							
MODEL								LOCATION DWG. NO.							
VENDOR								P. & I. DIAGRAM NO.							
PURCHASE ORDER NO.															
REMARKS: * m ² -°C/W (hr-ft ² -°F/Btu)															
** W/m ² -°C (Btu/hr-ft ² -°F)															
BECHTEL CA LINE FOCUS SOLAR CENTRAL POWER PLANT															
JOB No 13129 SHEET 2 OF 4															

NOTE: STRESS-RELIEVED PARTS MARK "SR"
RADIOGRAPHED PARTS MARK "XR"

DATE
APPROVALS
MATERIAL
SUPERVISOR
CHECK
DRAWING
ENGINEER
DESCRIPTION
REV.

1	SERVICE:	SUPERHEATER		EQUIPMENT NO.	
2		(HT SUBSYSTEM)		CONSISTS OF:	2 Series SHELLS
3	PERFORMANCE: (1 SHELL)	SHELL SIDE		TUBE SIDE	
4	FLUID	Salt		Steam	
5	FLUID QUINT. ENTERING	1,549x10 ³ kg/hr (3,415x10 ³ LB/HR)		347x10 ³ kg/hr (766x10 ³ LB/HR)	
6					
7					
8					
9					
10	SPECIFIC HEAT J/kg-°K (BTU/LB-°F)	1,552.3 (0.371)			
11	VISCOSITY 10 ³ Pa-sec	1.65(1.11) AT 430C (806 °F)		0.049(0.033) AT 364C (686 °F)	
12	(10 ³ lb/ft-sec)	0.78(0.524) AT 566C		0.034(0.023) AT 538C (1,000 °F)	
13	GRAVITY	AT °F		AT °F	
14	NUMBER OF PASSES	2		2	
15	VELOCITY FT/SEC.	AT °F		AT °F	
16					
17	PRESSURE AT INLET kPag (PSIG)	537.8 (78)		18,319 (2,657)	
18	PRESSURE DROP. kPa (PSI)	124 (18)		172 (25)	
19	FOULING FACTOR	0.00018 (0.001)*			
20	TEMPERATURE IN. °C (°F)	566 (1,050)		363 (686)	
21	TEMPERATURE OUT. °C (°F)	430 (806)		538 (1,000)	
22	HEAT EXCHANGED (BTU/HR: 154.57x10 ⁶)	45300 kW LOG. MTD (°F 80) 44.4°C		CORRECTED MTD °F	
23	TRANSFER RATE: SERVICE	1,136 (200)*		CLEAN. FOULING FACTOR-OVERALL:	
24	TRANSFER AREA: GROSS	835 m ² (8,985		SQ.FT.): EFFECTIVE SQ.FT.:	
25					
26					
27	DESIGN PRESSURE - kPag (PSIG)	1,034 (150)		19,306 (2,800)	
28	TEST PRESSURE - PSIG				
29	DESIGN TEMPERATURE - °C (°F)	593 (1,100)		566 (1,050)	
30	CORROSION ALLOWANCE				
31	TUBES - QUINT.	734 O.D. 29.05mm (.75" BWG) 16		LENGTH 8.53 m (28 ft) PITCH 25.4m (in.)	
32	SHELL - 1.270mm (50 in.) (1.0 D.O.)				
33	CONNECTIONS: -SHELL: INLET	457.2 mm (18 in.)		OUTLET - 457.2 (18 in.)	
34	(SIZE & RATING) -CHANNEL: INLET	203.2 mm (8 in.)		OUTLET - 203.2mm (8 in.)	
35					
36					
37					
38	CODES AND STANDARDS APPLIED	ASME I & TEMA CLASS R			
39	WEIGHT: DRY:	FLOODED: kg (lb)		OPERATING: 27,216 (60,000) BUNDLE ONLY:	
40	PART	MATERIAL		PART MATERIAL	
41	TUBES	347 SS		FLOAT TUBE SHEET	
42	SHELL	" "		FIXED TUBE SHEET 347 SS	
43	SHELL COVER	" "		TUBE SUPPORTS " "	
44	CHANNEL	" "		CROSS BAFFLE	
45	CHANNEL COVER			LONG BAFFLE 347 SS	
46	FLOAT HEAD COVER			GASKETS	
47	MANUFACTURER			FOREIGN PRINT NO.	
48	MODEL				
49	VENDOR			LOCATION DWG. NO:	
50	PURCHASE ORDER NO.			P.B I. DIAGRAM NO:	
51	REMARKS:	Longitudinal Flow			
52		* m ² - °C/W (hr-ft ² -°F/Btu); ** W/m ² -°C (Btu/hr-ft ² -°F)			
53

HEAT EXCHANGER
DATA SHEET

GA LINE FOCUS SOLAR CENTRAL POWER PLANT

JOB No	13129	REV.	
SHEET 3 OF 4			

DATE		APPROVALS		MATERIAL		SUPPLY		CHECK		DRAWING		ENGINEERING		REVISION		DESCRIPTION		REV.		DATE		JOB No		SHEET	
<div style="text-align: right;">NOTE: STRESS-RELIEVED PARTS MARK . . . RADIOGRAPHED PARTS MARK . . .</div>																									
1		SERVICE: REHEATER										EQUIPMENT NO.													
2		(HT SUBSYSTEM)										CONSISTS OF: 1		SHELLS											
3		PERFORMANCE: (1 SHELL)										SHELL SIDE		TUBE SIDE											
4		FLUID										Salt		Steam											
5		FLUID QUANT. ENTERING										790x10 ³ kg/hr (1,742x10 ³ LB/HR)		311x10 ³ kg/hr (686x10 ³ LB/HR)											
6																									
7																									
8																									
9																									
10		SPECIFIC HEAT J/kg-°K (BTU/LB-°F)										1,552.3 (0.371)													
11		VISCOSITY 10 ³ Pa-sec										1.65 (1.11) AT 430C (806 °F)		0.030 (0.020) AT 309C (587 °F)											
12		(10 ³ lb/ft-sec)										0.78 (0.524) AT 566C (1050 °F)		0.034 (0.023) AT 538C (1000 °F)											
13		GRAVITY										AT °F		AT °F											
14		NUMBER OF PASSES										2		2											
15		VELOCITY FT/SEC.										AT °F		AT °F											
16																									
17		PRESSURE AT INLET kPag (PSIG)										537.8 (78)		2,903 (421)											
18		PRESSURE DROP. kPa (PSI)										48 (7)		200 (29)											
19		FOULING FACTOR										0.00018 (0.001)*		-											
20		TEMPERATURE IN. °C (°F)										566 (1,050)		308 (587)											
21		TEMPERATURE OUT. °C (°F)										430 (806)		538 (1,000)											
22		HEAT EXCHANGED (BTU/HR: 157.67x10 ⁶) 46200 kW LOG. MTD °F 114.4) 63.6°C CORRECTED MTD °F																							
23		TRANSFER RATE: SERVICE: 767 (135)**										CLEAN.		FOULING FACTOR-OVERALL:											
24		TRANSFER AREA: GROSS. 907 m ² (9,760										SQ. FT): EFFECTIVE		SQ. FT.):											
25																									
26		CONSTRUCTION																							
27		DESIGN PRESSURE. kPag (PSIG.)										1,034 (150)		3,447 (500)											
28		TEST PRESSURE. PSIG.																							
29		DESIGN TEMPERATURE. °C (°F)										593 (1,100)		566 (1,050)											
30		CORROSION ALLOWANCE										-		-											
31		TUBES. QUANT. 1,000										0.019, 05mm (75" BWG		16 LENGTH 7.62m (25 ft) PITCH 25.4mm (in.)											
32		SHELL. 1,371.6mm (54") (I.D.) (O.D.)																							
33		CONNECTIONS: -SHELL: INLET.										304.8 mm (12 in.)		OUTLET. 304.8 mm (12 in.)											
34		(SIZE & RATING) -CHANNEL: INLET.										609.6 mm (24 in.)		OUTLET. 609.6 mm (24 in.)											
35																									
36																									
37																									
38		CODES AND STANDARDS APPLIED										ASME I & TEMA CLASS R													
39		WEIGHT: DRY:										FLOODED: kg (lb)		OPERATING: 27,216 (60,000) BUNDLE ONLY:											
40		PART										MATERIAL		PART		MATERIAL									
41		TUBES										347 S.S.		FLOAT TUBE SHEET		-									
42		SHELL										" "		FIXED TUBE SHEET		347 S.S.									
43		SHELL COVER										" "		TUBE SUPPORTS											
44		CHANNEL										" "		CROSS BAFFLE		347 S.S.									
45		CHANNEL COVER										" "		LONG BAFFLE		" "									
46		FLOAT HEAD COVER										" "		GASKETS											
47		MANUFACTURER												FOREIGN PRINT NO.											
48		MODEL																							
49		VENDOR												LOCATION DWG. NO:											
50		PURCHASE ORDER NO.												P. & I. DIAGRAM NO:											
51		REMARKS: * m ² - °C/W (hr-ft ² -°F/Btu)																							
52		** W/m ² - °C (Btu/hr-ft ² -°F)																							
53																									
54																									
55

HEAT EXCHANGER
DATA SHEET

GA LINE FOCUS SOLAR CENTRAL POWER PLANT

JOB No 13129

SHEET 4 OF 4

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

Preheater Design. The preheater is a U-tube design with longitudinal baffles. A major design problem centered around the large difference in water and salt flow rates. A trial design with the low pressure salt on the shell side resulted in a baffle spacing too large for a sound structural design. The salt, therefore, is placed on the tubeside and the high pressure water on the shellside. As a result, the shell thickness is greater than normal for this size of heat exchanger. The shell and the tubes are constructed of carbon steel.

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

places a limit on the height of the boiler. The second constraint, the length of the steam drum, places a limit on the maximum number of risers and downcomers. With the straight tube boiler design, the high pressure water and steam may be located on the tubeside of the heat exchanger. The shell and tubes are constructed of 347 stainless steel. The bonnets are constructed of carbon steel.

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

Superheater and Reheater Design. The superheaters and reheater are U-tube designs with longitudinal baffles. The main and reheat steam are on the tubeside to minimize the cost of the heat exchangers. Shells, tubes, and bonnets are all constructed of 347 stainless steel.

Steam Generator Temperature Profiles. Salt and water/steam temperature profiles for steam generator operation at 25% MGL (Maximum Guaranteed Load), 100% MGL and VWO + 5% OP heat balances are shown in Figure 4.4-6. These profiles show that the lowest temperature of salt flow leaving the steam generator, at 25% MGL, is 541K (513F); well above the 494K (430F) freezing temperature of the salt.

Further discussion of steam generator operating modes is presented below in section 4.4.3.

Figure 4.4-6 STEAM GENERATOR TEMPERATURE PROFILES

4.4.2.5 HTS Electrical Equipment

HTS electrical service consists of power supplies for the actuation of HTS valves and HTS trace heating. The 120 volt single phase required for operation of HTS valves is supplied by the HRS electric supply, run by the same transformers that supply the heat receiver actuator motors. The power supply for heat tracing of the HTS piping is described by the Figure 4.4-7 single line drawing (BNI Drawing ESK-4).

Twelve groups of five transformers - each rated at 75 kVA, 4160-489 Y/277 volt, 3 phase, 60 Hz - will supply power for heat tracing the HTS manifolds. The supply and return piping is heat traced by two groups of ten transformers, each rated the same as the 75 kVA transformers above. A total of fourteen 4160 volt feeders will supply the field.

4.4.2.6 Backup Heaters

The backup oil-fired heater and supporting fuel oil storage tank, shown on the HTS/ESS Flow Diagram (Figure 4.4-1) and the HTS Piping Layout (Figure 4.4-2), are optional equipment not included in the cost estimates reported in section 5. The purpose of such a heater is to make it possible to operate the plant with a substitute heat source during prolonged periods of cloudy weather. This feature has a mild effect on overall plant cost but has a large effect on the acceptability of solar power to a utility. Since the plant is capable of supplying rated power at

any time its real value to the utility is clear and not conditioned by statistical assumptions regarding the availability of solar insolation. With the backup heaters a solar powered plant can be used as part of the rolling reserve; without the heaters it cannot.

A backup heater system for the Line Focus Solar Central Power Plant was considered briefly for the purpose of assessing the approximate cost of adding this optional feature to a 100 MWe plant using a draw salt heat transport loop.

One heater is supplied on each side of the HTS field, near the warm and hot salt storage tanks. Whenever prolonged cloudy weather occurs and the supply of hot salt is depleted, warm salt from the HTS field pumps is directed to the backup oil-fired heater, is heated to 839K (1050F) and delivered to the hot salt storage tank.

The two heaters required are each rated at 126,250 kWt (431,000,000 Btu/hr). Their combined output is sufficient to permit continuous operation of the EPGS at a net output of 100 MWe.

Since the annual use of the heaters for a good solar site would be small, an all radiant heater design was selected. Such heaters are less efficient but cheaper than alternate heaters which transfer a greater portion of the heat by convection and exhaust combustion gases at a lower temperature. It was assumed

that the infrequent use of the heaters would make the annual fuel charge too low to justify increased cost for a more efficient heater. Each heater has 3250 m² (35,000 ft²) of heat transfer area, weighs approximately 317,450 kg (700,000 lb), is 11.6 m (38 ft) in diameter and 29 m (95 ft) high with a 13.7 m (45 ft) stack.

The cost of this option, based on heater prices obtained from KTI, is given on the following page. The total cost of 3.63 million dollars includes the heaters and a four day capacity oil storage tank, but excludes the cost of associated piping, valves and instruments.

4.4.3 HTS Performance, Operation and Control

4.4.3.1 HTS Performance

The measures of HTS performance are the auxiliary power required for HTS pumping and the total thermal loss from HTS piping. Detail discussion of heat losses and pumping requirements appear in Sections 4.4.2.2 and 4.4.2.3. The results are shown below.

CONSTRUCTION COST ESTIMATE

CLIENT DOE/GA DESCRIPTION OPTIONAL EQUIPMENT
 LOCATION _____ BACKUP HEATER CONT. NO. _____
 PROJECT _____ MADE BY _____
 APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil	1749	23		11	34
B	Concrete					
C	Structural Steel					
D	Buildings					
E	Machinery & Equipment	8450	110	177	2360	2647
F	Piping					
G	Electrical					
H	Instruments					
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	10199	133	177	2371	2681
L	Temporary Construction Facilities					32
M	Construction Services, Supplies & Expense					37
N	Field Staff, Subsistence & Expense					31
P	Craft Benefits, Payroll Burdens & Insurances					86
Q	Equipment Rental					32
	INDIRECT FIELD COSTS					218
	TOTAL FIELD COSTS					2899
R	Engineering					145
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					145
	TOTAL FIELD & OFFICE COSTS					3044
V	Contingency					457
W	Fee					131
	TOTAL CONSTRUCTION COST					3,632

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

<u>HTS Heat Loss:</u>	<u>kWt.</u>	<u>Percent of Max. HTS Input</u>	<u>Percent of Rated EPGS Input</u>
Field Piping	6044	1.10	2.41
Steam Generator Piping	<u>719</u>	<u>.13</u>	<u>.29</u>
Total	6763	1.23	2.70

HTS Pumping Power:

<u>EPGS Operating Mode</u>	<u>HTS Operating Mode</u>		
	<u>Morning</u>	<u>Noon</u>	<u>Night</u>
	(Power in Megawatts)		
25% MGL	.221	1.101	.090
100% MGL	.497	1.377	.366
VWO+5% OP	.551	1.431	.420
	(Percent of EPGS Power)		
25% MGL	.96	4.98	.39
100% MGL	.49	1.37	.36
VWO+5% OP	.49	1.30	.38

The HTS conceptual design piping heat loss and pumping power requirements are considered acceptable. Potential reduction of the piping heat loss is discussed in Section 4.4.2.2.

4.4.3.2 HTS Controls

HTS controls consist of instrumentation for monitoring the status of HTS operation, the HTS warm salt flow control, the steam generator hot salt flow control, steam generator salt flow balancing, control of steam generator exit salt temperature and control of steam generator supply salt temperature.

Instrumentation. Considerable instrumentation is provided for monitoring HTS operations. Readings from the following instrumentation are transmitted to the Master Control Subsystem (MCS):

- Temperature indicators located at regular intervals on all field piping for monitoring the need for and the effect of heat tracing
- Temperature indicators for hot salt returning from heat receiver field
- flow indicators for measurement of hot salt flow rate at discharge of steam generator salt pumps
- flow indicators for measurement of warm salt flow rate at discharge of field salt pumps
- Temperature indicators for steam generator supply salt
- Temperature indicators for steam generator exit salt.

HTS Warm Salt Flow Control. The flow of warm salt to the heat receiver field is controlled by varying the speed of the field salt pumps. The pump drives are fixed displacement hydraulic motors that are driven by variable displacement hydraulic pumps. Salt flow rate is controlled by variation of hydraulic pump displacement. Pump flow rate and temperature of the hot salt returning from the heat receiver field are measured and communicated to the MCS. The MCS creates a pump speed demand signal which is communicated to the pump drives. The purpose of this control, discussed in further detail in Section 4.7, is to maintain hot salt from the heat receivers near 839K (1050F).

Steam Generator Salt Flow Control. The flow of hot salt to the steam generator is controlled by means of variable speed salt pumps. Speed is modulated by variable speed hydraulic drives, similar to those described above, which receive their speed demand commands from the MCS. The MCS pump speed demand signal for the steam generator pumps, discussed further in Section 4.7,

is related to EPGS load with a trim to maintain turbine throttle pressure at the 16,547 kPa (2400 psig) operating level.

Steam Generator Salt Flow Balancing. Several controls are associated with the balancing of steam generator internal salt flows.

The first balances the parallel salt flows through the superheater and reheater so as to achieve equal exit salt temperatures. The purpose of the control is to avoid loss of available energy due to mixing of flows at different temperatures. Control is achieved by throttling exit salt flow from the reheater so as to minimize the temperature difference between superheater and reheater exit salt flows.

The second and third such controls perform a similar function for the parallel salt flows through the four vertically oriented boilers and the two preheaters. Exit flows from the four boilers are throttled as required to obtain equal exit salt temperatures. Equal preheater exit flow temperatures are obtained in the same way. While the flow through similar parallel paths might be expected to divide evenly without control, the unsymmetrical approach and exit plumbing and the eventual plugging and removal of heat exchanger tubes during the plant life require some means for balancing flows.

Steam Generator Exit Salt Temperature. To avoid the possibility of inadvertant freezing of salt, an override feature is provided

which changes salt flow rate or EPGS load as required to prevent salt exit temperature from dropping below 539K (510F). This provides a 45C (80F) margin above the salt freezing temperature of 494K (430F). The relationship between the 539K (510F) exit salt override temperature and the exit temperatures during normal operation of the EPGS is shown in Figure 4.4-8.

Steam Generator Salt Supply Temperature-- During normal operation the steam generator supply salt temperature remains constant at 839K (1050F). During shutdown of the EPGS, however, supply salt temperature is ramped downward from 839K (1050F) to 567K (560F) by blending of flows from the hot and warm salt tanks. During EPGS startup salt supply temperature is ramped back up to 839K (1050F) by the same blending of warm and hot salt flows. This blending process is controlled by the relative pump speed demand signals which the MCS communicates to the warm and hot salt pump drives. The measurement of salt supply temperature is communicated to the MCS for use in the control loop.

4.4.3.3 HTS Operation

Operation of the HTS involves:

- Draining and filling
- Freeze protection
- Initial filling of steam generator with salt
- Steam generator startup
- Steam generator shutdown.

Figure 4.4-8 STEAM GENERATOR SALT EXIT TEMPERATURE

Draining and Filling. All HTS piping is sloped at a 0.1 percent grade in a continuously descending drain path from the manifold vents and the steam generator vents to the drain sumps next to the warm and hot salt tanks. The drain sump liquid levels are 2.4 m (8 ft) below grade. HTS field line drainage flows from the heat receivers, at 46 m (14 ft) above grade, to the manifolds, at .3 to 1.2 m (1 to 4 ft) below grade, to the field lines, at 1.2 to 1.8 m (4 to 6 ft) below grade, and into the warm and hot salt sumps, at 2.4 m (8 ft) below grade. Sump salt is pumped into the warm and hot salt tanks for storage. When hot sump salt is below 893K (1050F), it is pumped to the warm salt tank to avoid degradation of energy stored in the hot tank.

The continuous slope of the HTS piping provides a continuously ascending fill path from the warm and hot salt pumps, to the vents at the ends to the warm and hot manifolds, and to the vents at the top of the steam generator. Thus complete filling is attained without trapping of air pockets just as complete draining is accomplished without residual liquid puddles.

The fill operation is done with the heat receiver isolation valves closed and is described in further detail in Section 4.4.2.2.

Freeze Protection. Draw salt freezes at 494K (430F). The freezing temperature can be slightly above this value at times depending on the level of impurities and the level of control of salt chemistry. Freeze protection of the HTS conceptual design

is based on the criteria that salt temperatures shall not be permitted to drop below 533K (500F). The time required for insulated warm and hot salt lines to cool to 533K (500F) in a 256K (0F) environment is tabulated below in Table 4.4-8.

TABLE 4.4-8

TIME TO COOL TO 533K (500F) - HOURS

Pipe Diameter <u>mm</u>	<u>inches</u>	Warm Pipe <u>561k(550F)</u>	Hot Pipe <u>839k(1050F)</u>
102	4	2	18
152	6	3	26
203	8	4	36
254	10	5	48
305	12	6	56
356	14	7	64

From the table it is clear that the 12 to 16 hour overnight shutdown periods pose no freezing hazard to the hot salt pipes. The overnight cooldown of stagnant salt in the warm lines, however, definitely requires some form of freeze protection.

The Heat Receiver Subsystem (HRS) is also vulnerable to freezing of the salt when solar energy collection is stopped. Convective and radiation losses from the receiver are sufficient to freeze the heat receiver salt in a fairly short period if the receiver salt is not drained. The HRS conceptual design is based upon the

draining of receiver salt each afternoon as solar energy collection is terminated.

Since the heat receivers cannot be completely drained without draining of the HTS manifolds, the complete draining of all HTS piping each night is one method of freeze protection that was considered. It minimizes thermal loss from the pipes at night. Several hours of heat trace operation is required each morning to preheat the HTS piping to 561K (550F) prior to refilling with salt. Approximately 10 MWh(t) are required for this purpose each day. This is approximately half as much heat as is lost from the warm and hot salt storage tanks each 24 hours.

An alternative approach to freeze protection of the HTS involves refilling of the HTS warm salt manifolds after the heat receivers have been drained and the heat receiver isolation valves have been closed. This is the HTS ready state that must precede HRS startup the following morning in any event. The hot manifolds and hot return lines remain empty. Freezing in the warm salt lines and warm manifolds is prevented by energizing the heat tracing system. The electric energy requirement is 2976 kWe. Operation for 12 to 16 hours consumes 36 to 48 MWh (electric) each night. The electric power requirement is the major disadvantage of this approach.

A third approach to freeze protection of the HTS is based on installation of a normally valved-closed section of pipe which bypasses the heat receivers and connects the warm and hot

manifold extremities so as to permit circulation of warm salt at night. With the heat receiver isolation valves closed and the HTS manifolds refilled with warm salt, the receiver bypass line is valved open to permit a small flow from the warm salt tank to be pumped through the warm manifolds, the bypass lines, the hot manifolds and then back to the warm tank. The additional heat lost from the warm salt tank during 12 to 16 hours of such operation amounts to 57 to 76 MWh (thermal). An additional 5 MWh of electric heat tracing is required during the drain and refill operation which precedes the recirculation of warm salt. Overnight recirculation cools the full warm salt storage tanks by from 14C (25F) to 19C (34F). The cooling of the warm tank can be avoided by blending a small amount of hot salt with the returning warm salt prior to its re-injection into the warm tank.

A comparison of the energy requirements of the three freeze protection methods discussed above is presented in Table 4.4-9 below.

TABLE 4.4-9
FREEZE PROTECTION ENERGY REQUIREMENTS

<u>Protection Method</u>	<u>Electric Energy MWh (e)</u>	<u>Thermal Energy MWh (t)</u>
Evening Drain/Morning Refill	10	0
Evening Drain & Refill Heat Trace at Night	36-48	0
Evening Drain & Refill Salt Circulation at Night	5	57-76

The freeze protection approach selected for the HTS conceptual design¹ is the first method discussed. The HTS lines are drained of salt each evening and refilled with salt the following morning. Approximately 2 hours of heat trace operation is required in preparation for the refill operation each morning.

The selection was made on the basis of minimum energy consumption. None of the approaches considered relieves the plant operator from the requirement of draining and refilling the HTS once each day. The approaches differ only in the timing of the refill operation and the total energy loss to the power plant.

Filling of the Steam Generator With Draw Salt. The dominant requirement of the salt filling procedure is that it must avoid freezing the salt. Satisfaction of this requirement leads to a procedure different from that normally used for nuclear and fossil fired steam generators. Fortunately it is straight forward and provides sure protection against freezing.

The steam generator is initially empty of water and salt. 561K (550F) salt is in the warm salt tank. The fill procedure requires the following steps:

- Initial filling of the preheaters, boilers and steam drums with cold water
- Preheating of the preheater and boiler water to 561K (550F) with the auxiliary heater loop shown in Figure 4.4-9

Figure 4.4-9 AUXILIARY HEATER LOOP

- Energizing the trace heating on the boiler superheater and reheater shells and all steam generator loop piping
- Opening of vents of the uppermost point of the loop, (the superheater and reheater salt supply connection)
- Pumping of salt from the warm salt tank into the steam generator supply and return lines
- Automatic closing of the steam generator vent when steam generator and supply salt line liquid levels reach the top of the steam generator.

When the steam generator and associated loop piping are filled with salt, the valving is adjusted to establish a small (5-10 percent) flow of 561K (550F) salt from the warm tank to the supply salt line, through the steam generator, and back to the warm tank. With this flow of warm salt established, the steam generator 561K (550F) standby condition is sustained by the circulation of warm salt, and the steam generator loop heat tracing can be turned off.

In order to fill the steam generator as outlined above, the relative elevation of the heat exchangers and the design of their interconnections is arranged so the ascending liquid level traps no air pockets and complete filling is attained. The same design feature also permits complete draining.

The fill procedure leaves the steam generator bedded down with 561K (550F), 7102 kPa (1030 psig) water in the preheaters, boilers and steam drum, and with a small flow of 561K (550F) salt circulating from the warm tanks to the steam generator and back. Once the steam generator has been charged with salt, it will be

sustained in this standby state until EPGS startup begins or until it is drained of salt. Heat losses associated with the standby mode are:

Overnight EPGS Shutdown 4 to 6 MWh (t)

Prolonged Shutdown 12 MWh(t) per day

The daily heat loss for prolonged shutdown is about half of that lost from the thermal storage tanks. Pipe insulation alternatives discussed in Section 4.4.2.2 could possibly reduce this heat loss penalty by a factor of four.

Draining of Steam Generator. Salt is drained from the steam generator by opening the steam generator vent and permitting gravity drainage to the warm salt tank. When liquid salt level in the steam generator descends to equal the liquid level in the warm salt tanks, subsequent drainage is routed to the warm salt sumps. The sump pumps transfer the salt drainage to the tanks.

Drainage of steam generator water is accomplished as follows. Water from the preheater inlet is delivered to the condenser, the flow being propelled by gravity and steam drum pressure. Sparqers diffuse the energy of the hot water as it flashes into the condenser.

Steam Generator Startup. Startup of the steam generator is related to, and is a part of, startup of the EPGS. There is some overlap of the following discussion with the EPGS startup.

discussion of Section 4.6.3. Startup of the steam generator is accomplished with the following sequential steps.

Using the auxiliary heater loop described earlier, and illustrated in Figure 4.4-9, the steam drum water is heated to 589K (600F) while the boiler and preheater is kept at 561K (550F) by the circulating flow of warm salt. Thermal stratification keeps the steam drum and boiler waters separate. This step prevents flashing of boiler and preheater water when steam flow is subsequently initiated.

Simultaneously establish a small bypass flow of steam to the condenser and begin the ramping upward of steam generator supply salt temperature by blending salt flows from the warm and hot salt tanks. This establishes salt and water temperature profiles required for stable operation. The maximum rate at which salt temperature can be ramped, and the programming of the salt supply temperature during startup, is a function of EPGS requirements and is discussed in Section 4.6.3. The steam generator start transient is complete once the rated supply salt temperature of 839K (1050F) is reached.

Steam Generator Shutdown. Shutdown of the steam generator is accomplished by the ramping downward of supply salt temperature from 839K (1050F) to 561K (550F) in a manner consistent with the shutdown requirements of the EPGS (discussed in Section 4.6.3). Supply salt temperature is varied by the blending of salt flows from the warm and hot salt tanks.

Operation at Low EPGS Loads. The merits of valving out unneeded boilers and preheaters at low EPGS loads were examined and rejected.

While individual boilers and preheaters can be valved out at low EPGS loads, there is no need to do so. Use of the full bank of heat exchangers at minimum heat balance results in a slightly lower pinch point temperature difference than is attained with fewer heat exchangers. Hence valving out of heat exchangers gives a slight reduction in performance.

On the other hand, isolation of a heat exchanger can create problems. Stagnant salt in a preheater will freeze at heat balance conditions where feedwater temperature falls below 494K (430F). If both the water and salt sides of a boiler or preheater are isolated, the eventual freezing of the salt can only be prevented by actuation of shell heat tracing.

REFERENCES

- 4.4-1 Bohlmann, E.G., Heat Transfer Salt for High Temperature Steam Generation, ORNL-TM-3777
- 4.4-2 DuPont Hitec, Heat Transfer Salt, E. I. DuPont de Nemours and Company, Explosives Department, Chemical Products Sales Division, Wilmington, Delaware 19898
- 4.4-3 Silvermann, M.D. and Engel, J.R., Survey of Technology for Storage of Thermal Energy in Heat Transfer Salt, ORNL/TM-5682, January 1977.
- 4.4-4 Sorokin, Y.I., and Tseitlin, K.L., Corrosion of Metals in Nitrate-Nitrite Melts at 500°C, Khim. Prom. 41 (1) 64 (1965)
- 4.4-5 Sukhotin, A.M. et al, Stability of Materials of Construction in Contact with Nitrate-Nitrite Melts, Sov. Chem. Ind. 6, 1 (1974) (translation of Khim Prom)
- 4.4-6 Conceptual Design of Advanced Central Receiver Power Systems, Martin Marietta Corporation Report, Contract No. EG-77-C-03-1724, September 1978.
- 4.4-7 Kramer, C.M., Smyrl, W.H., and Estill, W.B., Corrosion of Fe Alloys in Hitec at 823K, Sandia Laboratories Report SAND 78-8256, printed February 1979.
- 4.4-8 Telecon from A.C. Madsen to T. Tracey of Martin Marietta on March 23, 1979

4.5 ENERGY STORAGE SUBSYSTEM

4.5.1 ESS Requirements

The Energy Storage Subsystem (ESS) is designed to store energy from the Heat Receiver Subsystem (HRS) which is not used directly for electric power generation. The stored energy is delivered from the ESS to the EPGS during cloudy periods or at night. The ESS must provide salt storage tanks, cover gas that is free of moisture and carbon dioxide, a salt purification loop, and a salt charging and makeup system.

4.5.2 ESS Design

The ESS conceptual design addressed the following subjects

- Insulation system design
- Insulation thickness
- Tank size
- Tank design
- Foundation design
- Cover gas system
- Salt purification
- Freeze protection
- Salt charging and makeup.

The ESS tankage arrangement selected in the Task 2 Parametric Analysis requires two warm salt tanks and two hot salt tanks. One warm and one hot tank are located at the center of the

western half of the collector/heat receiver field. The remaining warm and hot tanks are at the center of the eastern half of the field. Associated with each pair of tanks is a cover gas system and a salt charging and makeup system. Each tank also has a purification loop with an auxiliary salt heater.

4.5.2.1 Drawings and Data

The Figure 4.5-1 HTS/ESS Flow Diagram (BNI Drawing M-1) shows the functional relationships between the ESS tankage and the purification and salt charging equipment. Equipment location is shown in the Figure 4.5-2 HTS Piping Layout (BNI Drawing M-2). The ESS Equipment and Data Lists are presented in Tables 4.5-1 and 4.5-2 respectively.

REVISION	DATE	BY	CHKD	APPD
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33				
34				
35				
36				
37				
38				
39				
40				
41				
42				
43				
44				
45				
46				
47				
48				
49				
50				
51				
52				
53				
54				
55				
56				
57				
58				
59				
60				
61				
62				
63				
64				
65				
66				
67				
68				
69				
70				
71				
72				
73				
74				
75				
76				
77				
78				
79				
80				
81				
82				
83				
84				
85				
86				
87				
88				
89				
90				
91				
92				
93				
94				
95				
96				
97				
98				
99				
100				

4.5-5/-6

TABLE 4.5-1

ESS EQUIPMENT LIST

Salt Storage Tanks

Warm Salt Tanks	Quantity	2
	Dimensions	18 m dia x 10.7 m high (59ft dia x 35ft high) Dished Roof
	Material	SA-516-Gr 70 Carbon Steel
	Plate Thickness	
	Wall	19mm (3/4in) Bottom 3.66m (12 ft) 14.3mm (9/16in) Middle 3.66m (12 ft) 7.9mm (5/16in) Top 3.35m (11 ft)
	Floor	7.9mm (5/16in) 15.9mm (5/8in) Outer .6m (2 ft)
	Roof	6.4mm 1/4in with stiffeners
	Connections	305mm (12in) dia and 356mm (14in) dia Each with Sparger-Diffusers
	Insulation	76mm (3in) Calcium Silicate with Aluminum Jacket
	Foundation	Compacted Dry Soil
Hot Salt Tanks	Quantity	2
	Dimensions	18.3m dia x 11m high (60ft dia x 36ft High) Dished Roof
	Material	SA-240-316 Stainless steel
	Plate thickness	
	Wall	30mm (1-3/16in) Bottom 3.66m (12 ft) 21mm (13/16in) Middle 3.66m (12 ft) 11mm (7/16in) Top 3.66m (12 ft)
	Floor	6mm (1/4in) 19mm (3/4in) Outer 1/2 ft
	Roof	6mm (1/4in) with Stiffeners
	Connections	305mm (12in) dia and 356mm (14in) dia Each with Sparger-Diffusers
	Insulation	127mm (5in) Calcium Silicate with Aluminum Jacket
	Foundation	Compacted Dry Soil
Cover Gas System	Quantity	2
	Capacity	.028m ³ /sec (60 ft ³ /min) capacity
	H ₂ O Desiccant	2-100% trains, silica gell desiccant with electric heat regeneration
	CO ₂ Desiccant	1-100% train, mono- and diethanol amine desiccant with electric heat regeneration
	Interconnect- ing Vents	43m (140 ft), 102mm (4 in) dia Sch. 10S, 316 SS
	Valves	1.4 kPa (+0.2 psig) pressure relief -1.4 kPa (-0.2 psig) vacuum relief

TABLE 4.5-1

ESS EQUIPMENT LIST (Continued)

Warm Salt Tank Heater	Quantity	2
	Type	Oil-fired natural convection with intermediate heat transfer salt
	Duty	2.64×10^6 kJ/hr (2.5×10^6 Btu/hr)
	Exchange Area	23.2 m^2 (250 ft ²)
Hot Salt Tank Heater	Tubes	256m (840 ft) of 25mm (1in) dia, Sch. 40, carbon steel
	Quantity	2
	Type	Oil-fired, natural convection with intermediate heat transfer salt
	Duty	4.22×10^6 kJ/hr (4.0×10^6 Btu/hr)
	Exchange Area	37.2 m^2 (400 ft ²)
	Tubes	411m (1350 ft) of 25mm (1in) dia Sch. 10S, 316 stainless steel

Salt Purification Loops

Warm Salt Circulation Pumps	Quantity	4
	Type	Vertical, multi-stage centrifugal
	Materials	316 SS impellers & bowls, carbide bearings
	Capacity	$.0227 \text{ m}^3/\text{sec}$ @ 23m tdh (360 gal/min, 75 ft tdh)
	Drive	17 kWe (20 bhp) induction motor
Hot Salt Circulation Pumps	Quantity	4
	Type	Vertical, multi-stage, centrifugal
	Materials	316 SS impellers & bowls, carbide bearings
	Capacity	$.0265 \text{ m}^3/\text{sec}$ @ 17m tdh (420 gal/min, 55 ft tdh)
	Drive	12 kWe (15 bhp) induction motor
NO ₂ Bubble Columns	Quantity	2
	Material	6mm (1/4in) carbon steel, 51mm (2in) calcium silicate insul.
	Dimensions	.3m dia x 1.8m high (1 ft dia x 6 ft high)
	Internals	Rings and perforated plates
	Comments	Gas vent chamber with salt level control
Warm Salt Precipitate Plate and Frame Filters	Quantity	4-50% capacity
	Type	Plate and frame
	Materials	Carbon steel, 51 mm (2in) calcium

TABLE 4.5-1

ESS EQUIPMENT LIST (Continued)

Flowrate		silicate insulation .0227m ³ /sec (360 gal/min)
<u>Salt Charging and Makeup System</u>		
NaNO ₃ Storage	Quantity	2
	Construction	203mm (8 in) thick concrete walls and floor, 5mm (3/16 in) CS gable roof
	Dimensions	3m (10 ft) wide x 3.6m (12 ft) high x 20m (65 ft)
	Comments	1.5m (5 ft) deep walls below grade, sloped at 45°, to tunnel conveyor
KNO ₃ Storage	Quantity	2
	Construction	203mm (8 in) thick concrete walls and floor, 5mm (3/16 in) CS gable roof
	Dimensions	3m (10 ft) wide x 3.6m (12 ft) high x 14m (45 ft) long
	Comments	1.5m (5 ft) deep walls below grade, sloped at 45°, to tunnel conveyor
NaNO ₃ Hoppers	Quantity	6
	Construction	6mm (0.25 in) carbon steel
	Diameter	1.8m (6 ft)
	Height	5.8m (19 ft)
KNO ₃ Hoppers	Quantity	6
	Construction	6mm (0.25 in) carbon steel
	Diameter	1.5m (5 ft)
	Height	5.8m (17 ft)
Mixed Salt Hopper	Quantity	2
	Construction	6mm (0.25 in) carbon steel
	Diameter	1.8m (6 ft)
	Height	5.2m (17 ft)
Salt Melt Tank	Quantity	2
	Construction	6mm (0.25 in) carbon steel
	Dimensions	2.4m (8 ft) high x 10 ft dia
	Insulation	76mm (3 in) calcium silicate with aluminum jacket
Molten Salt Transfer Pump	Quantity	2
	Type	Horizontal, centrifugal
	Material	316 SS
	Capacity	.0003m ³ /sec @ 30m tdh (5 gal/min, 100 ft tdh)

TABLE 4.5-1

ESS EQUIPMENT LIST (Continued)

	Drive	.45 kWe (0.5 bhp) induction motor
Dowtherm - Salt Heat Exchanger	Quantity	2
	Construction	25mm (1 in) dia, 316 SS tube
	Tube Length	134m (440 ft) coiled in melt tank
Dowtherm Vaporizer	Quantity	2
	Duty	1.06×10^6 kJ/hr (1.0×10^6 Btu/hr)
	Fuel	#2 fuel oil
	Capacity	2268 kg/hr (5000 lb/hr)
	Comments	Dowtherm A @ 561K (550F) Package unit
Dowtherm Flash Tank	Quantity	2
	Construction	A106, CS, 0.25 in. walls
	Dimensions	.8m (2.5 ft) dia x 1.5m (5 ft) high
	Insulation	51mm (2 in) calcium silicate with aluminum jacket
Dowtherm Condensate Tank	Quantity	2
	Construction	A106, CS, 6mm (0.25 in) wall
	Dimensions	.6m (2 ft) dia x 1.2m (4 ft) high
	Insulation	51mm (2 in) calcium silicate with aluminum jacket
Dowtherm Feed Pumps	Quantity	2
	Type	Horizontal, centrifugal
	Material	Cast steel
	Capacity	.001m ³ /sec @ 81m tdh (18 gal/min, 265 ft tdh)
	Drive	1.8 kWe (2 bhp) induction motor

Salt Conveyor

	<u>Tons/Hr</u>	<u>Length</u> <u>ft</u>	<u>m</u>
NaNO ₃ unloading conveyor	60	180	55
KNO ₃ unloading conveyor	40	140	43
NaNO ₃ tunnel conveyor	8	140	43
KNO ₃ tunnel conveyor	6	100	30
NaNO ₃ transfer conveyor	8	100	30
KNO ₃ transfer conveyor	6	100	30
NaNO ₃ weigh - feeder	6	80	24
KNO ₃ weigh - feeder	4	80	24

TABLE 4.5-1

ESS EQUIPMENT LIST (Continued)

Dowtherm Piping	Quantity	30m (100 ft)
	Material	A106 CS, with 25mm (1 in) calcium silicate
	Size	25mm (1 in) dia, Sch. 40
Molten Salt Piping	Quantity	152m (500 ft)
	Material	A106 CS, with 51mm (2 in) calcium silicate
	Size	25mm (1 in) dia, Sch. 40
	Comments	Heat traced

TABLE 4.5-2

ESS DATA LIST

A. Design Characteristics

1. Storage medium:
 Specific heat, J/kg-°K 1552.3 (0.371)
 (BTU/lb -°F)
 Viscosity, 10³ Pa-s 3-27 (2.2) @ 567 K (561 F)
 (10³lb/ft-sec) 0.78 (0.524) @ 839 K (1050 F)
 Density, kg/m³ (lb/ft³) 1,903.2 (118.7) @ 567 K (561 F)
 1,722.2 (107.4) @ 839 K (1050 F)
2. Storage Medium Mass,
 kg (lb) 9,489,600 (20,905,000)
3. Tank Characteristics

	<u>Hot</u>	<u>Warm</u>
Number	2	2
Dimensions, D x H, m (ft)	18.6 x 11 (61 x 36)	18.0 x 10.7 (59 x 35)
Volume, m ³ (ft ³)	2978 (105210)	2710 (95690)
Material	SA-240-316	SA-516-Gr. 80
Design Tem., °C (°F)	565.6 (1050)	293.9 (561)
Tank Mass, kg (lb)	129,365 (285,200)	93,349 (205,800)
Surface Area/Tank, m ² (ft ²):		
Top	298.7 (3215)	279.3 (3007)
Side	630.4 (6,786)	592.4 (6,377)
Bottom	271.4 (2922)	254.0 (2734)
Total	1200.5 (12923)	1125.7 (12118)
Insulation Thickness, mm (in)	76 (3)	127 (5)
Insulation Mass,	1,539 (3,393)	2,410 (5,313)

4-8. Included in Table 4.4-2, HTS Data List,

9&10. Pump Characteristics for purification loops only:

	<u>Hot</u>	<u>Warm</u>
Material	316 S.S.	316 S.S.
Number	4	4
Type	Centrifugal, vertical, suction canned	Centrifugal, vertical, suction canned
Speed, RPM	1,760	1,760
Head, m (ft)	16.8 (55)	22.9 (75)
NPSH, M (ft)	3 (10)	3 (10)
Oper. temp., °C (°F)	565.6 (1050)	293.9 (561)

TABLE 4.5-2

ESS DATA LIST (Continued)

Power, kWE (BHP)	12 (15)	16 (20)
Capacity, m ³ /sec (gpm)	.03 (420)	.02 (360)

B. Operating Characteristics-

1. Extractable capacity MWh (10⁹BTU) 1094 (3.733)
2. Charging rate, MW:

Maximum	307.5
Design	307.5
Minimum	0
3. Discharging rate, MW:

Maximum	274.0
Design	251.2
Minimum	0
4. Discharge rate duration, hr:

Maximum	4.20
Design	3.20
Minimum	-
5. Storage medium operating temperature, °C (°F):

Cold	293.9 (561)
Hot	565.6 (1050)
6. Not applicable
7. Operating pressures, MPa (PSIA):

	<u>Hot</u>	<u>Warm</u>
Fully charge - Top	0.10 (14.7)	0.10 (14.7)
Bottom	0.28 (40.8)	0.29 (42.7)
Fully discharge - Top	0.10 (14.7)	0.10 (14.7)
Bottom	0.11 (16.2)	0.11 (16.3)
8. Heat Loss Rate, MW: .967
9. Storage ramp rate, % of maximum discharge power/minute 100%

4.5.2.2 Insulation System Design

Calcium Silicate insulation with aluminum jacket weatherproofing was selected for the warm salt tanks. The insulation is placed on the outside surfaces of the 561K (550F) carbon steel tanks.

Exterior calcium silicate insulation with aluminum weather proofing was selected for the hot storage tanks also, but only after thorough evaluation and rejection of several internal insulation design approaches. The use of external insulation on a tank containing 839K (1050F) salt, requires the use of a 316 SS tank. The use of internal-insulation was pursued in hope it could permit the use of a less expensive carbon steel tank for storage of the hot salt. Some of the problems associated with internal insulation schemes are mentioned below.

The use of fiber insulation which is directly exposed to the 839K (1059F) salt is unproven for long life applications. Its use risks degradation of the insulation and consequent overheating of the carbon steel tank shell. The insulation degradation products provide an additional source of contamination which the salt purification system must accommodate.

The use of internal insulation, hermetically sealed in a stainless steel liner, is made difficult by the relative thermal expansion of the outer carbon steel shell, the insulation and the liner. This relative expansion tends to prevent the liner from transmitting hydrostatic forces through the insulation to the

tank shell under all operating conditions (e.g. initial charge). If the liner is designed to carry hydrostatic forces, it becomes as thick as the tank shell.

The cost of the support structure for spanning the roof of the tank with refractory insulation tends to diminish the benefits accruing from use of carbon steel instead of stainless steel.

Finally, a stainless steel tank bottom is required even if internal insulation is adopted. The insulating effect of the dry foundation soil is very substantial and causes the central position of the tank bottom to exceed carbon steel limits even if it is internally insulated.

For these reasons, internal insulation of the salt tanks was rejected. The ESS conceptual design is based on the use of conventional external insulation.

4.5.2.3 Insulation Thickness

Optimum insulation thicknesses were calculated for the warm and hot salt storage tanks, based on minimizing of the cost of insulation plus the cost of CS, HRS, and HTS equipment required to supply the lost heat. The optimum calcium silicate thicknesses of 76 mm (3 in) for the warm salt tanks and 127 mm (5 in) for the hot salt tanks were selected for the ESS conceptual design.

The warm and hot salt tank heat losses resulting from this insulation design are shown below.

Two Warm Tanks	353 kWt
Two Hot Tanks	<u>614</u> kWt
Total	967 kWt

Less than one percent of the ESS tank heat loss is through the bottom of the tank due to the thermal insulating effect of the dry soil foundation. Heat from the tanks drives moisture from soil in proximity to the tanks.

4.5.2.4 Tank Storage Volume

The ESS tank storage volumes are sized to provide for:

- 420 MWh(e) of EPGS operation from storage,
- 24 hour heat loss from ESS salt tanks,
- Overnight heat loss from steam generator loop, (when EPGS is shutdown)
- Overnight generation of auxiliary steam, (when EPGS is shutdown)
- 10 MWh (electric) of additional EPGS operation to account for power drawn from grid each morning for heat tracing prior to refilling with salt.

The required thermal energy storage is shown in Table 4.5-3.

TABLE 4.5-3

ESS THERMAL STORAGE REQUIREMENT

	<u>MWh (thermal)</u>
EPGS Operation	1046.3
ESS Tank Heat Loss - 24 hr	23.2
Overnight Heat Loss from Steam Generator Loop	4.0
Overnight Generation of Auxiliary Steam	18.7
Heat Trace Operation Prior to Morning HTS Pipe Refilling	25.0
Total	1117.2

The EPGS thermal requirement is based on operation at 100% Maximum Guaranteed Load heat balance except for one-half hour of operation at 25% MGL heat balance which occurs during startup and shutdown of the EPGS.

The 1117.2 MWh (thermal) requirement is satisfied by heating 9,478,000 kg (20,900,000 lb) of draw salt from 567K (561F) to 839K (1050F) and placing it in the hot storage tank. This leads to specification of an active storage volume of 2750 m³ (97300 ft³) for each of the two hot salt tanks and 2480 m³ (87800 ft³) for each of the two warm salt tanks.

4.5.2.5 Tank Design

In the Requirements Definition document, the Barstow site is classified as Seismic Zone 3. Seismic considerations, therefore, have a strong influence on the tank design. These considerations include the tank height to diameter ratio and the tank floor design.

A tank height to diameter ratio of 0.6 was selected for the conceptual design for three reasons. First, the low ratio reduces the tank overturning moment due to seismic accelerations. Second, an earthquake produces waves in the salt inventory of the tank. If these waves become large enough, they can buckle the tank as they strike the walls. A low tank height reduces the size of these waves. Third, tank overturning moments due to earthquakes increase the soil bearing stress in certain locations of the tank foundation and the low tank height reduces these stresses.

In normal tank designs, stresses in the floor material are considerably less than in the wall material. As a result, the floor thickness is less than the wall thickness. If a large overturning moment is applied to such a tank, however, one edge of the tank may lift from the foundation. In this situation, the stress at the floor-to-wall weld can rupture the thin floor. To minimize the chances of this occurring, the floor thickness of the tanks in the conceptual design is doubled for the outer 0.6 m (2 ft) of the tank radius.

Two additional design parameters, the location of the piping penetrations and the minimum tank freeboard, have an influence on the design dimensions of the salt tanks. The centerlines of the piping penetrations are located 0.6 m (2 ft) above the tank floor to reduce the wall thermal and pressure stresses near the penetration. The salt inlet lines continue past the wall penetration to a pipe which follows the perimeter of the tank floor and discharges incoming flow through mixing spargers. These spargers mix intermediate temperature salt, entering the warm tank during HTS startup and shutdown operations, with the warm tank inventory and minimize tank temperature excursions. They also tend to keep precipitates in suspension, making them available for transport to the purification loop. The salt outlet lines have a similar perimeter manifold arranged to collect, for transport to the purification loop, as many suspended solids as possible. In order to guarantee that salt is always available to the pump suction, a minimum liquid level of .6 m (2 ft) from the tank bottom is always maintained. In addition a salt freeboard of .3 m (1 ft) is provided at the top of the tank for the cover gas volume. Therefore the total tank height which is unavailable for thermal storage is .9 m (3 ft).

Based on the above discussions, final designs were prepared for the hot and warm salt tanks. The design details are summarized in Figures 4.5-3 and 4.5-4. The warm tanks have slightly smaller dimensions than the hot tanks due to the greater density of the warm salt. A flat, rather than a shallow cone bottom has been selected for the conceptual design. The static friction between

Figure 4.5-3 WARM SALT STORAGE TANK DESIGN

Figure 4.5-4 HOT SALT STORAGE TANK DESIGN

the tank and the foundation is sufficient to prevent the tank from sliding during an earthquake. The wall course thicknesses are based on the maximum hydrostatic forces and the allowable material stresses of the ASME Pressure Vessel Code, Section 8, Division 1. The API tank code does not cover the high operating temperatures of these salt tanks. The tank designs include a corrosion allowance of 3.2 mm (1/8 in) on the carbon steel and 1.6 mm (1/16 in) on the stainless steel. The hemispherical section roof includes radial stiffeners to accommodate snow and ice loads. An access manhole is also included in the roof.

4.5.2.6 Foundation Design

The design soil for the Barstow location is UBC Class 3 sandy gravel. This soil, when dry, is a suitable foundation for the warm and the hot salt tanks.

The foundations for the conceptual design tanks are constructed by excavating the tank area to a depth of 3 m (10 ft). At this depth, the soil has a bearing strength of 49,000 kg/m² (10,000 lb/ft²). The excavation volume is then backfilled with compacted soil so as to maintain the 49,000 kg/m² (10,000 lb/ft²) bearing strength. The tanks are placed directly on the compacted soil. Heat transfer through the bottom of the tank will dry the foundation soil. Unlike clay, the sandy gravel soil will not shrink appreciably as it dries. The tendency for tank settling during the first years of plant operation is therefore small. With a tank filled with warm salt, the foundation bearing load is

approximately 2060 kg/m² (4200 lb/ft²). The design bearing strength of 49,000 kg/m² (10,000 lb/ft²) has been specified to minimize the possibility of tank settling following an earthquake.

Site soils containing moderate to large amounts of clay, are less suitable as tank foundations. Alternative foundation designs for such soils are shown in Figure 4.5-5. The design shown in Figure 4.5-5(a) is suitable for soils with a moderate amount of clay. The foam glass insulation and compacted sandy soil are thermal insulation for the undisturbed clay. The sand acts as a buffer to prevent tank movements due to thermal expansion from abrading the fragile foam glass. The rock and gravel perimeter ring contains the sand and foam glass. The foundation design shown in Figure 4.5-5(b) has been developed for soils with predominate amounts of clay. The expanded shale is a thermal insulation and bearing material which prevents drying and shrinkage of the undisturbed clay.

4.5.2.7 Cover Gas System

The cover gas system is designed to provide air that is free of H₂O and CO₂ to the storage tanks and to minimize the tank ventilation heat losses. The use of relatively pure air minimizes salt dissolution and the formation of hydroxides and carbonates in the storage salt. Ventilation heat losses would amount to approximately 1.5 MWh(t) per day if all tanks are allowed to vent directly to the atmosphere. The cover gas system

(a) FOR SOILS WITH MODERATE AMOUNT OF CLAY

(b) FOR SOILS WITH PREDOMINANT AMOUNTS OF CLAY

Figure 4.5-5 ALTERNATE FOUNDATION DESIGNS

includes a silica gel water vapor desiccant, an amine based carbon dioxide absorbant, and interconnecting tank vent lines which minimize dumping of hot cover gas.

The water vapor desiccant is a commercial silica gel unit. It consists of two 100 percent capacity desiccant trains with an electric heater for regenerating the silica gel. The carbon dioxide absorbant system is also a commercial unit. It contains an absorber column, a stripper column, a heat exchanger, a pump, and controls. The absorbant is a combination of monoethanol and diethanol amines. Heat for regeneration of the absorbent is provided by an electric heater.

The salt tanks at each storage location are interconnected by a 102 mm (4 in) diameter vent line. As a tank is filled with salt, air expelled from the tank flows through the vent to the adjacent, emptying tank. As air from a cold tank enters a hot tank, it expands as it is heated from 567K (561F) to 839K (1050F). A pressure relief valve, set to open at 0.7 kPa (0.1 psig), is located on the vent line to ensure that the tanks are not exposed to large positive (above-atmospheric) pressure. As air from a hot tank enters a warm tank, it contracts as it is cooled from 839K (1050F) to 567K (561F). To protect the tanks from operation at large subatmospheric pressures, a vacuum relief valve, set to open at -0.7 kPa (-0.1 psig), is also installed in the vent line. The interconnecting vent system minimizes the amount of ambient air brought into the tanks during a charge-discharge cycle. This reduces the size of the H_2O and CO_2

removal systems, reduces the heat transferred from the salt to ambient air entering the cover gas system, and reduces the amount of hot cover gas that is dumped to the atmosphere.

4.5.2.8 Salt Purification

Air is the preferred cover gas for the salt storage tanks. Carbon dioxide and water vapor in the air, however, react with the salt to form carbonates and hydroxides. The carbonates and hydroxides form precipitates in the warm salt which may foul heat transfer surfaces in the steam generator and the receiver tubes. To minimize the annual expense of salt blowdown to control carbonate and hydroxide concentrations, salt purification loops are included in the ESS conceptual design.

A salt purification loop is provided for each salt storage tank. Each loop consists of an NO_2 bubble column, a salt circulation pump, and a salt heater (for freeze protection). Plate and frame precipitate filters are also provided in those loops serving the warm salt tanks. During the evening, salt is circulated through the NO_2 bubble column. The column is a counter-flow reaction vessel designed to convert the carbonates and hydroxides back to the original nitrates. The water vapor and carbon dioxide formed in the conversion reactions are vented from the column.

The filters in the warm salt loop are designed to trap any carbonate or hydroxide precipitates which do not react in the NO_2 column. Two full capacity filters are provided to ensure

continuous operation during filter cleaning or maintenance. Discharge from the filter, or in the case of the hot tanks, from the NO₂ bubblers, flows to the salt heater and returns to the storage tank.

4.5.2.9 Freeze Protection

Two kinds of salt freeze protection have been provided in the ESS conceptual design. They are electric heat tracing of the tank floor and the oil-fired salt heater in the salt purification loops.

The electric heat tracing is used to preheat the tank floor as salt is initially introduced to a tank during the salt charging operation. This ensures that molten salt from the salt charging system will not freeze as it enters the tank. Once the salt level submerges the tank ports, the purification loop pump is started and, the oil-fired salt heater is used to maintain the salt temperature. The heat tracing is then turned off.

A salt heater is located in the purification loop of each salt tank. The warm tank heaters are designed to maintain a salt temperature of 567K (561F). The hot tank heaters are designed to maintain a temperature of 839K (1050F).

The natural convection heaters include fire tubes at the bottom, an intermediate heat transfer salt, and heat exchanger tubes containing the salt from the storage tanks at the top. This

design ensures that local, high temperature areas of the fire tube do not affect the storage salt. When the heater is in operation, salt flows from the storage tank to the salt circulation pump, through the heater, and back to the storage tank. Fuel oil for the heaters is stored in the central fuel oil storage tank adjacent to the turbine building.

4.5.2.10 Salt Charging and Makeup

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

Individual charging systems are located adjacent to each warm salt storage tank. Each system is sized to supply the molten salt required for one-half of the plant within a 4 1/2 month period and includes the following equipment:

- 30 day concrete storage bunkers for solid NaNO_3 and KNO_3
- Tunnel and transfer conveyors for solid NaNO_3 and KNO_3
- 48 hour storage hoppers for solid NaNO_3 and KNO_3
- Weigher-feeders for solid NaNO_3 and KNO_3
- 8 hour mixed salt hopper

- Melt tank with an integral heat transfer coil
- Dowtherm A vaporizer, flash tank, condensate tank, and feed pump
- Molten salt transfer pump.

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

The oil-fired Dowtherm vaporizer produces saturated vapor at approximately 175 kPa (25 psia) and 561K (550F). The vapor is delivered to a flash tank to separate any liquid which is carried over with the vapor. The vapor then flows to the heat transfer coil, where it condenses to form a saturated liquid at 561K (550F). The condensate flows to a condensate storage tank, through a feed pump, and returns to the vaporizer. This system is used, in preference to a direct-fired salt heater, for the uniform and close temperature control of the melt tank inventory afforded by a condensing heat exchanger.

Although a salt purification system is provided in the ESS conceptual design, it is anticipated that a small salt blowdown from the storage tanks or the steam generator may be required. Also, salt spills and leaks will inevitably occur. Makeup salt is provided by operation of the salt charging system on an intermittent basis as required.

4.5.3 ESS Performance, Operation and Control

4.5.3.1 ESS Performance

The measure of ESS performance is the fraction of thermal energy, delivered to the hot storage tank, which is eventually discharged for use in the power plant.

On a day when the ESS hot salt tanks are completely filled, a total of 1117 MWh(t) are stored. Of this, 2.1 percent is lost in the form of heat transfer from the salt storage tanks. An additional 4.3 percent is required for HTS and EPGS maintenance through the night. 93.6 percent of the stored energy is made available to the steam generator for operation of the EPGS. 97.9 percent of the energy stored in the ESS is retrieved for subsequent EPGS operation or for service to the HTS.

4.5.3.2 ESS Control

The major instrument readings communicated from the ESS to the Master Control System (MCS) are temperature and liquid level indications for each salt storage tank. The operation of the tank valves and the purification loops and the salt charging systems is in response to commands of the MCS.

4.5.3.3 ESS Operation

All warm salt flow from the steam generator is normally routed into the warm salt tank before it is permitted to pass on to the heat receiver field. In this way thermal mass of the warm salt tank smoothes out the temperature variations of the entering warm salt flow and places a lower limit on the temperature of salt departing for the receiver field.

The preferred operating mode for the hot salt tank passes hot salt flow from the receivers, through the tank before permitting it to pass on to the steam generator. This smoothes out any temperature variations of the hot receiver salt flow and provides a very constant steam generator supply salt temperature.

In order to provide for direct transfer of heat from the Heat Receiver Subsystem (HRS) to the steam generator (e.g., if a hot or warm tank is valved off for inspection or repair), bypass plumbing is provided to permit flow around the warm and hot tanks when desired by the plant operator.

The salt purification system is operated from the Master Control Room daily as required to maintain and control acceptable salt purity levels. Once the ESS is initially charged with salt, the salt charging system is operated intermitantly as required to provide make up salt.

4.6 ELECTRICAL POWER GENERATION SUBSYSTEM

4.6.1 EPGS Requirements

The function of the EPGS is to convert heat from the Energy Storage Subsystem (ESS) and from the Heat Receiver Subsystem (HRS) into electricity. The EPGS must be capable of delivering a net electric output of 100 MWe at Maximum Guaranteed Load (MGL) operating conditions. It is utility industry practice to require the capability for operating the EPGS with valves wide open and with 5 percent overpressure in the steam generator (VWO + 5% OP). This operating condition provides an electric output of about 110 MWe. The EPGS is also required to operate at part load. Minimum load operation for this conceptual design is at 25 percent of MGL turbine throttle flow. Net electric output at this condition is approximately 23 MWe.

4.6.2 EPGS Design

The EPGS conceptual design is based on a detailed consideration of the following equipment:

- Turbine-generator
- EPGS Cycle Equipment
- Heat Rejection Equipment
- EPGS Auxiliaries
- Electrical Plant Equipment.

4.6.2.1 Drawings and Data

A layout of the turbine building is presented in Figure 4.6-1 (BNI Drawing M-7). Heat balances for EPGS operation at 25% MGL, 100% MGL and VWO + 5% OP are presented in Figures 4.6-2 through 4.6-4 respectively. Figures 4.6-5 and 4.6-6 show the single line diagrams for the EPGS electrical system (BNI Drawings ESK 1 and 2). The EPGS equipment list is presented in Table 4.6-1 and the EPGS Data List is presented in Table 4.6-2.

4.6-3/-4

[illegible]

4.6-5/6

LEGEND
ENGLISH SI

W = LB/HR = .4535 kg/HR
P = PSIA = 6.895 kPa
F = °F = (°F - 32) / 1.8 = °C
H = BTU/LB = 6.46 x 10³ J/kg
HEAT RATE = BTU / KWHR = 1.055 kJ / BTU

TURBINE TYPE: G.E. TC2F - 16.5" L.S.B. 3600 RPM

STEAM GENERATOR DUTY = 242.86 x 10⁶ BTU/HR

GROSS UNIT OUTPUT = 25,945 KWe

GROSS TURBINE CYCLE = $\frac{242.86 \times 10^6 \text{ BTU/HR}}{25,945 \text{ KWe}} = 9,361 \text{ BTU/KWHR}$

GROSS TURBINE CYCLE EFFICIENCY = $\frac{3413}{9,361} \times 100 = 36.5 \%$

NET UNIT OUTPUT = 23117 KWe

NET CYCLE HEAT RATE = $\frac{242.86 \times 10^6 \text{ BTU/HR}}{23,117 \text{ KWe}} = 10,506 \text{ BTU/KWHR}$

NET CYCLE EFFICIENCY = $\frac{3413}{10,506} \times 100 = 32.5 \%$

NOTE: STEAM GENERATOR DRUM BLOWDOWN TO WASTE IS MADE UP TO CONDENSER

Figure 4.6-2. Heat balance at 25% of guaranteed throttle flow

REVISION	DATE	BY	CHKD	APP'D
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				
13				
14				
15				
16				
17				
18				
19				
20				
21				
22				
23				
24				
25				
26				
27				
28				
29				
30				
31				
32				
33				
34				
35				
36				
37				
38				
39				
40				
41				
42				
43				
44				
45				
46				
47				
48				
49				
50				
51				
52				
53				
54				
55				
56				
57				
58				
59				
60				
61				
62				
63				
64				
65				
66				
67				
68				
69				
70				
71				
72				
73				
74				
75				
76				
77				
78				
79				
80				
81				
82				
83				
84				
85				
86				
87				
88				
89				
90				
91				
92				
93				
94				
95				
96				
97				
98				
99				
100				

ISSUED FOR REPORT

BECHTEL
SAN FRANCISCO

GA LINE FOCUS SOLAR CENTRAL POWER PLANT

25 % OF GUARANTEED THROTTLE FLOW
HEAT BALANCE

100 No. 13129 DRAWING No. M-4 REV. 0

4.6-7/8

LEGEND

ENGLISH	SI
W = LB/HR	= .4535 kg / HR
P = PSIA	= 6.895 k Pa
F = °F	= (F-32) / 1.8 °C
H = BTU/LB	= 6.46 x 10 ⁴ KWHR / kg
HEAT RATE	BTU / KWHR = 1.055 kJ / BTU
TURBINE TYPE: G.E. TC2F - 10.5' L.S.B. 3600 RPM	
STEAM GENERATOR DUTY = 857.25 x 10 ⁶ BTU/HR	
GROSS UNIT OUTPUT = 106,199 KWe	
GROSS TURBINE CYCLE HEAT RATE = $\frac{857.25 \times 10^6 \text{ BTU/HR}}{106,199 \text{ KWe}} = 8,072 \text{ BTU/KWHR}$	
GROSS TURBINE CYCLE EFFICIENCY = $\frac{3413}{8,072} \times 100 = 42.3\%$	
NET UNIT OUTPUT = 100,548 KWe	
NET CYCLE HEAT RATE = $\frac{857.25 \times 10^6 \text{ BTU/HR}}{100,548 \text{ KWe}} = 8526 \text{ BTU/KWHR}$	
NET CYCLE EFFICIENCY = $\frac{3413}{8526} \times 100 = 40.0\%$	

NOTE: STEAM GENERATOR DRUM BLOWDOWN TO WASTE IS MADE UP TO CONDENSER

Figure 4.6-3. Maximum guaranteed load heat balance

DATE	ISSUED FOR REPORT	BY	DATE	BY	DATE
10-1-79					
BECHTEL					
SAN FRANCISCO					
GA LINE FOCUS SOLAR CENTRAL POWER PLANT					
MAXIMUM GUARANTEED LOAD HEAT BALANCE					
200 No.	13129	DRAWING No.	M-5	REV.	0

Figure 4.6-4. VWO +5% O.P. load
heat balance

4.6-11/12

NOTES:
1) DIESEL GENERATOR RATED AT 2000 KW HAS BEEN SELECTED AND BASED UPON ASSUMED USE OF MULTIPLEXING OF CONTROL UNDER EMERGENCY CONDITIONS.

Figure 4.6-5. Main single-line diagram

0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95	96	97	98	99																																																																																																				
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75		76		77		78		79		80		81		82		83		84		85		86		87		88		89		90		91		92		93		94		95		96		97		98		99	
0		1		2		3		4		5		6		7		8		9		10		11		12		13		14		15		16		17		18		19		20		21		22		23		24		25		26		27		28		29		30		31		32		33		34		35		36		37		38		39		40		41		42		43		44		45		46		47		48		49		50		51		52		53		54		55		56		57		58		59		60		61		62		63		64		65		66		67		68		69		70		71		72		73		74		75

Figure 4.6-6. Single-line diagram for 480-Volt power distribution

△									
△									
△									
△									
△	9-10-77	ISSUED FOR REPORT							
NO.	DATE	REVISION	BY	CHKD	DESIGNED	ENGR	PROJ	APPV	
<p align="center">BECHTEL SAN FRANCISCO</p> <p align="center">GENERAL ATOMIC COMPANY</p> <p align="center">480 VOLT POWER DISTRIBUTION SINGLE LINE</p>									
	JOB No.	DRAWING No.	REV.						
	13129	ESK-2	0						

TABLE 4.6-1

EPGS EQUIPMENT LIST

Mechanical Equipment

Main Turbine

Quantity	1
Type	Tandem compound two-flow with 419 mm (16.5 in) last stage blades
Main Steam Conditions	Single reheat - Two casing,
Reheat Steam Temp.	16,547 kPa, 811K (2400 psig, 1000F)
Condenser Pressure	811K (1000F)
Design Speed	3 in. HgA
Maximum Guaranteed Load	3600 rpm
Gross Unit Output	107,205 KWe
HP Turbine Eff'cy	82.87%
Reheat Turbine Eff'cy	91.38%
Includes	Lube oil System; all turbine inlet valves and turbine-generator control system package: steam seal equipment with regulating system, steam packing exhauster, valves & piping; turbine exhaust spray system, motor driven turning gear; insulation & lagging

Generator

Quantity	1
Type	Hydrogen-cooled
Maximum Guaranteed Load Output	122,720 KVA max
RPM	3600
Power Factor	0.9
H ₂ Pressure	414 kPa (60 psig)
Frequency	60 Hz
Phase	3 ϕ
Includes	Voltage & excitation controls; hydrogen cooling system; seal oil system; stator conductor cooling system; all pertinent coolers

Feedwater Heater No. 1

Type	Horizontal, U-tube, 2-pass
Duty	22.26 x 10 ⁶ kJ/hr (21.10 x 10 ⁶ Btu/hr)
Tube Size	15.9 mm (5/8 in) O.D.; 13BWG
Tube Material	304 L
Gross Surface	430.5m ² (4640 ft ²)

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Feedwater Heater No. 2

Type	Horizontal, U-tube, 2-pass
Duty	44.08 x 10 ⁶ kJ/hr (41.78 x 10 ⁶ Btu/hr)
Tube Size	15.9 mm (5/8 in) O.D. 13 BWG
Tube Material	304 L
Gross Surface	306.2m ² (3300 ft ²)

Feedwater Heater No. 3

Type	Horizontal, U-tube, 2-pass
Duty	51.04 x 10 ⁶ kJ/hr (48.38 x 10 ⁶ Btu/hr)
Tube Size	15.9mm (5/8 in) O.D., 13 BWG
Tube Material	304 L
Gross Surface	390 m ² (4200 ft ²)

Feedwater Heater No. 5

Type	Horizontal, straight tube
Duty	41.51 x 10 ⁶ kJ/hr (39.35 x 10 ⁶ Btu/hr)
Tube Size	15.9mm (5/8 in) O.D. 22 BWG
Tube Material	304
Gross Surface	259.8m ² (2800 ft ²)

Feedwater Heater No. 6

Type	Horizontal, straight tube
Duty	24.68 x 10 ⁶ kJ/hr (23.39 x 10 ⁶ Btu/hr)
Tube Size	15.9mm (5/8 in) O.D. 22 BWG
Tube Material	304
Gross Surface	213.4m ² (2,300 ft ²)

Feedwater Heater No. 7

Type	Horizontal, straight tube
Duty	21.09 x 10 ⁶ kJ/hr (19.99 x 10 ⁶ Btu/hr)
Tube Size	15.9mm (5/8 in) O.D., 18 BWG
Tube Material	304
Gross Surface	259.8m ² (2800 ft ²)

Deaerating Heater

Type	Direct Contact Tray, Vertical
Material	285 Gr C
Outlet Capacity	352,261 kg/hr (776,760 lb/hr) max.
Operating Pressure	414 kPa (60 psia)
Temperatures	Extraction steam, 569K (565F)
	Saturation steam 414 kPa, 418K (60 psia, 292F)
	Condensate inlet, 400K (260F)
	Drain Inlet, 427K (308F)

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Deaerator Storage Tank

Capacity	28.7 m ³ (7,600 gal)
Diameter	2.74 m (9 ft)
Overall length	5.97 m (19 ft-7 in)
Shell thickness	17.5 mm (11/16 in)
Head	20.6 mm (13/16 in)
Type Heads	Flanged & dished
Design press	517 kPa (75 psig)

Steam Generator Feedwater Pump

Quantity	2
Type	Multistage, horizontal, barrel
Capacity	.044 m ³ /sec @ 2207 m head (703 gpm @ 7242 ft head)
Motor Power	1128 kWe (1390 hp)
RPM	3500
NPSH	10.7 m (35 ft)

Condensate Pumps

Quantity	3
Type	Multistage, vertical can
Capacity	.0371 m ³ /sec @ 157 m head (588 GPM @ 514 ft head)
Motor Power	77 kWe (95 hp)
NPSH	3 m (95 ft)

Condenser (less tubes)

Quantity	1
Type	Single pressure & shell, divided waterbox,
Duty	518.1 x 10 ⁶ kJ/hr (491.1 x 10 ⁶ Btu/hr)
Turbine Exhaust Flow	227,802 kg/hr (502320 lb/hr)
Cooling Water Flow	2.65 m ³ /sec (42,000 GPM)
Inlet/Outlet CW Temp.	302K/316K (84F/108F)
Supply includes:	Circ. water line expansion joints; main turbine exhaust hood & expansion joints; main turbine exhaust transition pieces with heater mounting flanges

Condenser Tubes

Quantity	7,936
No. of passes	2
Length/pass	8.2 m (27 ft) effective
Tube O.D./BWG	22 mm (7/8 in)/18
Material	Admiralty
Surface	4518 m ² (48,700 ft ²)
Cleanliness	85%
Design Velocity	2.1 m/sec (7 ft/sec)

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Heat Transfer Coefficient	.352 W/cm ² -K (620 Btu/hr-ft ² -F)
Impingement & air removal tubes	22 mm (7/8 in), 20 BWG, 304 SS

Steam Jet Air Ejector Package

No. of stages	2
Includes	Inter & after condensers
Capacity	.01-015 m ³ /sec (20-30 SCFM)

Hogging Ejector:

No. of stages	1
Capacity	.165 m ³ /sec (350 SCFM)

Circulating Water Treatment Package

Quantity	1
Includes	Storage tanks, pumps and injectors for sodium hypochlorite and sulfuric acid

Cooling Tower

Quantity	1
Type	Mechanical Draft
Flow Rate	2.65 m ³ /sec (42,000 GPM)
No. of Cells & Fans	4 (rectangular)
Cell dimensions	15.5 m x 15.5 m (51 ft x 51 ft)
Fan Drivers	Induction motors, 4 x 100 kW shaft (4 x 135 HP)
Approach	5.6C (10F)
Range	13.3C (24F)

Instrument and Service Compressed Air Package

Quantity	1
Includes	Two .353 m ³ /sec, 862 kPa (750 SCFM/125 psig) compressors, 3 after coolers, 1 air receiver, 3 moisture separators, 3 intercoolers, 6 oil coolers and filters, 3 inlet filter silencers, 2 dryers, 2 prefilters, 2 after filters, 2 instrument air dryer coolers

Fire Water Pump and Diesel Drive

Quantity	1
Type	Vertical, centrifugal
Capacity	.1262 m ³ /sec @ 107 m head (2000 GPM @ 350 ft head)
RPM	1700

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Efficiency	77%
Driver	Diesel engine, 172 kW shaft (230 BHP)

Fire Water Pump and Motor

Quantity	1
Type	Vertical, centrifugal
Capacity	.1262 m ³ /sec @ 107 m head (2,000 GPM @ 350 ft head)
RPM	1700
Efficiency	82%
Driver	Induction motor, 183 kW shaft (245 BHP)

Fire Water Jockey Pump and Motor

Quantity	1
Type	Vertical, centrifugal
Capacity	.00315 m ³ /sec @ 49 m head (50 GPM @ 160 ft head)
RPM	1700
Efficiency	40%
Driver	Induction motor, 3.7 kW shaft (5 BHP)

Demineralized Water Storage Tank/Condensate Storage Tank:

Quantity	1 each
Capacity	189 m ³ (50,000 gal)
Dimensions	6.7 m dia. x 5.5 m high (22 ft x 18 ft high)
Material	Stainless steel

Circulating Water Pumps

Quantity	2
Capacity	1.325 m ³ /sec @ 14.3 m head (21,000 GPM @ 47 ft head)
Type	Vertical, wet pit, multistage
Motor Power	224 kW shaft (300 HP)

Condensate Polisher Package

Quantity	3-50% capacity trains
Capacity	.082 m ³ /sec (1300 GPM)

Demineralized Water Makeup Storage Transfer Package

Quantity	2 trains
Capacity	.0035 m ³ /sec (55 GPM)

Lube Oil Storage Transfer & Purification

Includes	2 lube oil storage tanks and pumps
	1 lube oil centrifuge

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Startup Steam Generator Feedwater Pump

Quantity	1
Type	Multistage, horizontal
Capacity	.0252 m ³ /sec @ 24580 kPa head (400 GPM @ 3,565 psi head)
Motor Power	746 kW shaft (1000 HP)

Emergency Diesel Generator

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

No. 2 Oil Storage Tank

Quantity	1
Type	Aboveground
Dimensions	6.7 m dia x 5.5 m high (20 ft x 18 ft high)
Capacity	159 m ³ (1,000 barrels)
Plate Material	Carbon Steel
Plate Thickness	6.3 mm to 12.7 mm (1/4 in. to 1/2 in.)

Makeup Water Pump

Quantity	2
Type	Deepwell, centrifugal
Capacity	.0379 m ³ /sec @ 50 m (600 GPM @ 165 ft)
Motor Power	37 kW shaft (50 hp)

Auxiliary Closed Cooling Water Heat Exchangers

Quantity	2
Effective Surface	928 m ² (10,000 ft ²)
Material	Stainless Steel
Duty	12.92 x 60° kJ/hr (12.25 x 10° Btu/hr)
Type	Horizontal, U-tube

Auxiliary Closed Cooling Water Water Pumps

Quantity	2
Type	Horizontal, centrifugal
Capacity	.1578m ³ /sec @ 16.8m head (2,500 GPM @ 55 ft)
Motor Power	18,6 kW shaft (25 hp)

Service Water Pumps

Quantity	2
Capacity	.1262 m ³ /sec @ 76 m TDH (2,000 GPM @ 250 ft TDH)
Type	Vertical, wet pit turbine

TABLE 4.6-1

EPGS EQUIPMENT LIST (Continued)

Motor Power	149 kW shaft (200 hp)
<u>Critical Piping</u>	
Main Steam Piping	ASTM A335 P-11 318 mm (12.5 in) O.D. 65 mm (2.561 in) wall thickness Length, 61 m (200 ft) 102 mm (4 in) insulation
Hot Reheat	ASTM A335 P-11 521 mm (20.5 in) O.D. 20.4 mm (0.803 in) wall thickness Length, 61 m (200 ft) 102 mm (4 in) insulation
Cold Reheat	ASTM 106 Gr. B 610 mm (24 in O.D.) Schedule 30 Length, 6.1 m (20 ft) 64 mm (2.5 in) insulation
Feedwater	ASTM 106 Gr. C 168 mm (6.625 in) O.D. Schedule 120 Length, 76 m (250 ft) 51 mm (2 in) insulation
Circulating Water	Concrete 1.37 m (54 in) I.D. Length, 1462 m (4,800 ft)

TABLE 4.6-2
EPGS DATA LIST

A. Design Characteristics

1. Turbine-generator type and nameplate rating, MW

Tandem compound, two-flow with 49 mm (1.6 in) L.S.B., single reheat, two casing, 3600 RPM and rated at 106.2 MW gross unit output.

Hydrogen cooled generator, 3600 RPM, 0.9 power factor, 60 Hz, 3-phase output, 122,720 KVA max at maximum guaranteed load.

2. Turbine inlet conditions

Temperature, °K (°F)	811 (1000)
Pressure, MPa (psig)	16.5 (2400)
Flowrate, 10 ³ kg/hr (10 ³ lb/hr):	
Maximum	347.6 (765.2)
Design	314.3 (691.9)
Minimum	78.6 (173.0)

3. Turbine reheat conditions

Temperature, °K (°F)	811 (1000)
Pressure, MPa (psig)	2.6 (377)
Flowrate, 10 ³ kg/hr (10 ³ lb/hr):	
Maximum	311.8 (686.3)
Design	282.4 (621.6)
Minimum	67.7 (149.0)

4. Number and location of feedwater heating extractions

Seven stages: 3 HPS and 4 LPs

5. Condenser type and configuration

Single pressure/shell with divided water box, 2 pass	
Water flow rate, 10 ⁶ kg/hr (10 ⁶ lb/hr)	9.5 (21)
Water temperature in, °K (°F)	302 (84)
Water temperature out, °K (°F)	315 (108)
TTD, °K (°F)	4 (7)

6. Feedwater Pump Stages

14

7. Auxiliary Steam Supply,	9.1 (20)
10 ³ kg/hr (10 ³ lb/hr)	
@ rated pressure MPa (psig)	1.72 (250)

8. <u>Sealing Steam Requirements</u>	1,395 (3,075)
@ 25% load, kg/hr (lb/hr)	

9. EPGS Performance, Operation, and Control

TABLE 4.6-2

EPGS DATA LIST (Continued)

Refer to Section 4.6.3.

A.10 PUMP CHARACTERISTICS FOR ALL EPGS PUMPS

<u>Pump</u>	<u>Steam Feedwater</u>	<u>Condensate</u>	<u>Circulating Water</u>
Quantity	2	3	2
Type	Centrifugal Horizontal Multistage Split Case	Centrifugal Vertical Canned Suction	Centrifugal Vertical Wet Pit
Speed, rpm	3,600	1760	590
Head, m (ft)	2,207 (7,242)	156.7 (514)	14.3 (47)
NPSH, m (ft)	10.7 (35)	3 (10)	6.1 (20)
Oper temp, °C (°F)	147.8 (298)	46.1 (115)	28.9 (84)
Power, KWe (BHP) @ MGL	1,128 (1,390)	77 (94)	243 (300)
Capacity, m ³ /sec (GPM)	.04 (696)	.04 (582)	1.32 (21,000)

TABLE 4.6-2

EPGS DATA LIST (Continued)

A.10 PUMP CHARACTERISTICS FOR ALL EPGS PUMPS

<u>Pump</u>	<u>Service Water</u>	<u>Aux. Closed Cooling Water</u>	<u>Make-up Water</u>
Quantity	2	2	2
Type	Centrifugal Vertical Wet Pit	Centrifugal Horizontal	Centrifugal Vertical Wet Pit (Deepwell)
Speed, rpm	1770	1750	1760
Head, m (ft)	76.2 (250)	16.8 (55)	50.3 (165)
NPSH, m (ft)	6.1 (20)	6.1 (20)	3 (10)
Oper temp, °C °F	28.9 (298)	31 (115)	15.6 (84)
Power, KWe (BHP) @ MGL	126 (155)	37 (45)	27 (33.3)
Capacity, m ³ /sec (GPM)	0.13 (2,000)	0.16 (2,500)	0.04 (600)

TABLE 4.6-2

EPGS DATA LIST (Continued)

A.10 PUMP CHARACTERISTICS FOR ALL EPGS PUMPS

<u>Pump</u>	<u>S.G.</u>	<u>Firewater</u>		
	<u>Startup</u> <u>Feedwater</u>	<u>Diesel</u> <u>Drive</u>	<u>Main</u>	<u>Jockey</u>
Quantity	1	1	1	1
Type	Centrifugal Horizontal Multistage	Centrifugal Vertical Wet Pit	Centrifugal Vertical Wet Pit	Centrifugal Vertical Wet Pit
Speed, RPM	3,600	1,760	1,760	1,760
Head, m (ft)	2,642 (8,669)	107 (350)	107 (350)	49 (160)
NPSH, m (ft)	4.6 (15)	6.1 (20)	6.1 (20)	6.1 (20)
Oper temp, °C (°F)	114 (237)	18.9 (84)	28.9 (84)	28.9 (84)
Power, kWe (BHP) @ MGL	784 (967)	187 (230)	187 (230)	4 (5)
Capacity, m ³ /sec (GPM)	.02 (372)	0.13 (2,000)	0.13 (2,000)	.003 (50)

TABLE 4.6-2

EPGS DATA LIST (Continued)

A.11 CYCLE PARAMETERS

<u>Load</u>	<u>VWO + 5% O.P.</u>	<u>100% (MGL)</u>	<u>100% (MGL)</u>	<u>25%</u>
Time of Day	Noon	Noon	Evening	Evening
Gross Turbine Cycle Efficiency, %	42.4	42.3	42.3	36.5
Net Turbine Cycle Efficiency, %	40.2	40.0	40.4	32.5
Gross Cycle heat, rate, kJ/kW-hr (Btu/kw-hr)	8492 (8049)	8516 (8072)	8516 (8072)	9876 (9361)
Net Cycle heat rate, kJ/kW-hr (Btu/kW-hr)	8943 (8477)	8995 (8526)	8909 (8445)	11,084 (10,506)
Turbine exhaust pressure, mm (in) Hg	76.2 (3)	76.2 (3)	76.2 (3)	76.2 (3)
Feedwater temperature, °C (°F)	238 (460)	232 (450)	232 (450)	166 (331)
Net Unit Output, kWe	110,310	100,548	101,505	23,117
Auxiliary Loads, kWe	5,868	5,651	4,694	2,828
Gross Unit Output, kWe	116,178	106,199	106,199	25,945

4.6.2.2 Turbine-Generator

The turbine is a 3600 rpm tandem-compound unit rated at 106.2 MWe gross electrical output. At maximum guaranteed load, the throttle steam conditions are 16,547 kPa/811K (2400 psig/1000F) and the reheat steam conditions are 2600 kPa/811K (377 psig/1000F). The low pressure turbine exhausts to a 3 in Hg (absolute) condenser backpressure. The high pressure, intermediate pressure, and low pressure sections supply extraction steam for seven stages of feedwater heating as well as steam for the plant auxiliaries. The turbine has an electro-hydraulic control system and operates with constant throttle pressure over the load range. The EPGS conceptual design is based on use of a General Electric two casing machine with a combined high pressure-intermediate pressure casing. The low pressure turbine is a two flow machine with 419 mm (16.5 in) last stage blades. The turbine is provided with the necessary instrumentation connections for an ASME performance test.

4.6.2.3 EPGS Cycle Equipment

Feedwater Heaters. Feedwater is heated in seven stages with one direct contact deaerating heater and six stages of closed feedwater heating. Three stages of low pressure (LP) heating are located between the condenser and the deaerator. They are installed in the condenser steam dome. The first stage heater is provided with an individual bypass that is opened when operating at low plant loads. Three single shell horizontal high pressure

(HP) heaters complete with individual bypass are located in the feedwater cycle between the feedwater pumps and the steam generator.

Extraction Steam and Water Drains. The HP feedwater heaters receive extraction steam from the cold reheat line and from the intermediate pressure turbine. Normally, the drains from the HP heaters cascade to the deaerator. Each heater is also provided with an individual dump line to the condenser.

The deaerator and LP feedwater heaters take extraction steam from the LP turbine. The three LP feedwater heaters are installed in the condenser steam dome to minimize extraction steam pipe runs and to conserve space. During 25 percent load operation, cold reheat steam is supplied to the deaerator to maintain positive (above atmospheric) pressure and the lowest LP feedwater heater is valved out of service.

Bleeder trip valves and block valves, in the extraction lines prevent turbine overspeed, when the turbine is tripped, and protect the turbine from damage due to water induction as required by the turbine manufacturer.

Feedwater Pumps. Two half-capacity feedwater pumps take suction from the deaerator storage tank and supply feedwater to the three high pressure feedwater heaters. The feedwater pumps are horizontal, multistage, barrel type units and are motor driven. Since the EPGS must start-up and shutdown daily, an auxiliary

motor-driven horizontal multistage pump, sized at approximately 1/4 unit rated flow, is provided to supply feedwater during start-up.

Condensate Pumps. Three one-third capacity vertical multistage canned type condensate pumps circulate condensate from the hotwell, through the condensate polishers, the gland steam condensers, the air ejector inter- and after-condensers, the LP feedwater heaters and to the deaerating heater. In the event one of the pumps fails, the other two pumps are capable of meeting system requirements. Minimum flow for the condensate pump and air ejector inter- and after- condenser is maintained by recirculation control valves.

Condensate Polishers. An in-line mixed bed condensate polisher removes dissolved and suspended solids from the condensate and maintains feedwater chemistry within allowable limits. It is sized to permit continued EPGS operation when there are small condenser tube leaks. Three polisher vessels, each sized for 50 percent maximum continuous condensate flow, are provided together with four resin charges (two in service, one on standby, one in regeneration). A complete system for automatic regeneration of exhausted resins is also provided.

Condenser and Auxiliaries. A single shell single pressure condenser with divided waterboxes, admiralty tubes and a deaerating hot well was selected for the conceptual design. The condenser hot well level is controlled by makeup and reject

valves in piping connected to the demineralized water storage tank. A full capacity steam driven hogging ejector is provided for establishing vacuum and a separate two-stage twin element steam driven air ejector, with inter-condenser and after-condenser, is provided for holding service.

4.6.2.4 Heat Rejection System

Circulating Water Loop. The circulating water loop carries EPGS waste heat from the condenser to the cooling tower where it is rejected to the atmosphere. It consists of a bank of pumps, concrete conduit, the condenser, and the cooling tower and basin. Two half capacity vertical wet pit mixed flow pumps circulate water from the cooling tower basin to the condenser and back to the cooling tower. Motor operated butterfly valves are located at each pump discharge port and at the inlets and outlet to the condenser.

Cooling Tower. A mechanical draft cooling tower with four 15.5 m (51 ft) rectangular concrete cells was selected for the conceptual design. Each cell contains a 8.5 m (28 ft) diameter fan. The tower size is based on 100% MGL heat balance heat rejection, 297K (74F) wet bulb temperature, 3 inch Hg condenser pressure, a 5.6C (10F) approach temperature and a 13.3C (24F) range.

Cooled water from the tower enters the tower basin where it supplies the circulating water pumps, the service water pumps and

the fire water pumps. Fixed screens prevent debris from entering the pumps. Equipment is provided for adding sulfuric acid to the basin water as required for pH adjustment and the prevention of calcium carbonate scaling. Chlorine solution is added as required for algae control.

4.6.2.5 EPGS Auxiliaries

Compressed Air System. Instrument and service air is provided by two full capacity centrifugal compressors, each with an associated moisture separator and aftercooler. The compressors discharge to a common receiver, from which separate headers supply the service and instrument air systems. The instrument air system includes two automatic desiccant type air dryers operating on an eight hour cycle.

Lube Oil System. Lube oil is stored in two tanks, one for clean and one for dirty oil. Each tank has sufficient capacity to provide a complete oil change for one turbine generator. Two rotary pumps are provided to unload and transfer oil between tanks and the turbine reservoirs. A centrifuge purifies dirty oil stored in the storage tank and in the main turbine lube oil reservoir.

Auxiliary Closed Cooling Water. An auxiliary closed loop cooling water system serves to carry heat from the hydrogen coolers, the generator stator coolers, the exciter coolers, the turbine lube oil coolers, the bearing lube oil coolers, and the main chillers

to heat exchangers in the circulating water loop. Two full capacity horizontal centrifugal pumps circulate buffered condensate. A head tank accommodates thermal expansion and contraction of the coolant as well as any surging in the system.

Raw Water. Raw water is assumed to be available from a deep well at the plant site. Two half-capacity vertical wet pit turbine pumps are provided to pump water through a single pipe to the cooling tower basin for makeup. The same pumps supply the plant makeup water treatment system.

Service Water. Two full capacity vertical wet pit turbine type pumps provide service water to plant auxiliary components and the auxiliary cooling water heat exchangers.

Auxiliary Steam. Auxiliary steam is supplied as required to the building heating system, to the deaerator, to seal turbine glands, and to the air ejector for hogging. During normal plant operation, this steam is taken from the steam cycle. When this source is unavailable, steam to the process systems only is supplied as required from an auxiliary oil-fired boiler.

Emergency Power. Power required for start-up is normally provided from the grid to which the plant is connected. A diesel generator is provided for supplying power for emergency shutdown and start-up of the unit under plant blackout conditions (no power available from the grid). A detailed accounting of minimum auxiliary power required for black startup and shutdown resulted

in the specification of a 2 MWe diesel generator capacity. The diesel generator also provides stand-by power for defocussing the heat receivers during a loss of power emergency.

Fuel Oil System. No. 2 fuel oil is provided for the ESS tank oil heaters, the auxiliary boiler, the fire pump diesel driver, and the emergency diesel generator. One rotary pump is provided to unload oil from tank trucks. The oil is stored in an above ground outdoor storage tank with a 159 m³ (42,000 gal) capacity. Three half capacity rotary pumps, taking suction from the tank, deliver oil to various services.

Fire Protection System. The fire protection system is installed in accordance with rules and regulations of the National Fire Protection Association. All hoses, nozzles, and cabinets are labeled or listed by the Underwriters Laboratories, Inc. A plant fire loop and a turbine building fire loop are provided. Firefighting water is taken from the cooling tower basin by a motor-driven vertical turbine pump. A smaller vertical motor-driven jockey pump replenishes losses from leaks and maintains pressure when the system is not in use. The motor-driven fire pump is backed up by a diesel-driven horizontal centrifugal pump.

Automatic fixed spray systems, hose connections and hydrants are installed in areas with the potential for fires. Portable extinguishers, including pressurized water, carbon dioxide and dry chemical are provided as required.

4.6.2.6 Electric Plant Equipment

The steam turbine-generator is rated 106.2 MW, 0.9 PF, 13.8 kV, 3 phase, 60 Hz. The main power transformer steps up the voltage to connect with the utility grid system. A main auxiliary transformer supplies indoor metal-clad switchgear, rated at 250 MVA, 4160 volts and 3000 amperes, to power the plant auxiliary equipment. Plant startup power is provided from a reserve auxiliary transformer. A double ended 3000 kVA load center substation provides the necessary power for plant utilization at 480 volts.

A diesel generator set rated at 2000 kW provides stand-by power for defocusing the receivers under emergency conditions or loss of normal power. Multiplexed control prevent operating all drives instantaneously, thus assuring that the capacity of the diesel generator set is adequate.

4.6.3 EPGS Performance Operation and Control

4.6.3.1 EPGS Performance

Gross Electric Power Output. EPGS heat balances were computed for 100% maximum guaranteed load, for 25% of MGL throttle flow and for valves-wide-open 5% overpressure operating conditions. These heat balances, corresponding generally to minimum, design

point and maximum operation for the EPGs, are presented in Section 4.6.2.1. The gross electric output of the EPGs is:

<u>25% MGL</u>	<u>100% MGL</u>	<u>VWP + 5% OP</u>
25,945 kWe	106,199 kWe	116,178kWe

The total auxiliary power for the plant must be subtracted from the gross electric output in order to obtain the net electric output of the plant.

Auxiliary Power. The auxiliary power required for power plant operation varies with the operating modes of the plant. While EPGs auxiliary power varies with electric output, HTS auxiliary power variation is chiefly due to the power demand of the field salt pumps which change with the time of day. HTS field pump salt flow requirements vary with solar insolation from maximum flow at noon to a minimum flow in the morning and afternoon when HTS field flow is started and shut down. In principle, the minimum field flow which could generate 839K (1050F) hot salt is zero flow. In practice, the minimum flow at which 839K (1050F) salt generation begins during the Heat Receiver Subsystem start transient (discussed in Section 4.3) is approximately 40 percent of the maximum (noon) flow capacity. The minimum HTS field flow during evening shutdown is somewhat less than this. At the end of the operating day, solar insolation is declining rapidly and shutdown of the HRS begins shortly before the flow of 839K (1050F) salt would have dropped to zero. For the purpose of estimating the minimum power required to drive the field pumps, minimum HTS field flow is conservatively estimated to be one-

third of the maximum noon flow rate. The field pump power reported in Table 4.6-3 for morning/afternoon operation is based on this assumption.

Table 4.6-4 shows the influence of morning, noon, and nighttime HTS operation on the plant auxiliary load requirements associated with EPGS operation at the 25% MGL, 100% MGL and VWO + 5% OP heat balances. The resulting net electric output of the power plant and the corresponding EPGS heat rates are also shown. The table shows the effect of shutdown HTS field pumps on nighttime operation from thermal storage. It also shows the variation in plant performance with time of day. Due to variations in auxiliary power for HTS pumping, net EPGS power output is least at noon and is greatest for morning and evening operation.

TABLE 4.6-3

PLANT AUXILIARY LOADS, kWe

EPGS Mode -	25% MGL	100% MGL	VW05%OP
HTS Mode -	<u>Morning/Afternoon</u>	<u>Noon</u>	<u>Noon</u>
HTS			
Field Pumps	131	1011	1011
Steam Generator Pumps	90	366	420
HRS			
Receiver Motors	194	50	50
control Valves	100	100	100
MC			
Master Control	100	100	100
EPGS			
Boiler Feed Pumps	1240	2255	2377
Condensate Pumps	84	153	161
Cooling Tower Fans	240	437	461
Circulating Water Pumps	260	472	497
Transformer/ HVAC	284	517	545
Miscellaneous	<u>105</u>	<u>190</u>	<u>200</u>
TOTAL	2828	5651	5922

TABLE 4.6-4

PLANT PERFORMANCE SUMMARY

<u>EPGS Heat Balance</u>	<u>HTS Operating Mode</u>		
	Morning 33% Field <u>Flow</u>	Noon 100% Field <u>Flow</u>	Night 0% Field <u>Flow</u>
	<u>Plant Auxiliary Load, kWe</u>		
25% MGL	2828*	3840	2883
100% MGL	4639	5651*	4694
VWO + 5% OP	4856	5868*	4911
	<u>Net Plant Electric Output, kWe</u>		
25% MGL	23,117*	22,105	23,062
100% MGL	101,560	100,548*	101,505
VWO + 5% OP	111,322	110,310*	111,267
	<u>Net Heat Rate, kJ/kWh (Btu/kWh)</u>		
25% MGL	11,084 (10506) *	11,591 (10987)	11,110 (10531)
100% MGL	8905 (8441)	8995 (8526) *	8909 (8445)
VWO + 5% OP	8862 (8400)	8943 (8477) *	8866 (8404)

* Values used for heat balances in Section 4.6.2.1

Conventional power plant controls are required for operation of the EPGs. These consist of extensive local and control board instrumentation for monitoring the status of critical equipment, and alarms and turbine trip provisions to guarantee operational safety and prevent damage to hardware. The major control affecting turbine-generator, steam generator, and feedwater system operation is the three element feedwater control. Its function is to control feedwater flow as required to satisfy the turbine steam demand without incurring excessive variation in steam drum water level. EPGs load indicators, such as the turbine admission valve stem position and steam drum pressure are monitored by the Master Control Subsystem for use in controlling steam generator salt flow.

Controls dealing with the salt side of the steam generator are discussed in Section 4.4.3.

Cycling Operation. The EPGs operating day varies, with time of the year, from near 16 hours in the summer to less than 12 hours in the winter. It follows that the turbine is shutdown for a period of 8 to 12 hours each night. During periods of inclement weather and shutdown for maintenance, the turbine shutdown periods are longer. Therefore, the turbine inner shell

temperature at startup varies from ambient to approximately 100K (180F) below the 811K (1000F) admission steam temperature.

Thermal fatigue limits for turbines exposed to many start and shutdown cycles require restrictions on the time required to bring a turbine to full operating temperature, speed and load. The extent of the start-up restrictions vary considerably with the turbine design. Applications involving cycling operation of peaking units, where it is necessary to come on line very quickly, require turbines designed to accommodate rapid heating without overstressing. Some machines of this type are designed to be brought to full capacity operation in less than 15 minutes. Their selection involves additional turbine expense and some decrease in full load performance.

The EPGS turbine for this conceptual design does not require such a rapid start-up. Although its operation is cyclic, it operates much as an intermediate loaded plant; in that it comes onto the grid whenever it has power to deliver, rather than in response to quickly arising peak loads.

The turbine cooldown information for double-shell turbines, such as the General Electric machine selected for the conceptual design, indicates that normal turbine shutdown followed by 8 to 12 hours of turbine cooldown lowers turbine case temperature by 145C to 160C (260F to 290F). GE cyclic life information indicates that under those conditions a start-up time of 1.6 hours will not impair the 30-year turbine life. This is an

acceptable restraint for the routine daily start-up of this plant. It is not quite as long as the time required to bring the Heat Receiver Subsystem to full operating temperature; as discussed in Section 4.3. Greater start-up time is provided as required for start-up of a cold turbine after prolonged shutdown.

Cold Startup. Cold startup occurs after a prolonged shutdown period when the high pressure turbine case is nearly at room temperature.

The initial state of the steam generator, its standby state whenever the EPGS is shutdown, is the 567K (560F) condition in which the steam drum, boilers and preheaters are bedded down with 561K (550F), 7100 kPa (1030 psig) water and in which 561K (550F) salt is circulated through the superheaters, reheater, boilers, and preheaters. With the steam generator conditioned at 561K (550F), the following events are initiated:

- Start auxiliary steam supply
- Start turbine turning gear
- Start condenser vacuum
- Admit auxiliary steam to turbine seals
- Open main steam valve to preheat main steam line
- Start minimum feedwater flow
- Peg deaerator with auxiliary steam
- Recirculate heated feedwater to the condenser
- Start steam generator recirculation/heater loop (discussed in Section 4.4.3) to heat drum water to 589K (600F).

The purpose of heating the drum water is to prevent boiling of the 561K (550F) preheater water when steam flow is subsequently initiated. With the main steam line preheated to 589K (600F), with feedwater available at approximately 394K (250F), with 589K (600F) water in the steam drum and with 567K (560F) water in the boilers and preheaters, the following events are initiated:

- Open feedwater valve to the steam generator
- Establish steam generator flow by initiating a small bypass flow (5% - 10%) of main steam to the condenser
- Ramp salt supply temperature upward to 644K (700F) by blending flow from the warm and hot salt tanks.

When the above is completed:

- Open turbine admission valve
- Bring turbine up to speed per turbine suppliers instructions
- Continue ramping salt supply temperature at approximately 83K (150F) per hour as required to give desired turbine case heating rate
- Terminate main steam bypass to condenser
- Subsequently apply additional turbine load as required to bring the turbine case to normal operating temperature.

The cold start described above requires approximately 5 hours, following an initial 2 hour preheat period on turning gear.

The control of salt flow rate during this transient, as during normal operation, is attained by modulating salt pump speed in response to speed demand signals generated by the Master Control

Subsystem. The MCS computes pump speed demand as a function of turbine load and main steam pressure. A low temperature override prevents running at exit salt temperatures below 533K (500F).

Warm Startup. Warm startup applies when initial temperature of the high pressure turbine case is in the range of 422K (300F) to 644K (700F). The start procedure is the same as for cold start-up with the following changes.

As main steam bypass flow to the condenser is initiated the salt supply temperature is ramped upward, by blending of flow from the warm and hot salt tanks, to a temperature between 644K (700F) and 839K (1050F) as required to match steam temperature with turbine case temperature when the turbine admission valve is subsequently opened. Warm start-ups require approximately 2 to 5 hours, depending on turbine case temperature. No prior preheat on turning gear is necessary.

Hot Startup. Hot startup applies when initial temperature of the high pressure turbine case is above 644K (700F). Typically after a normal shutdown and 8 to 12 hours of overnight cooldown the turbine case temperature is at 650K (710F) to 667K (740F). The start-up is the same as for a cold start with the exception that when main steam bypass to the condenser is initiated, the salt supply temperature is ramped upward to its full rated temperature of 839K (1050F). This matches steam temperature to turbine case temperature when the turbine admission valve is subsequently opened. Hot start-up is completed in approximately 1.6 hours.

Shutdown. A programmed shutdown is achieved by reducing load at a rate that produces a turbine case cooldown rate of approximately 83C (150F) per hour. When the load is reduced to 3 to 5 MWe, the turbine is tripped automatically.

At this time the salt supply temperature is 839K (1050F) and ramping downward to the 567K (560F), by blending flow from the warm and hot salt tanks, is initiated. The progressive lowering of salt supply temperature eventually cools the entire steam generator to 567K (560F) and transports residual heat from the steam generator to the warm salt storage tank. Following turbine trip, however, bypass of some steam to the condenser is required to limit the steam generator pressure transient. The steam bypass can be terminated when salt supply temperature has dropped below approximately 617K (650F).

The questions of size of the required steam bypass, the resulting steam generator pressure and temperature transients, and maximum allowable ramp rates of the supply salt are all reasonable subjects for future investigation.

4.7 MASTER CONTROL SUBSYSTEM (MCS)

4.7.1 MCS Requirements

The MCS is comprised of individual subsystem control (ISC), the master control function (MCF) or integrating controls, and the data acquisition system (DAS). The ISC systems provide control of subsystem operation such that the system level function demand is met. The MCF monitors ISC functions and operational demands, providing the intersystem control functions and setpoints for ISC operation.

The operation of the LFSCPS will be performed by the plant operators as in a conventional fossil-fuel power plant. The MCS has been designed to enhance and aid these operations by reduction of burdensome routing operations and promotion of human skills of judgment, deductive reasoning, and short-term efficiency of tactile/visual manipulation.

The following sections describe MCS control functions. The basis for selection of the type of control and its implementation is discussed in Section 2.6. Also described are the receiver, storage, and steam generation control instrumentation requirements (non-conventional portion of plant) and the DAS philosophy.

In addition to providing the coordinating control and setpoints for subsystem control, the MCF of the MCS must:

1. Provide the calculations required to assure that operator set storage/generation requirements are met.
2. Provide protection actions and/or alarms to prevent equipment damage.
3. Provide setpoints and demand ramps to aid in operator and automatically initiated startup/shutdown actions.

4. Provide reference "exception" and demand data displays for the plant operators.
5. Provide data acquisition of summary and "exception" form to reduce data load.

4.7.2 MCS Design

A simplified schematic of the MCS is shown in Figure 4.7-1. The schematic is broken into three major subdivisions to simplify the discussion and to match the configuration/implementation discussed in Section 2.6. The subdivisions are the "collector controls" on the left; the "EPGS controls" on the right; and the "integrating controls" in the center, which control storage and salt flows, provide protective actions, and provide setpoints for the collector and EPGS system operation. Many minor subsystems, support systems, and peripheral controls are not indicated in the simplified schematic since they tend to be standard and are not directly involved in the unique aspects of the LFSCPS control. The MCS provisions are described in this section and the collector and EPGS subsystem controls are described in Section 4.7.3.

Most large fossil and nuclear plants operate in a boiler-follow-turbine (BFT) mode. Owing to the very sluggish response of a turbine-follow-boiler (TFB) mode, with these plants the BFT mode, though less stable than the TFB mode, is normally used. New designs often incorporate a mixed mode with the BFT mode dominant. The unusual character of the LFSCPS operation from storage presents (for short-term changes) the equivalent of an infinite source, rapid response boiler. A mixed mode with the TFB mode dominant has therefore been chosen for the reference LFSCPS control.

Normal Load Range Operation. The MCS controls the salt outlet temperature from the collector field by a temperature control trim of the field pump* flow. A gross reference flow demand is provided to the pump by a signal derived from the normal solar insolation measured by one or more of the

*Pump refers to the pump set.

Fig. 4.7-1. Simplified schematic of master control system

four solar position indicating units (SPIUs) of the collector subsystem. The flow to the transport pipes from the field pumps is distributed, by the collector subsystem, through the various receivers by the row inlet valves as discussed in Section 4.7.3; is re-collected in the hot transport pipes; and is mixed and transported to the "storage-generation control valve set" (SGCVS). Temperature at the SGCVS is fed back to a temperature controller whose output signal is used to trim the pump flow demand. Alternative "pump head" control (versus flow) is provided by measurement of pressure rise across the pump for conditions where this mode of control may provide the most stable flow distribution.

Steam generator (SG) pump speed is controlled to control the SG salt flow. The nominal speed is set as a function of load demand and is trimmed by a pressure error signal from the main steam pressure control. The load demand is set by the load/storage calculation base on operator setup or is directly set by the operator. The SG pump speed signal is fed forward to the EPGS as a load index. This basic TFB mode of control is actually modified to a mixed mode by a fractional load signal to the EPGS to aid in responses.

The storage operation logic will normally control the SGCVS (valve C) such that all the salt from the collector subsystem is delivered to the hot storage tank. Storage versus generation is, therefore, simply controlled by the difference in collector and SG flow.

The return salt temperature is sent to the individual group controllers as a reference for the two temperature difference controllers described in Section 4.7-3.

Protective Functions. Protective functions include hot tank level, warm tank level, hot tank inlet temperature, and warm tank inlet temperature. A hot tank level that is too high initially alarms the operators to increase load and at a second level initiates collector system defocus and flow reduction. A warm tank level that is too low follows the same procedure.

A hot tank inlet temperature that is too high initiates collector defocus (pumping must already be maximum or limited since normal control prevents this condition). A warm tank inlet temperature that is too low (which should not occur under normal load conditions) initiates SG salt flow increases; if this problem is due to low feedwater temperature, EPGS shutdown is initiated and plant securing is started. Excessive pressure in the collection system will result in pump flow cutback.* Excessive pressure in the hot tank will initially result in cover gas vent and ultimately defocus and cutback of collection.* Excessive warm tank pressure will initially result in cover gas vent and ultimately load cutback.*

Additional subsystem protective functions are discussed in the next section.

Startup Operation. Initial operator action is taken to condition the field piping and initiate SG and turbine warmup. The startup conditioning control then:

1. Monitors the SGCVS inlet (collector system outlet) temperature, warm tank temperature, hot tank temperature, and reheat and superheater temperatures.
2. Controls hot and warm tank pump flow for initial warmup of the SG through control of valves A and C and the two pump speeds.
3. Signals the reheat temperature control to adjust valve E to equalize reheat and superheat temperatures.
4. Switches off valves A and D and cuts off the warm tank pump flow. Continues the hot tank pump flow for continued SG conditioning.
5. On initiation of collection (after heat transport system piping thermal conditioning), valve F is opened, valve C is closed, and the warm tank pump flow is started through the collector system.
6. As collector system temperature is rising, the salt is delivered to the warm tank; then as the temperature approaches the hot tank

*Pressure relief will also be provided to code requirements for these cases. However, the provisions indicated should prevent reaching the relief set-point.

temperature, valve F is gradually closed as valve C is opened to the hot tank/SG pump.

7. Control is released from startup conditioning to the normal control.

Rolling of the turbine, synchronization, loading, and initiation of automatic load range control will be performed by the operators in a time frame controlled largely by the amount of residual hot storage. The timing would nominally be such that the stored energy from the previous day would supplement collection to meet the rise to design load in a manner equivalent to "direct operation from collection starting with hot tank empty." This strategy will result in the best utilization of storage on maximum-energy days and will be supported by display by the MCS to the operators of calculated stored energy and expected collection energy and time.

It is during startup operations that the protective functions previously noted are anticipated, on some occasions, to be needed. Sequential errors or mismatch of conditions may more readily occur under the transient conditions and numerous activities in progress at such times.

Shutdown Operation. Shutdown control is not specifically illustrated in the simplified schematic of Figure 4.7-1. However, some of the major elements of the shutdown process use the load range controls previously described. The sequence of normal shutdown is:

1. Collected energy has reduced to a point where the flow at hot tank outlet temperature has reached a preset minimum.
2. As the collector return temperature starts to drop, the outlet temperature is started on a ramp-down and valve C is gradually closed while valve F is gradually opened to divert the collector flow to the warm tank.
3. When the collector return temperature reaches a preset level, the field pumps are stopped and drain and isolation are initiated.

4.7.3 Subsystem Controls

The two major subsystems are the EPGS and collector system. Several other minor subsystems, such as water treatment, fire control, salt purification, and hydraulic systems, are not specifically discussed here owing to the standard character of these subsystems and the lack of direct interaction with the unique aspects of the LFSCPS control.

EPGS. EPGS controls, beyond the establishment of steam conditions, are standard and are described in Section 4.6.3.2. Stop/admission valve and exciter controls provide the turbine/SG control. Condenser/condensate controls via the circulating water and condensate pumps are typical. The three-mode controller for the feedwater pump/valve is standard with only the load index being slightly unusual due to the TFB-dominant mixed mode control.

Collector Subsystem. The collector subsystem control has been distributed throughout the field in the form of "group controller" microprocessor process control units and four SPIUs. Each group controller provides the control of seven or eight collector rows. The group controllers accomplish identical functions and perform a replicated function for each collector row. The description which follows will therefore be in terms of control of one collector row.

The row receiver is positioned to the approximate location of the concentrator focal line by one of three methods:

1. Position demand from the central unit based on calculated solar position.
2. Measured solar position from one or more of the SPIUs.
3. A "locate" mode where the high-speed positioning (stow motor) is commanded to run the receiver from a position on one side of the expected image to the other side. When one or more of the solar

cells (three pairs, with the cells of each pair on opposite sides of the receiver) enter the beam, the solar cell "null seeking" mode takes over.

Figure 4.7-2 illustrates the position control and null seeking control obtained by motor drive of the receiver assembly. The position mode, as noted, brings the left or right photocell into the concentrated beam, and the difference in signal from the cells is amplified by the difference amplifier control compensation (DACC) to create a position demand bias. At this time the track and hold function which has been tracking (-2 times) solar angle is placed in hold, leaving the system on photocell null seeking control.

For the sweep mode of placing the receiver in the beam, SW1 is switched to "angle reference control" and the motor controller is placed in the high-speed mode. When the DACC transition noted above occurs, both the track and hold switch and a switch to low-speed motor control result.

When a "stow" command is received from the central unit, position demand is switched to a local reference stow position demand and high-speed motor control is switched in at the motor controller. As the stow position is approached, the "stow proximities switch" signals the motor controller to the slow motion and the stow stops are gently approached.

Failure of the local group controller or central command places SW2 into remote, allowing direct motor control from the control center.

Local temperature difference control (TDC) is provided by adjustment of the row inlet control valve. Under collection conditions (hot fluid from the field has reached the tank and/or SG), this controller acts to trim the flow through the row to match (with fixed offset) the temperature reference signal from the central system, which is a measure of the bulk collector system return temperature. This controller, therefore, acts to maintain

Fig. 4.7-2. Simplified schematic of receiver position control

each row outlet temperature uniform with those of all other rows while the central system field pump control maintains the desired bulk outlet temperature.

During startup, the TDC is disabled until the row outlet temperature sensor has been flooded (sensible reading). Subsequent to this time and until the bulk outlet fluid has reached a stable (though rising in temperature) condition, the TDC acts to control outlet temperature to a startup reference from the central unit. The local TDC does not know the difference between this reference and the reference based on bulk outlet temperature measurement and therefore reacts the same from the time it is enabled, throughout the rest of the collection period. The central unit ensures a smooth transition from startup to bulk temperature reference.

Defocus control is provided to locally defocus the row receiver in the event that one of the row temperatures exceeds set limits. Three receiver pipe skin temperatures and two fluid temperatures are monitored. The central system defocus command is also accepted, and this function acts through the position/null-seeking control as noted previously.

The two fluid temperature measurements are also compared with "minimum levels." If a minimum is reached, a signal is transmitted to the central system, activating drain and isolation of the appropriate portion of the field.

Data for the rows of each group controller are monitored by the group controller, and data are transmitted under two distinct conditions:

1. Intermittent (summary) data for the main parameters are updated at preset intervals.
2. Under abnormal conditions, data pertaining to the deviation and and exception interrupt are transmitted to the central unit.

Data Processing. The central unit will process the exception interrupt and transmitted data from any subsystem controller and generate the appropriate alarms/displays to the operators. In addition, the exception data will be directly placed in nonvolatile data storage. This will provide assured retrieval of data for deviant conditions.

The summary data from subsystems will be further processed, updating direct and demand display information. A volatile data storage of approximately 15 min will be maintained of all summary data, and at selected intervals a second level of summary data will be made by the central data processing which will be placed on permanent storage. This arrangement will:

1. Provide major condensation of routine data, which will be voluminous.
2. Allow high-speed dump to recoverable storage of a major malfunction from its inception plus the previous 15 min, with summary and exception data without second level summary data compaction.

5. PERFORMANCE AND COST ESTIMATES

5.1 INTRODUCTION

The performance of the as-designed collector field was computed by GA, and the performance of the as-designed HTS, ESS, and EPGS was calculated by Bechtel National. Thus, the performance data presented here are consistent with the design presented in Section 4. It is noted that the reported performance of the plant is different from that given in Ref. 5.1-1 primarily because more conservative properties have been adopted for the solar selective coating on the receiver heat duct.

A word of caution should be expressed regarding the plant performance estimates, which are point equilibrium data. Accurate performance estimates require that cumulative energy accounting be performed. Operationally the plant is subdivided at the ESS, since the collector field supplies as much hot salt to the ESS as it can and the EPGS draws off enough hot salt to generate steam sufficient to support the load demand. Although the equipment provides for direct input of hot salt from the field to the steam generator, removing the ESS from the loop and exposing the steam generator to potential temperature transients is undesirable. The temperature of the hot salt coming into the ESS depends on the set point temperatures at the end of the receiver and the flow rate (since the thermal losses in the piping are roughly constant irrespective of flow). The temperature of the salt in the hot tank depends on the temperature history of accumulated inflows and the ESS salt inventory since the thermal losses from the storage tanks are approximately constant. The temperature of the hot salt into the steam generator is a function of the salt flow (since the thermal losses in the storage piping are roughly constant) and the source (hot tank) salt temperature.

The significance of the preceding discussion is that the hot and warm tank salt temperatures depend strongly on what went in and out of the tanks and on the time of the charging and discharging. Variations in these salt temperatures affect the steam generator salt flow and pumping power and the steam side temperature, thus affecting plant electric output. Consequently, it is necessary to keep a running record of all ESS inflow and outflows so that plant output can be determined. This energy accounting has not been performed because it is more complex than is appropriate for the conceptual design level of the plant. The point to be emphasized is that without such accounting, the performance of the plant over a typical day of startup, shutdown, variable power demand, and possibly partial cloud cover cannot be readily determined with precision, while the heat and mass balances given in Section 5.2 pertain to equilibrium conditions.

The capital cost estimate of the plant has been prepared by Bechtel National, which has brought a large historical background of cost information to bear on the system. An extensive justification for the cost factors employed in the estimate is presented in Section 5.3. The cost of the receiver subsystem fabrication was prepared by GA as well as an estimate of the field installation labor. The concentrator subsystem cost estimate in terms of subcontracted material cost and fabrication/assembly labor was also prepared by GA. These costs were then translated into fully burdened costs including field indirects, engineering costs, contingency, and fee by Bechtel National. The resulting capital cost of the first plant is considered to be realistic and achievable for a first commercial plant following the construction and operation of a prior demonstration system of some magnitude. The cost estimate in essence assumes that a large-scale plant [up to 25 MW(t)] has been operated previously so that the fabrication and assembly technology is established and enough information is available to prepare a detailed and accurate construction plan and schedule.

REFERENCE

- 5.1-1. Schuster, J. R., et al., "Line Focus Solar Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.

5.2 PERFORMANCE ESTIMATES

This section presents performance estimates, including waterfall efficiency charts, for key days of the year and heat and mass balances for the appropriate energy modes.

Figure 5.2-1 gives the energy loss train for March 21 and December 21, both the noon value and the day average. The highest system efficiency (light to electricity) occurs at noon on March 21 and has a value of 23.8%. Figure 5.2-2 gives the June 21 and year-round average data, the latter having an overall efficiency of 12.9%. Additional details of the concentrator and receiver loss train are presented in Section 3. Operation from storage only is also shown in Figure 5.2-2. The storage efficiency (without penalty for shutdown/startup conditioning of the steam generator) is 97.9%.

Figure 5.2-3 provides the heat and mass balance for case (a), where the concentrator and receiver subsystems (CS-RS) drives the EPGS directly at design power. Note that in this mode, which is unlikely to be utilized in practice, the system parameters are not burdened with the energy storage thermal losses. In this mode, the output exceeds the rated output of 100.5 MW(e) net because the field piping pumping power is less than the peak value.

Figure 5.2-4 provides the heat and mass balance for case (b), where the CS-RS drives the EPGS and charges the ESS, evaluated at equinox noon, the time at which the maximum heat collection rate occurs. It is noted that the maximum salt pumping power in the field piping is only 1.01 MW(e). Plant net output is the rated value of 100.5 MW(e). It is also significant that since the steam generator salt side temperatures are kept constant, the different flow rates between case (b) and case (a) result in 5.7 K (10.3°F) less temperature rise across the collector field for case (b).

Figure 5.2-5 gives the heat and mass balances for case (c), where the CS-RS drives the EPGS when there is a low incident flux such as early in the morning or late afternoon. A collected output of 150 MW(t) is assumed,

Fig. 5.2-1. March and December loss trains

Fig. 5.2-2. June, storage, and year average loss trains

Fig. 5.2-3. Heat and mass balance, CS-RS driving EPGS (design power)

Fig. 5.2-4. Heat and mass balance, CS-RS driving EPGS and charging ESS (normal solar power operation) at equinox noon

Fig. 5.2-5. Heat and mass balance, CS-RS driving EPGs with ESS discharging (low solar power operation)

and the steam generator inlet and outlet temperatures are held constant at design conditions to generate the values given in the figure. The widening of the differential temperature across the collector field is observed; it has climbed to 291.5 K (524.6°F), i.e., 8.4 K (15.1°F) more than for case (b). With the low mass flow, the field piping pumping power is down so that the net energy output climbs to 101.2 MW(e) net.

Figure 5.2-6 gives the heat and mass balance for case (d), where the CS-RS drives the EPGS with the ESS discharging under conditions of intermittent cloudiness. This balance is calculated at equinox noon and assumes that there have been prior incident clouds such that the hot salt temperature delivered to the salt-steam generator is 11 K (20°F) less than normal. It is assumed that the salt flow to the steam generator is unchanged so that the steam conditions from the superheater and reheater decline. The EPGS gross output therefore declines, and a net output of only 99.6 MW(e) results. In practice, some increasing of salt flow to minimize the loss of throttle steam conditions may be a more desirable response in this mode of operation.

Figure 5.2-7 gives the heat and mass balance diagrams for case (e), where the CS-RS charges the ESS only. This case is evaluated at equinox noon and has operating parameters identical to the same portion of the system given in Figure 5.2-4.

Figure 5.2-8 gives the heat and mass balance diagram for case (f), where the ESS discharges to drive the EPGS. This case is identical to the same portion of the system given in Figures 5.2-3 and 5.2-4, except that since the output is not burdened with the salt pumping power, it rises to give 101.8 MW(e), a maximum among all these reported modes.

Figure 5.2-9 provides a heat and mass balance for case (g), overnight shut-down and hold of all plant subsystems. This case is based on an average overnight heat loss in the steam generator and turbine of 3.17 MW(t). In fact, this heat is not lost. It is mostly a degradation of availability as the hot salt and warm salt are blended during the startup and shutdown process.

Fig. 5.2-6. Heat and mass balance, CS-RS charging ESS with ESS discharging (intermittent cloudiness prior to equinox noon)

Fig. 5.2-7. Heat and mass balance, CS-RS charging ESS only at equinox noon

Fig. 5.2-8. Heat and mass balance, ESS discharging driving EPGS only

Fig. 5.2-9. Heat and mass balance, overnight shutdown condition

5.3 COST ESTIMATES

5.3.1 Basis of Cost Estimates

5.3.1.1 General Requirements

The cost estimate is based on the conceptual design and engineering information prepared for the study in the form of engineering drawings, outline specifications, and equipment lists. Estimating methods consistent with the conceptual nature of the design information were employed and rely on informal vendor contact as well as extrapolation from Bechtel and General Atomic Company historical information. The client was assumed to be an electric utility company.

The construction cost estimate is composed of field costs, engineering services, contingency and fee. The largest category, field costs, comprises the direct cost of permanent plant equipment, labor, and the indirect cost of temporary construction materials, supervision, etc., that are to be distributed across the entire facility. The estimate anticipates an engineer-

constructor direct-hire operation employing field construction labor forces.

5.3.1.2 Price Levels

The estimate is at second quarter 1979 price and wage levels. No allowance has been made for future escalation.

5.3.1.3 Field Construction Costs

Equipment Budgetary supplier's estimates based on conceptual designs and specification were obtained verbally and in writing for the major equipment items:

- Turbine Generator - General Electric
- Main Condenser - Ecolaire Corp.
- Feedwater Heaters - Southwestern Engineering Co
- Steam Generator & Accessories - Struthers-Wells Corp.
- Fired Heater - KTI
- Pumps w/variable speed drivers - Linde Hydraulics Corporation and Von Roll Ltd.

Other equipment was estimated using Bechtel data derived from recent purchase orders and vendor contacts.

Material Quantity estimates were made from drawings for the identifiable materials of concrete, steel, pipe, wire, cable, etc., and these quantities were priced using current unit prices

from Bechtel and General Atomic Company sources. Other materials that would be delineated on the detail design drawings (not developed in this conceptual design stage) have been evaluated by experience as a percentage of the installed cost of the identified materials.

The general level of pricing includes the following:

Civil Structural:

- Delivered Concrete \$ 40.00 per cubic yard
- Reinforcing steel 400.00 per ton
- Fabricated embedded metal .80 per pound
- Fabricated Structural Steel 1000.00 per ton

Electrical:

- Cable tray - 24" \$ 11.00 per linear ft
- 5kV mcm Cable 3C250 8.70 per linear ft
- Cable, 600V, 3C #10 .78 per linear ft
- Rigid Conduit - 3" 4.75 per linear ft

Piping, including freight:

- Carbon steel, sch 10s \$.38 per pound
- Stainless steel, sch 10s 2.30 per pound
- Stainless steel, sch 20s 2.19 per pound

Valves:

Vendor quotes were obtained for all major valves in the Energy Storage Subsystem and Heat Transport Subsystem. The vendors were:

- Ball valves - Kamyr Valves Inc.
- Check valves - Clow Corporation.

Salt

The 1979 cost of draw salt, based upon telephone quotes, is 33.08¢/kg (15¢/lb). This is derived from bulk costs of 35.83¢/kg (16.25¢/lb) for KNO_3 and 16.98¢/kg (7.7¢/lb) for NaNO_3 ; or 27.34¢/kg (12.4¢/lb) for the draw salt mixture. The transportation cost of 5.51¢/kg (2.5¢/lb) brings the total cost at the plant site to approximately 33.08¢/kg (15¢/lb).

Subcontracts - Typical subcontracts for installation of equipment and materials were estimated and priced in accordance with Bechtel and General Atomic Company experience from current sources. Major subcontracts for this project are assumed to be let for the fabrication of the collectors, all steel tanks, pipe and tank insulation and the augering of the collector foundations.

5.3.1.4 Construction Labor

The direct construction cost for the installation of the plant equipment and material was estimated using recent productivity experience, a wage rate based on labor contracts and fringe benefits for Barstow, California.

The wage rate of \$13 per hour used in the direct field cost estimate is based on a craft mix appropriate to this type of construction. Fringe benefits such as health and welfare, pension and vacation, F.I.C.A. taxes, workmans compensation, allowances for casual overtime, show up time and subsistence are included in indirect field costs. Sufficient manual labor to complete the project within the construction schedule was assumed to be available in the project vicinity, or to be attracted to the area by the travel allowance.

Indirect Field Costs The indirect field costs are those items of construction costs that cannot be directly attributed to the permanent plant facilities and thus are accounted separately. They were estimated on the basis of Bechtel experience on previous jobs in the western United States and adjusted to reflect the specific characteristics of this project. This resulted in an indirect cost equal to 163 percent of direct labor costs.

The items covered by indirect field costs are as follows:

- Temporary Construction Facilities: Temporary buildings, working areas, parking areas, utility system and general purpose scaffolding. (24% of direct labor)
- Construction Services Supplies, and Expense: General job cleanup, maintenance of construction equipment and tools, material handling and surveying, construction equipment, small tools, consumable supplies and purchased utilities (28% of direct labor)
- Field Staff, Subsistence and Expense: Field craft supervision, engineering, procurement, scheduling, personal warehousing, first aid and the cost of operating the field office (23% of direct labor)
- Craft Benefits, Payroll Burdens and Insurances: Associated with the hiring of personnel are certain payroll additives. These include health and welfare, pension, vacation, FICA, workmans' compensation taxes, subsistence to cover an allowance and insurance. (64.5% of direct labor)
- Equipment Rental: An allowance to cover the rental of construction equipment such as cranes, earth moving equipment, trucks, etc. (24% of direct labor)

5.3.1.5 Engineering Services

Engineering services include engineering and other home office supports. Engineering includes preliminary engineering and optimization studies, material and subcontract specifications, detailed engineering, vendor drawing review, site investigation, and support to vendors. Other home offices costs are procurement, cost engineering, planning and scheduling quality assurance, pre-operational testing, and project management.

The allowance for the cost of engineering services is based on Bechtel historical information for steam power plants and other

projects of a repetitious nature, including aluminum plants. While engineering services historically run from 4 percent to 10 percent depending on the complexity of the project, a figure of 5 percent was selected for the Line Focus Solar Central Power Plant. This value is judged to be typical for a plant that, while new in concept, does not depart radically from established construction practices and includes a significant amount of repetitious activities.

5.3.1.6 Contingency

Included in the estimate is a contingency allowance for costs that, while undefined at the time of the estimate, are always incurred during the course of a project and are within project scope. Historical information shows contingency to range from as low as 10 percent, for well developed construction technologies such as power plants and refineries, to 25 percent or more, for new and relatively undeveloped construction technologies. Based on historical data a 10 percent contingency allowance was included for the EPGS, Administrative Areas and Land and Site Preparation. A 15 percent contingency was assigned to all other subsystems. Contingency is a real construction cost and cannot be considered as a source of funds for overruns or additions to the project scope.

5.3.1.7 Fee

A typical prime contractors fee for a project of this magnitude and nature has been included at 3.75 percent of total project cost, including contingency.

5.3.1.8 Qualifications

The following are the major qualifications in the estimate:

5100 Land, General Site Preparation. The cost of the land was based on recent information issued by Sandia Laboratories. The cost was based on a price of \$890 per acre, with a total of 573 acres being required.

The cost of land clearing, rough grading, fencing, roads, firewater piping, sewage treatment, yard lighting, grounding and instrumentation are based on recent Bechtel experience. The costs have been appropriately factored to meet specific project requirements and land use.

5200 Administration Areas. The cost of the administration and maintenance buildings, along with miscellaneous structures such as guard shacks, lunch rooms and changing rooms, have been estimated on the basis of historical Bechtel experience. The estimate includes a HVAC system and plant equipment, but excludes office equipment and furniture.

5300 Concentrator Subsystem. The estimate for this subsystem is based on utilizing a subcontractor for the augering of the collector foundations, with the main contractor building the collector foundation and installing the collector. The collectors are assumed to be fabricated by a subcontractor at a facility adjacent to the construction site.

The estimated cost for the foundations is based on previous Bechtel construction experience as well as verbal quotations from various contractors and material suppliers in the Southwest United States. Due to the highly repetitious nature of the work involved, it was determined that a 25 percent labor savings could be expected over standard installation rates.

In the case of the collector estimate, the material costs were developed through written vendor quotations. Shop labor rates, including allowances for material handling and quality control were developed, in line with General Atomic Company recent experience. Shop assembly labor was utilized whenever possible in place of field assembly labor.

5400 Receiver Subsystem. The material costs for the receiver were developed from vendor quotations. Shop labor rates, including an allowance for material handling and quality control, were developed in line with General Atomic Company recent experience. A 90 percent learning curve was utilized reflecting

the repetitive nature of the work involved. Shop assembly labor was utilized whenever possible in place of field assembly labor.

The receiver subsystem electrical estimate is based on the single line drawings. It includes the transformers, motor control centers, switches, cable tray, and the cable required to supply power to the receiver motors. All of these items have been based on take off quantities and recent vendor quotations.

The receiver instrumentation estimate assumes a group of eight 900 ft receiver units being linked to remote micro processor units. Each microprocessor is linked to the control room through a series of cables. The microprocessor unit cost is contained within the receiver subsystem instrumentation requirements. The development cost for the microprocessor system is not included within this estimate.

5500 Master Control Subsystem. The cost of the control room structure, excluding furniture and equipment, has been factored from recent Bechtel experience. The cost includes a HVAC system.

The data acquisition and control consoles are connected to the microprocessors and are assumed to be located in the control room. The cost for the hardware and installation is included in the control building cost. An allowance was included in the control building cost for additional console instrumentation required for manual controls and instruments.

5600 Heat Transport Subsystem. The capital cost estimate for the Heat Transport Subsystem includes all the costs associated with the Heat Transport piping, heat tracing, insulation and associated equipment. Generally the costs were based on the equipment list and material quantity takeoffs. The material unit rates have been derived from vendor contact or recent Bechtel experience in the case of the piping, insulation, electrical and equipment requirements. Labor productivity rates have been used in line with recent Bechtel experience, reflecting the repetitive nature and the level of complexity of the work. An allowance has been included reflecting the instrumentation requirements.

5700 Thermal Storage Subsystem. The major capital cost within this subsystem is that of the molten salt storage tanks. These costs have been estimated on the basis of a subcontract installation. Vendor quotes were obtained regarding the current price of the required steel.

A detailed valve list was derived and vendor quotes were obtained to establish their cost contribution. An allowance was used in the case of the instrumentation and electrical portion of the estimate. All other costs were established from material takeoffs and recent Bechtel experience.

5800 Electric Power Generation Subsystem. The major capital cost item in this subsystem is that of the turbine generator. The cost of this item was derived from conversations with General Electric. The price obtained was also consistent with recent

experience given the size of the turbine generator required. Other equipment costs, such as that for the condenser, feedwater heaters, and diesel generator, were obtained from vendor quotations. The remainder of the equipment was estimated in line with recent experience.

The turbine building and associated equipment were factored from a recent Bechtel project. They included allowances for the earthwork, concrete, structural steel, piping, electrical, instrumentation, painting, and insulation cost categories. The basis for the factoring was building area and volume.

Account 5800 contains the main transformer, the main auxiliary transformer, the reserve auxiliary transformer, connecting lines, and associated switch gear. It does not contain the utility substation.

Other Qualifications. Within the equipment estimate, a factor of seven percent was added to all equipment costs to reflect the cost of freight, vendor representatives, and installation bulk materials. This factor is consistent with Bechtel experience.

5.3.1.9 Exclusions

The following items were excluded from the project scope and are not therefore included in the estimate:

- Any special construction such as widening and strengthening existing roads

- Railroads
- Switchyard and power transmission lines beyond the plant high voltage terminals
- Client engineering, financing, and other client costs
- Process royalties and licenses
- Allowance for funds during construction including interest during construction
- Training plant operators
- Plant operation
- Escalation of costs beyond second quarter 1979 price level
- Final site landscaping.

It is important to realize that though not included within this estimate, the contribution of an allowance for funds during construction will be a major cost factor. Given the assumed 42 month construction period, these costs can add another 35 percent to the constructed cost in constant dollars. This assumes a cost of money of 10 percent and an inflation rate of 8 percent.

5.3.2 Plant Capital Costs

Capital costs for the line focus power plant are presented according to the DOE-specified format on the following pages. Table 5.3-1 presents the cost estimate summary by account, and Table 5.3-2 presents the summary by category. Table 5.3-3 presents costs by account number, and Table 5.3-4 presents the functional distribution of costs. Tables 5.3-5 and 5.3-6 give a more detailed cost breakdown of the concentrator and receiver subsystems in order to highlight the principal contributors to these subsystem accounts.

TABLE 5.3-1

PLANT COST ESTIMATE SUMMARY BY ACCOUNT

ACCT. NO./DESCRIPTION	<u>MH</u>	<u>Labor</u>	<u>S/C</u>	<u>Mat</u>	<u>Total</u>
5100 LAND, SITE PREP.,	23,931	312	3,323	745	4,380
5200 ADMINISTRATION AREAS	12,295	161	371	298	830
5300 CONCENTRATOR SUBSYSTEM	259,312	3,371	69,427	9,907	82,705
5400 RECEIVER SUBSYSTEM	340,418	4,426	-	50,589	55,015
5500 MASTER CONTROL SUBSYSTEM	5,120	67	-	339	406
5600 HEAT TRANSPORT SUBSYSTEM	506,079	6,579	2,238	17,611	26,428
5700 THERMAL STORAGE SUBSYSTEM	21,596	281	3,683	687	4,651
5800 EPGS SUBSYSTEM	181,900	2,365	1,990	20,280	24,635
Total Direct Field Costs	1,350,651	17,562	81,032	100,456	199,050
Indirect Field Costs					28,712
Total Field Costs					227,762
Engineering & Home Office Costs					11,390
Total Field & Home Office Costs					239,152
Contingency					34,061
Fee @ 3.75%					10,245
Total Construction Cost					283,458
Salt Cost					3,564
Total Cost					287,022
Cost Per kWe					2,870

TABLE 5.3-2

PLANT COST ESTIMATE SUMMARY BY CATEGORY

CONSTRUCTION COST ESTIMATE

CLIENT DOE/GADESCRIPTION TOTAL CAPITAL
COST SUMMARY

LOCATION _____

CONT. NO. _____

PROJECT _____

MADE BY _____

APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
		60,551	787	4877	615	6,279
A	Excavation & Civil	387,450	5038	-	5287	10,325
B	Concrete	82,970	1082	-	8749	9,831
C	Structural Steel	17,765	229	521	442	1,192
D	Buildings	294,958	3833	72,191	62,524	138,548
E	Machinery & Equipment	145,873	1897	152	7,674	9,723
F	Piping	277,241	3604	-	8,170	11,774
G	Electrical	80,843	1052	-	6,935	7,987
H	Instruments	3,000	40	-	60	100
J	Painting	-	-	3,291	-	3,291
K	Insulation					
	DIRECT FIELD COSTS	1350651	17562	81,032	100,456	199,050
L	Temporary Construction Facilities					4,215
M	Construction Services, Supplies & Expense					4,917
N	Field Staff Subsistence & Expense					4,039
P	Craft Benefits, Payroll Burdens & Insurances					11,326
Q	Equipment Rental					4,215
	INDIRECT FIELD COSTS					28,712
	TOTAL FIELD COSTS					227,762
						11,390
R	Engineering					
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					11,390
	TOTAL FIELD & OFFICE COSTS					239,152
V	Contingency					34,061
W	Fee					10,245
Z	Salt					3,564
	TOTAL CONSTRUCTION COST					287,022

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

TABLE 5.3-3

**PLANT COSTS BY ACCOUNT NUMBER
CONSTRUCTION COST ESTIMATE**

CLIENT DOE/GA DESCRIPTION Account 5100
Land and General Site
Preparation
 LOCATION _____
 PROJECT _____

CONT. NO. _____
 MADE BY LDS
 APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			TOTALS
			LABOR	SUBCONTRACTS	MATERIALS	
		16,741	218	3,171	510	3,899
A	Excavation & Civil					
B	Concrete					
C	Structural Steel					
D	Buildings					
E	Machinery & Equipment					
F	Piping S/C MH = 3200	4,000	52	152	177	381
G	Electrical	2,200	29		50	79
H	Instruments	990	13		8	21
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	23,931	312	3,323	745	4,380
L	Temporary Construction Facilities					75
M	Construction Services, Supplies & Expense					87
N	Field Staff, Subsistence & Expense					72
P	Craft Benefits, Payroll Burdens & Insurances					201
Q	Equipment Rental					75
	INDIRECT FIELD COSTS					510
	TOTAL FIELD COSTS					4890
R	Engineering					245
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					245
	TOTAL FIELD & OFFICE COSTS					5135
V	Contingency					513
W	Fee					212
	TOTAL CONSTRUCTION COST					5,060

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

PLANT COSTS BY ACCOUNT NUMBER (Continued)
CONSTRUCTION COST ESTIMATE

APPROVED _____

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

TABLE 5.3-3

PLANT COSTS BY ACCOUNT NUMBER (Continued)

CONSTRUCTION COST ESTIMATE

CLIENT DOE/GADESCRIPTION Account 5300
Collector Subsystem

LOCATION _____

CONT. NO. _____

MADE BY _____

PROJECT _____

APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			TOTALS
			LABOR	SUBCONTRACTS	MATERIALS	
A	Excavation & Civil (S/CMHR's=50,082)			1706		1706
B	Concrete	222,352	2891		3305	6196
C	Structural Steel	36,960	480		6602	7082
D	Buildings					
E	Machinery & Equipment (collector)			67,721		67,721
F	Paving					
G	Electrical					
H	Instruments					
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	259,312	3371	69,427	9907	82,705
L	Temporary Construction Facilities					809
M	Construction Services, Supplies & Expense					944
N	Field Staff, Subsistence & Expense					775
P	Craft Benefits, Payroll Burdens & Insurances					2174
O	Equipment Rental					809
	INDIRECT FIELD COSTS					5511
	TOTAL FIELD COSTS					88,216
R	Engineering					4411
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					4411
	TOTAL FIELD & OFFICE COSTS					92,627
V	Contingency					13,894
W	Fee					3995
	TOTAL CONSTRUCTION COST					110,516

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

PLANT COSTS BY ACCOUNT NUMBER (Continued)
CONSTRUCTION COST ESTIMATE

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil					
B	Concrete					
C	Structural Steel					
D	Buildings					
E	Machinery & Equipment	217,808	2832		42340	45172
F	Piping					
G	Electrical	57,760	751		2713	3464
H	Instruments	64,850	843		5536	6379
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	340,418	4426		50589	55015
L	Temporary Construction Facilities					1062
M	Construction Services, Supplies & Expense					1239
N	Field Staff, Subsistence & Expense					1018
P	Craft Benefits, Payroll Burdens & Insurances					2855
Q	Equipment Rental					1062
	INDIRECT FIELD COSTS					7236
	TOTAL FIELD COSTS					62,251
R	Engineering					3113
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					3113
	TOTAL FIELD & OFFICE COSTS					65,364
V	Contingency					9805
W	Fee					2819
	TOTAL CONSTRUCTION COST					77,988

5.3-19

TABLE 5.3-3

PLANT COSTS BY ACCOUNT NUMBER (Continued)

CONSTRUCTION COST ESTIMATE

CLIENT DOE/GA DESCRIPTION Account 5500
Master Control Subsystem
 LOCATION _____
 PROJECT _____ CONT. NO. _____
 MADE BY _____
 APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil					
B	Concrete					
C	Structural Steel					
D	Buildings	3670	48		114	162
E	Machinery & Equipment					
F	Piping					
G	Electrical					
H	Instruments	1450	19		225	244
J	Painting					
K	Insulation					
	DIRECT FIELD COSTS	5120	67		339	406
L	Temporary Construction Facilities					16
M	Construction Services, Supplies & Expense					19
N	Field Staff, Subsistence & Expense					15
P	Craft Benefits, Payroll Burdens & Insurances					43
Q	Equipment Rental					16
	INDIRECT FIELD COSTS					109
	TOTAL FIELD COSTS					515
R	Engineering					
	Design & Engineering					26
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					26
	TOTAL FIELD & OFFICE COSTS					541
V	Contingency					81
W	Fee					23
	TOTAL CONSTRUCTION COST					645

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

TABLE 5.3-3

PLANT COSTS BY ACCOUNT NUMBER (Continued)
CONSTRUCTION COST ESTIMATE

CLIENT DOE/GADESCRIPTION Account 5600Heat Transport Subsystem

LOCATION _____

CONT. NO. _____

MADE BY _____

PROJECT _____

APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
		30,210	393-	-	-	393
A	Excavation & Civil	133,228	1732	-	1674	3406
B	Concrete	34,010	442-	-	1627	2069
C	Structural Steel	-	-	-	-	-
D	Buildings	15,800	205	169	4604	4978
E	Machinery & Equipment	104,637	1360	-	6353	7713
F	Piping	184,519	2399	-	3037	5436
G	Electrical	3,675	48	-	316	364
H	Instruments	-	-	-	-	-
J	Painting	-	-	2069	-	2069
K	Insulation	-	-	-	-	-
	DIRECT FIELD COSTS	506,079	6579	2238	17,611	26,428
L	Temporary Construction Facilities					1579
M	Construction Services, Supplies & Expense					1842
N	Field Staff Subsistence & Expense					1513
P	Craft Benefits, Payroll Burdens & Insurances					4243
Q	Equipment Rental					1579
	INDIRECT FIELD COSTS					10756
	TOTAL FIELD COSTS					37,184
R	Engineering					1859
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					1859
	TOTAL FIELD & OFFICE COSTS					39043
V	Contingency					5856
W	Fee					1684
	TOTAL CONSTRUCTION COST					46,583

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

PLANT COSTS BY ACCOUNT NUMBER (Continued)

CLIENT DOE/GA

DESCRIPTION Account 5700
Energy Storage Subsystem

LOCATION _____

CONT. NO. _____

MADE BY _____

PROJECT _____

APPROVED _____

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

TABLE 5.3-3

PLANT COSTS BY ACCOUNT NUMBER (Continued)

CONSTRUCTION COST ESTIMATE

CLIENT DOE/GADESCRIPTION Account 5800
Electric Power Generation
Subsystem

LOCATION _____

CONT. NO. _____

PROJECT _____

MADE BY _____

APPROVED _____

A/C NO.	ITEM & DESCRIPTION	MAN HOURS	ESTIMATED COST			
			LABOR	SUBCONTRACTS	MATERIALS	TOTALS
A	Excavation & Civil	8500	110		40	150
B	Concrete	26500	345		270	615
C	Structural Steel	12000	160		520	680
D	Buildings	1800	20	150	30	200
E	Machinery & Equipment	58600	760	1100	15200	17060
F	Piping	35300	460		1060	1520
G	Electrical	27000	350		2300	2650
H	Instruments	9200	120		800	920
J	Painting	3000	40		60	100
K	Insulation			740		740
	DIRECT FIELD COSTS	181900	2365	1990	20280	24635
L	Temporary Construction Facilities					568
M	Construction Services Supplies & Expense					662
N	Field Staff Subsistence & Expense					544
P	Craft Benefits Payroll Burdens & Insurances					1525
Q	Equipment Rental					568
	INDIRECT FIELD COSTS					3867
	TOTAL FIELD COSTS					28502
R	Engineering					1425
	Design & Engineering					
	Home Office Costs					
	R & D					
	TOTAL OFFICE COSTS					1425
	TOTAL FIELD & OFFICE COSTS					29927
V	Contingency					2992
W	Fee					1234
	TOTAL CONSTRUCTION COST					34,153

DATE _____ REVISION NO. _____ REVISION DATE _____ PAGE NO. _____

TABLE 5.3-4

FUNCTIONAL DISTRIBUTION OF CAPITAL COSTS
1979 Dollars

5100	<u>Land and Site Preparation</u>	<u>\$1000's</u>
	Land Purchase	510
	Perimeter Ditch	10
	Clear & Grub/Rough Grading	208
	Fencing	235
	Roads	2,904
	Parking Lot	32
	Firewater Piping and Hose Houses	294
	Makeup Water Wells, Pumps & Piping	79
	Sewage Treatment Plant	53
	Yard Lighting	79
	Instrumentation	21
	Direct Costs	4,380
	Indirects, Engineering Contingency & Fee	1,480
	Total	5,860
5200	<u>Administrative Areas</u>	<u>\$1000's</u>
	Administration Building	416
	Maintenance Facility	299
	Guard Shack, Lunch Rooms, Change House	115
	Direct Costs	830
	Indirects, Engineering, Contingency & Fee	481
	Total	1,311
5300	<u>Collector Subsystem</u>	<u>\$1000's</u>
	Excavation	1,706
	Foundations	6,196
	Collectors (Installed)	67,721
	Collector Supports	7,721
	Direct Charges	82,705
	Indirects, Engineering, Contingency & Fee	
	Total	110,516

TABLE 5.3-4

FUNCTIONAL DISTRIBUTION OF CAPITAL COSTS (Continued)

5400	<u>Heat Receiver Subsystem</u>	<u>\$1000's</u>
	Receivers and Drives	45,172
	Electrical Supply	3,464
	Instruments	<u>6,379</u>
	Direct Costs	55,015
	Indirects, Engineering, Contingency & Fee	<u>22,973</u>
	Total	77,988
5500	<u>Master Control Subsystem</u>	<u>\$1000's</u>
	Control Room* and Plant Equipment	73
	MCS Equipment	244
	HVAC Equipment	<u>89</u>
	Direct Costs	406
	Indirects, Engineering, Contingency & Fee	<u>239</u>
	Total	645

* Control Room structure is part of turbine building.

5600	<u>Heat Transport Subsystem</u>	<u>\$1000's</u>
	Piping	2,913
	Valves	4,800
	Trenches & Pipe Supports	5,868
	Drain System	101
	Vent System	129
	Salt Pumps	1,333
	Insulation	2,069
	Heat Tracing	2,415
	Electrical Supply	3,021
	Instruments	364
	Steam Generator	3,035
	Miscellaneous Mechanical	<u>380</u>
	Direct Costs	26,428
	Indirects, Engineering, Contingency & Fee	<u>20,155</u>
	Total	46,521

TABLE 5.3-4

FUNCTIONAL DISTRIBUTION OF CAPITAL COSTS (Continued)

<u>5700 Energy Storage Subsystem</u>		<u>\$1000's</u>
Tank Foundations & Berms		131
Warm Salt Tanks		566
Hot Salt Tanks		2,530
Purification Loops		130
Salt Charging Systems		436
Cover Gas Systems		6
Piping		109
Insulation		482
Instruments		59
Valves		57
Heat Tracing & Electrical Supply		<u>145</u>
Direct Costs		4,651
Indirects, Engineering, Contingency & Fee		<u>1,751</u>
Total		6,402
 <u>5800 Electric Power Generation Subsystem</u>		 <u>\$1000's</u>
Turbine Building		1,745
Turbine Generator		11,237
Condenser		595
Feedwater Heaters		447
Feedwater & Condensate Pumps		345
Heat Rejection System		1,772
Feedwater/Main Steam Piping		75
Electrical		2,650
Instruments		920
Auxiliary Subsystems		<u>4,849</u>
Direct Costs		24,635
Indirects, Engineering, Contingency & Fee		<u>9,518</u>
Total		34,153
<u>Salt</u>		3,564
<u>Plant Capital Cost</u>		287,022

TABLE 5.3-5
CONCENTRATOR SUBSYSTEM COST DETAILS

	<u>\$ Thousands</u>	<u>\$ Thousands</u>
<u>Concentrator</u>		84,840
Casting Beds	5,634	
Forms	105	
Precast Panels	33,655	
Mirrors	28,327	
Engineering, Contingency and Fee	17,119	
 <u>Foundation Excavation</u>		 2,137
Excavation	1,706	
Engineering, Contingency and Fee	431	
 <u>Foundations and Supports</u>		 23,539
Concrete	6,196	
Steel	7,082	
Engineering, Contingency and Fee	10,261	
 <u>Total</u>		 110,516

TABLE 5.3-6
RECEIVER SUBSYSTEM COST DETAILS

	<u>\$ x 10³</u>
Heat Receiver Assembly	10,278
Truss Assembly	4,870
Jacks/Actuating Mechanism	17,325
Drive Subsystem	2,511
Control and Stow Motors	1,226
Electrical Supply	3,464
Receiver Instrumentation/Control	6,379
Receiver Subsystem Installation	2,832
Miscellaneous	<u>6,130</u>
Direct Charges	55,015
Indirects, Engineering Contingency and Fee	<u>22,973</u>
Total Cost	77,988

5.3.3 Operating and Maintenance Cost

5.3.3.1 General

The materials, supplies, and labor for plant operation and maintenance were estimated to reflect second quarter 1979 price and wage levels. They were included in the estimate on the basis of the following discussion.

5.3.3.2 Solar Power Plant Portion

An annual cost for the operation and maintenance of the solar portion of the plant was included on a subsystem basis. The cost was broken down as follows:

	<u>Material \$</u>	<u>MHRs</u>
Collector Subsystem	290,000	16,400
Receiver Subsystem	460,000	2,500
Mirror Cleaning	<u>120,000</u>	<u>18,500</u>
Total	870,000	37,400

Table 5.3-7 provides the basis for concentrator and receiver subsystem scheduled and unscheduled maintenance labor. Materials and parts for maintenance purposes are detailed in Table 5.3-8. The frequency and duration were modeled after the maintenance protocol of Table 3.5-1. It should be noted that the basis is a judgment estimate and does not have an experience base. Therefore, the data of Table 5.3-7 should be used with caution. A burdened labor rate of \$12.50/hr is believed to reflect the relatively unskilled nature of the solar collector field maintenance. This then gives a total yearly operation and maintenance cost for the collector field of \$1,340,000. Some interesting observations from the data of Table 5.3-7 are as follows:

- The receiver subsystem requires about as much maintenance as the concentrator subsystem if cleaning is included.

TABLE 5.3-7

MAINTENANCE LABOR COSTS (Steady State--excludes shakedown costs)

Item No.	Description	Frequency (No./yr)	Duration (hr)	Number	Time (hr)
	<u>Modules</u>				
1.	Visually inspect mirror modules for cracking, settling	1	0.04	24,600	984
2.	Check mirror row alignment	1	0.25	984	246
3.	Realign mirror row	0.2	4	984	787
4.(a)	Replace mirror (assume 1/2 mirror replaced over plant lifetime of 30 yr at twice labor of installation = $7343 \times 2 / 30$ hr)	--	--	--	<u>490</u>
	Subtotal items 1-4				2,507
	<u>HRA Support Structure</u>				
5.	Inspect	0.5	0.02	25,584	256
6.	Paint	0.12	0.1	25,584	<u>307</u>
	Subtotal items 5-6				<u>563</u>
	<u>HRA Actuating/Steering Mechanism</u>				
7.	Inspect jacks	4	0.01	25,584	1,023
8.	Check alignment of row	1	0.25	984	246
9.	Clean/lubricate jacks	2	0.05	25,584	2,558
10.(a)	Replace failed jacks (20% failed over plant lifetime)	0.00667	0.5	25,584	85
11.	Inspect cog belts	4	0.005	25,584	512
12.	Replace cog belts (5-yr life of belt)	0.2	0.3	25,584	1,535
13.	Align actuating/steering	0.25	0.2	25,584	1,279
14.	Inspect bearings	4	0.005	25,584	512
15.	Lubricate bearings	1	0.01	25,584	<u>256</u>
	Subtotal items 7-15				<u>8,006</u>

5.3-30

TABLE 5.3-7 (Continued)

Item No.	Description	Frequency (No./yr)	Duration (hr)	Number	Time (hr)
	<u>HRA Drive Train</u>				
16.	Inspect drive shaft assembly	2	0.05	984	492
17.	Lubricate bearings	1	0.01	25,584	256
18.	Inspect motors	2	0.05	984	98
19.(a)	Replace motors	0.1	1	984	<u>98</u>
	Subtotal items 16-19				<u>944</u>
	<u>Control Valves</u>				
20.	Inspect	2	0.01	984	20
21.(a)	Replace actuator (5-yr life)	0.2	0.2	984	39
22.(a)	Replace valves (once over plant lifetime)	0.033	2	984	<u>66</u>
	Subtotal items 20-22				<u>125</u>
	<u>HRA Assembly</u>				
23.	Inspect (glass cracks, leaks, coating failures, fasteners)	2	0.05	24,600	2,460
24.	Paint HRA	0.12	0.5	24,600	1,476
25(a)	Replace cover glass (50% over plant lifetime)	0.0167	0.1	24,600	41
26(a)	Repair leaks (1 per 10 rows/yr)	0.1	4	984	394
27.	Check and calibrate HRA instrumentation	2	0.5	984	984
28.(a)	Replace instrumentation (replace all twice over plant lifetime)	0.067	4	984	262
29.	Inspect downcomers	2	0.05	984	98
30.(a)	Replace downcomers	0.067	1	1,968	131
31.(a)	Replace secondary concentrator reflector	0.1	0.5	25,584	<u>1,279</u>
	Subtotal items 23-31				<u>7,125</u>
	(a) Corrective Maintenance = 4,420 hr Scheduled Maintenance = 14,850 hr Total = 19,270 hr				
	<u>Cleaning</u>				
32.	Clean concentrator mirrors [3.048 m (10. ft)/min - man]	12	1,472	--	17,712
33.	Clean receiver cover glass (5% of item 32)	12	74	--	886

TABLE 5.3-8

YEARLY EXPENDITURES FOR MAINTENANCE MATERIALS AND SPARE PARTS (\$)

Collector Subsystem

Small tools and alignment instrumentation	2,500
Mirrors (50% over 30 yr) = $\frac{17,259,000}{30 \times 2}$	287,650
Adhesive	<u>Included above</u>
	290,150

Receiver Subsystem

Grease/dry lubricant	2,500
Paint [1 gal/man-hour at \$15/gal] = (307+1535) x 15	27,630
Small tools/handling equipment	5,000
Jacks, 170 x \$287	48,790
Cog belts, 5117 (7.04 + 5.80)	65,702
Drive Motors (984 x \$1248.35)	122,838
Valve actuators, 197 x \$157	30,929
Valves, 33 x \$550	18,150
Cover glass, $\frac{24600 \times .5}{30} \times \12.96	5,314
Welding/fitting supplies	2,000
Secondary concentrator replacement, $\frac{24600}{10} \times \9.36	23,026
Downcomers/uprisers, 984 x 2 x 0.0667 x \$208	27,303
Instrumentation, 2,340,500/30	<u>78,017</u>
	458,099

Mirror Cleaning Materials

Water	105,000
Detergent	12,000
Brushes/other cleaning supplies	<u>2,000</u>
	119,000

- Scheduled maintenance far exceeds unscheduled maintenance (this favors system reliability since collector rows can be readily taken in and out of service).
- Receiver subsystem corrective maintenance significantly exceeds concentrator subsystem corrective maintenance.

5.3.3.3 Balance of Plant

The basis for this information was to survey the operating and maintenance costs of similar size power plants throughout the United States. The source of this information was recently published Federal Power Commission information. A range from \$1.2 million to 2.8 million per annum (excluding fuel) was found with the most consistent number being \$1.9 million (1979 dollars). \$1.9 million was the cost used in this study, representing both material and labor expenses.

5.3.3.4 Total Annual Operating Costs

The first year operation and maintenance cost was calculated in mills per kWh. Within the calculation, an annual capacity factor of 45.6 percent was utilized. The annual operation and maintenance costs for the 106 MWe (gross) plant were estimated at \$3,240,000 dollars per annum. A breakdown of costs is as follows:

	<u>Annual Cost (dollars)</u>	<u>mills/kWh</u>
Solar Portion	1,340,000	3.4
Balance of Plant	<u>1,900,000</u>	<u>4.7</u>
Total	\$3,240,000	8.1

5.3.3.5 Operating Personnel

Computation of the cost of operating personnel was based on an annual salary of \$25,000. Inherent within this average salary is the assumption of a mix of plant personnel, including supervision and management.

5.3.4 Construction Schedule

The engineering and construction schedule is based on the conceptual design engineering and conceptual estimate prepared for the study. The schedule is based on experience gained in the design and construction of fossil-fired plants adjusted for facility differences.

5.3.4.1 Construction Sequence

The construction sequence and methods generally follow an approach to the construction of commercial plants which permits procurement and construction to commence before engineering is complete, thus shortening the schedule. The schedule is predicated on the economic use of manpower as included in the conceptual estimate, rather than the earliest possible completion date.

5.3.4.2 Schedule

Figure 5.3-1 shows the engineering and construction schedule for the plant and indicates that the project could be completed within a 42-month period. The governing factor is the order, manufacture, delivery and installation of major equipment.

The following are the major items included in the schedule and the assumptions that form the basis for the time durations and sequence.

Figure 5.3-1
ENGINEERING, PROCUREMENT AND CONSTRUCTION SCHEDULE
SOLAR LINE FOCUS POWER PLANT

5.3.4.3 Engineering Services

Engineering. Contract award is the point at which siting studies are complete and optimization studies leading to design engineering begin. Design starts with the development of sufficient detail to initiate the procurement of the major equipment. It is approximately 60 percent complete when construction begins, and continues for 12 months thereafter. Engineering support is provided throughout the construction period.

Environmental Report. An environmental report must be made to the appropriate government agency which must approve the project before the notice to proceed with construction is given to the client. It is the underlying assumption of the schedule that the report preparation and approval will not be a critical time constraint.

Long Leadtime Items. Delivery leadtimes for the following items were based on current information from potential vendors:

<u>Item</u>	<u>Leadtime</u>
Turbogenerator	120 weeks
Main condenser	58 weeks
Steam generator (erected)	145 weeks
Salt valves	52 weeks
High pressure feed heater	52 weeks

It is assumed that the purchase orders for these items can be issued in a manner consistent with the jobsite requirements.

5.3.4.4 Field Construction

Civil/Structure. The site work can begin as soon as the notice to proceed is received. The excavation and concrete work will not be a governing factor in scheduling.

Mechanical. Installation of mechanical equipment can begin near completion of civil/structural work. Major equipment will begin to arrive and be installed in the later part of the construction period. The critical elements of the schedule center around the purchasing, delivery and erection of the steam generator and the turbine generator. At the time of project inception, the leadtimes of major items should be ascertained and the project schedule adjusted accordingly.

Piping, Insulation, Valves and Instrumentation. Prior to erection in the field, it is assumed that the pipe will be prefabricated in the field. Once the pipe trenches are completed, the pipe and valve installation will begin. Insulation of the pipe is assumed to begin as soon as piping systems have been tested and heat tracing installed. The instrumentation will be installed during the later part of the pipe installation work, and will be the last construction work completed.

Electrical. Underground cabling will begin as soon as the cable arrives on site. The heat tracing will begin as piping systems have been tested. Electrical work will continue through final plant checkout.

Acceptance Testing. This occurs at a time when most of the equipment and bulk materials have been placed and continues until all testing is complete at turbine roll.

5.3.4.5 Ready for Plant Startup

The plant is ready for startup when all functional testing has been completed by the construction crew and the plant is accepted by the owner for his own operator startup.

5.3.5 Cost of Electricity

The first plant cost of \$287.0 million has been converted to BBCOE using the methodology and data base supplied by DOE and listed in Appendix I.* The LFSCPS design described here with a 420-MW(e) storage capacity has a capacity factor for the given insolation of 45.6%. These values translate to a BBCOE of 151 mills/kW-hr.

The BBCOE declines as the capacity factor is increased as shown in Figure 5.3-2. It should be noted that all the data of Figure 5.3-2 were obtained from previous computer runs (Ref. 5.3-1) with the SS2 code using the preliminary cost data then available. The data have simply been renormalized to the current capital cost. The capacity factor may be increased by enlarging the size of the collector field and thermal storage system. For the 100-MW(e) plant, the BBCOE

*Note that 1979 was used as the reference year and that plant commissioning occurred in 1990.

Fig. 5.3-2. BBCOE as a function of capacity factor and plant rating - first plant values

declines to 143 mills/kW-hr at an 83% capacity factor. The reduction of BBCOE occurs because the fixed-cost portions of the plant, such as the EPGS and buildings, have a higher utilization, i.e., are amortized over a larger yearly electric output. Plant capacity factors much in excess of 83% are difficult to achieve irrespective of weather conditions. This is because the collected solar energy on the best days of the year, the equinoxes, fills up the thermal storage completely and it is not emptied by full-power operation in the succeeding 24 hr. Thus, further capacity increase can come only from the use of long-term thermal storage, or by discarding some collected energy, or by using the backup fossil heater on days when the storage is not completely filled. All three of these strategies can be expected to increase the BBCOE for capacity factors above 83%.

The BBCOE declines as the plant rating is increased up to a point, as shown in Figure 5.3-2. The reasons for the BBCOE reduction are that the EPGS efficiency increases modestly with plant rating, that the costs of some portions of the plant (such as software, some buildings, the salt charging system) do not change, and that other portions of the plant, such as EPGS components, scale less than linearly with plant rating. The collector field costs scale almost directly with plant rating. The heat transport system cost scales non-linearly with plant rating since the piping not only increases as the collector field increases, but experiences disproportionately greater thermal losses and salt pumping power. As a consequence, the BBCOE declines with increasing plant rating but then shows a lesser advantage with increasing plant capacity factor. In the case of a 500-MW(e) plant, a minimum occurs at about a 55% capacity factor.

The BBCOE considered thus far has been based on the first plant cost which has been realistically calculated for the given design. Reductions in the first plant cost will occur as the plant is replicated. However, the computation of the cost of such a replicated or Nth plant is essentially an exercise in forecasting, which is not properly supported by a design base. While such forecasting has been performed and reported in the literature, it is GA's belief that such costs are exercises in optimism and largely understate the true costs. Certainly, a believable rationale for the actual costs of the 10-MW(e) Barstow solar electric plant and the predicted costs of the advanced central receiver systems remain to be developed. The learning factors given for the various accounts on the central receiver system in the RFP that resulted in this study have not been substantiated and GA and Bechtel National cannot support them. Consequently, this report does not contain BBCOE data for an Nth plant.

REFERENCE

- 5.3-1 "Line Focus Solar Central Power Systems, Phase I. Midterm Topical Report for the Period September 30, 1978 through March 31, 1979," DOE Report GA-A15408 (Draft), General Atomic Company, May 1979.

6. LIMITED SUBSYSTEM AND COMPONENT EXPERIMENTS

6.1 INTRODUCTION

The purpose of this task was to perform subsystem and component experiments that address key aspects of the selected line focus technology and could be accomplished within the limited schedule and budget of this conceptual study. After considering the various aspects of technical uncertainty in the selected design approach, it was decided to evaluate the weatherability of glass mirrors attached to concrete, since this is the construction used in the primary concentrator of the FMSC and a 30-yr life in the field is sought.

A test plan for these tests was prepared, and the test samples were fabricated to a tightly controlled specification. The test samples were then sent to Sandia Laboratories in Albuquerque, New Mexico, where they were placed in an environmental chamber for accelerated freeze-thaw testing. At the time of this writing, the samples have not yet been placed in the chamber owing to the lack of test space. Space is expected to become available in the near future.

6.2 TEST PLAN

6.2.1 Objective

The objective of these tests is to evaluate various mirror back surface protective coatings in sunlight and in an accelerated environmental test chamber. The coatings to be evaluated are used primarily to protect the silver or copper enamel on back surface mirrors and to seal the edges of these mirrors. Exposure in the environmental test chamber produces accelerated weathering in humidity and temperature but not in sunlight. Ninety cycles in the chamber are about equal to 1 yr of natural exposure. A 30-yr life test would require about 2700 cycles or 900 days of exposure at 3 cycles per day. During these tests edge creep of oxidation and spot size growing from holes in the coating can be measured. From these data rates of deterioration will be determined. Some samples will be exposed to sunlight but can be measured in real time only.

6.2.2 Background

Sandia Albuquerque has conducted a number of exposure tests on concrete and glass mirror samples for GA. These tests were conducted in their environmental test chamber under the FMSC collector subsystem program for Sandia. Tests were conducted on several different bonding and protective agents. Initial results were not encouraging for long-term survival, and a second phase of testing followed, principally using a sprayed-on polyurethane enamel for a protective coating and 3M's Type 468 transferable contact adhesive for bonding the mirrors to the concrete facet. The procedure for treating the mirror samples was as follows:

1. Clean the glass.
2. Tin process.
3. Silver and bake.

4. Apply copper enamel and bake.
5. Seam the edges.
6. Spray on the polyurethane enamel and let dry.
7. Apply Type 468 adhesive.
8. Stick the mirror to the concrete block.

Samples were tested that had been prepared using steps 1 through 6, 1 through 7, and 1 through 8, with and without step 5. The samples tested without step 5 peeled from the edges and corroded into the silver from the edge in a relatively short time, i.e., 2 to 3 months in the chamber, which is equivalent to 2 to 3 yr of natural exposure. The samples prepared with step 5 are surviving after some 16 months in the chamber, i.e., equivalent to about 16 yr of natural exposure. However, mirrors installed outdoors on the collector subsystem show that the polyurethane is deteriorating under sunlight exposure after 1-1/2 yr.

6.2.3 Justification

It is becoming generally known that protecting the silvered surface of mirrors exposed to the elements is a universal problem. The mirror optical properties are dependent on the quality of the silver, and collector performance suffers severely if the silvering deteriorates. Also, good bonding agents are required to keep the mirror solidly bonded to its substrate to increase its resistance to hail damage. The advances noted above were an improvement but do not satisfy the requirement for the 30-yr lifetime projected for solar plants. New materials and techniques are becoming available which may solve these problems. General Atomic's background experience provides direction for a test program, and GA has a need for an effective mirror protective coating and bonding agent. The problem is sufficiently severe and the need great enough to justify a test program for mirror coatings.

6.2.4 Approach

Mirror samples were prepared using various coatings and bonding agents, and these were shipped to Sandia Laboratories' Environmental Test Chamber for exposure. This chamber cycles the exposure temperature from 244 K (-20°F) to 328 K (+130°F) and the relative humidity from 20% to 80% three times daily as indicated in Figure 6.2-1. At the 244 K (-20°F) point, the samples are well below freezing and any entrapped moisture will be solidly frozen. Samples will also be exposed to direct sunlight at Albuquerque, New Mexico, to take advantage of the higher elevation. Sandia Laboratories has indicated that it will cooperate on such a test program.

6.2.5 Test Program

The tests will be conducted on samples that are nominally full size, and the mirrors are mounted in the same way as they would be on an actual collector. Therefore, the samples are 50.8 to 76.2 mm (2 to 3 in.) wide by 127 cm (50 in.) long. Each mirror is attached to a single block of concrete which is slightly larger to permit observation of the bonded edges and be convenient to handle. The glass used for the mirrors is either Corning 0317 fusion glass, low-iron float glass, or clear float glass. It is planned to test two silvering processes, at least two protective coatings, two adhesives, and a few samples of Schott mirrors. The test sample matrix is given in Table 6.2-1.

The mirror samples were made up as a group, identified, and shipped to Sandia Laboratories' Environmental Test Chamber. Some samples will be exposed to sunlight to measure the serviceability of the coatings in sunlight. Sandia personnel will then observe the test and open the chamber periodically to check on the results of the test. Sandia will call GA when measurements other than number 1 in Table 6.2-2 should be made. General Atomic personnel will then review the test results and make necessary

Fig. 6.2-1. Environmental test chamber freeze/thaw cycle conditions

TABLE 6.2-1
FIRST TEST MATRIX

Sample Number ^(a)	Roman Numeral ^(b)	Mirror ^(c)	Back Coating	Edge	Adhesive ^(d)	Remarks
1	I	C-SD	Polyurethane	Seamed	3M-468	Only baked-on coating
2	II	C-SD	DC-Q1-2577 ^(e)	Seamed	468-468	
3	III	C-SD	DC-808	Seamed	468	
4	IV	Schott	None	Cut	468	
5	V	C-SD	Q1-2577	Seamed	426-468	
6	VI	C-SD	808	Seamed	428	
13	XIII	C-SD	Q1-2577	Seamed	411-468	Only baked-on coating
14	XIV	C-SD	808	Seamed	411	
15	XV	Schott	None	Cut	426	
16	XVI	Schott	None	Cut	411	Only baked-on coating
17	XVII	C-SD	None	Seamed	468	Copper enamel only
18	XVIII	C-SD	None	Seamed	411	Copper enamel only
10	X	Coupons		Cut		

(a) Inked on end of concrete block.

(b) Chiseled into side of block; chisel mark then inked.

(c) C = Corning 0317 glass; S.D. = San Diego Mirror silvering.

(d) 468 = 0.127-mm (0.005-in.) transferable adhesive,
426 = 0.762-mm (0.030-in.) neoprene-filled double tape,
411 = 0.381-mm (0.015-in.) composition-filled double tape.

(e) DC = Dow Corning silicones.

TABLE 6.2-2
TEST OBSERVATIONS AND MEASUREMENTS

No.	Test
1	Visual-observe deterioration in from edges and oxidation spots and record location and size.
2	Measure depth of edge creep.
3	Measure spot size of spots observed in No. 1.
4	Measure spectral reflectance of mirror coupons.
5 ^(a)	Measure reflectivity of mirror samples taken from mirror strip.
6 ^(a)	Measure reflectivity of deterioration spots on samples taken from mirror strip.
7	Inspect bond adhesion and measure depth of loose points developing from edges. Finally, when mirror is removed, measure size of loose spots.

(a) These measurements will have to be made on samples removed from the test module, which may result in breakage. The main mirror strip cannot be measured until after the test is over since it has to be removed from the concrete.

measurements. Special small coupon samples are included where appropriate to facilitate measurements, such as transmissivity and reflectivity, which Sandia will be requested to perform. The observation schedule is given in Table 6.2-3.

6.2.6 Materials to be Tested

The following materials and techniques are to be evaluated in these tests:

1. Silvering - chemical-deposited silver with copper base enamel. The silvering is baked on after application and the enamel is also baked on.
2. Edge treatment. The edges will be seamed, i.e., the sharp break ground to allow the sealant to cover the silver glass interface. Some samples will be included without seamed edges for comparison.
3. Protective coating - Dow Corning 808 two-part silicone coating.
4. Protective coating - Dow Corning Q1-2577 single-part silicone coating.
5. Other protective coatings - if it appears appropriate to test them at the time.
6. Adhesive - 3M Type 468 acrylic transfer adhesive.
7. Adhesive - 3M Type 426 and Type 411 polyester tape.
8. Other adhesives - if it appears appropriate to test them at the time.

Appendix IV gives the fabrication specification for the concrete beams to be used in the tests.

TABLE 6.2-3
TEST OBSERVATION AND MEASUREMENT SCHEDULE

Observation Schedule Elapsed Time (weeks)	Evaluation on Measurement from Table 6.2-2
0	GA, No. 4 on all coupons
1	Sandia, No. 1
2	Sandia, No. 1
3	Sandia, No. 1
4	GA, No. 1, 2, 3, 7 Remove coupons for No. 4.
8	Sandia, No. 1
12	GA, No. 1, 2, 3, 7 Remove coupons for No. 4, No. 5, and No. 6. Remove obviously deteriorating samples.
16	Sandia, No. 1
20	Sandia, No. 1
24	GA, No. 1, 2, 3, 7 Remove coupons for No. 4 and remove obviously deteriorating samples for No. 5 and 6.
32	Sandia, No. 1
40	Sandia, No. 1
48	GA, No. 1, 2, 3, 4, 5, 6, 7

7. COMMERCIAL PLANT ASSESSMENT

Section 7.1, Potential Improvements, was prepared by GA, while Section 7.2, Plant Limitations, and Section 7.3, Utility Perspective of Plant, were prepared by SPS.

Solar generating plants inherently will be very site specific owing to factors such as land requirements, available insolation, alternate fuel availability, and costs, as well as the nature of the load cycle of the particular utility company. The financing structure may also vary somewhat from one utility company to another.

The assessment that follows attempts to establish the limitations, potential operating problems, potential improvements, and economic value of the LFSCPS to the electric utility. The analysis by SPS assumed the plant would be located within the SPS electrical grid and would operate under meteorological conditions of the area. The economic evaluation was conducted with the same financial parameters and methods that SPS would use to evaluate any other source of electrical power. The actual 1978 fuel cost and electrical demand were used in the calculation of fuel savings.

The above factors tend to reflect SPS's own site-specific perspectives into the assessment. However, they should be characteristic of many investor-owned electric utilities in the Southwestern United States.

7.1 POTENTIAL IMPROVEMENTS

The design which is presented in Section 4 and whose performance and cost are given in Section 5 represents the results of an 11-month design study. The results pertain only to a first plant. The study has uncovered areas where the cost appears high and where re-engineering can be expected to lead to improvements. Performance improvements have also been identified. This section summarizes the areas of potential improvement. It should be noted that when projections are made to reach Nth plant costs, rationales to support these projections should be made so that the resulting costs have credibility. Table 7.1-1 illustrates the sources of potential improvement which are seen to come from three areas: improved performance, improved design, and improved manufacturing/plant assembly. These sources are detailed in Sections 7.1.1, 7.1.2, and 7.1.3, respectively.

7.1.1 Improved Performance

Table 7.1-2 shows the improved performance that results from using a set of optical properties for the solar selective coating on the receiver that is less conservative than the reference set. Table 7.1-2 also shows the improved performance from the projected reduction of the coupling and shadowing losses together with the use of a back-silvered mirror on the secondary concentrator (reflectivity of 0.94) instead of the present polished aluminum (reflectivity of 0.88). The use of slightly curved mirror strips leading to reduced light spillage also yields a performance improvement as given in Table 7.1-2. The sum of these particular improvements is a net reduction of 17.3% in the number of concentrators and receivers.

TABLE 7.1-1
SOURCES FOR COST REDUCTION AND LEARNING

Improved Performance

1. Higher mirror reflectivity
2. Higher solar absorptivity by heat duct
3. Lower IR emissivity by heat duct
4. Improved reflectivity of secondary concentrator
5. IR reflective coating on cover glass
6. Curved mirror strips
7. Higher solar transmission through receiver window
8. Lower-conductivity insulation
9. Lower thermal losses from buried field piping

Improved Design

1. Longer receiver and collector row
2. Foamed plastic panels (simpler foundations)
3. Wider panels
4. Thinner material sections
5. Simplified HTS trenchings
6. Lower-cost electric trace heating and combined electric supplies

Improved Manufacturing/Assembly

1. Zero tooling/setup costs
2. More automation with higher throughput
3. Better trained manufacturing/assembly workers
4. Factory fabrication of concentrator panels
5. Factory assembly of receiver onto FMSC panel
6. Lower-cost materials from higher throughput

TABLE 7.1-2
CALCULATED PERFORMANCE IMPROVEMENT EFFECT ON BBCOE

	BBCOE		Percent Reduction
	mills/kW-hr	%	
Reference data (including: Cornell coating, 1% coupling, ^(a) 5% shadow ^(b))	150.1	100	0
Reference with change to Engelhard coating	133.8	89.2	10.9
As above plus: no coupling, shadow = 2%, and second surface mirror on secondary concentrator	127.7	85.1	15.0
As above with slightly curved mirror strips	124.2	82.7	17.3

(a) Change from early design eliminated receiver section coupling but loss allowance was retained.

(b) 5% shadow calculated "worse case" for original compound parabolic secondary concentrator on receiver and support retained despite major reduction in receiver projection and structure change.

In many cases, the performance improvements can be fully realized only by suitable system redesign. For instance, the curved mirror strips can be taken advantage of in any of the following ways:

1. Reducing the number of facets for a given concentrator aperture (lower capital cost),
2. Reducing the light spillage relative to the same number of flat mirror strips,
3. Increasing the concentration factor of the collector, thus leading to reduced thermal losses.

Improvements in the receiver can be realized by reducing the infrared (IR) emission with a suitable IR-reflective coating on the innermost surface. At the same time, the use of a solar anti-reflective coating on the outermost surface can reduce the reflection of the incoming concentrated insolation. Such coatings are being developed but require testing at the operating temperatures expected for the receiver cover glass. It seems reasonable that following successful development, a performance improvement of 5% can be projected from the appropriate combination of these two coatings.

The thermal loss by conduction through the receiver is relatively small. Nevertheless, it can be reduced by either adding more insulation or using lower-conductivity insulation, particularly at the hot end of the receiver. [The present insulation thickness of 39.5 mm (1.375 in.) represents an optimum average.] It is then reasonable to project a performance improvement of 1%.

The thermal loss in the HTS piping may be improved by burying the piping in loose insulation that permits expansion instead of wrapping it as in the current design. A factor of 4 reduction in heat loss is projected. This corresponds approximately to a 1-1/2% performance improvement.

The combination of the above multiplication factors leads to a total projected improvement of 26.1% reduction in concentrator surface area.

7.1.2 Improved Design

Concentrator Subsystem. The major design change which leads to improved system economics is to make the primary concentrator panel from a lightweight material so that it can be factory fabricated and shipped. Such a material should have adequate stiffness, strength, and resistance to environmental exposure and should be of low cost. A material having these properties appears to be fiberglass-reinforced foamed polyurethane. The factory fabrication of the panel also leads to factory emplacement of the mirror strips and assembly of the receiver superstructure and drive train components on the panel so as to minimize the more costly field installation. This approach was not utilized in the reference design because it involves some projection and because the requisite material properties are not adequately established, but it can lead to substantial reductions in the cost of the concentrator subsystem. The panel is sufficiently lightweight that it can be readily emplaced and lined up, thus reducing field labor costs.

A corollary to using a lightweight panel is reduced handling and shipping costs. In addition, it is possible to make the panels a little wider and still retain their ease of handling. Increasing the panel width can increase the geometric concentration ratio and reduce the receiver thermal losses.

The lighter concentrator panel [181 to 227 kg (400 to 500 lb)] will require a smaller foundation than the reference concentrator panel cost. Hence, some reduction in foundation costs can be expected with this approach.

The net impact of the above change to the primary concentrator is expected to yield a reduction in the direct installed primary concentrator subsystem cost of 25% for the same quantity of panels.

Receiver Subsystem. The trussing structure on the receiver is not fully optimized, and it is reasonable to predict that with modest adjustments such as changing the length, thickness, or spacing of the trussing tubing, which have a negligible impact on cost, a span as long as 14.6 m (48 ft) can be attained while still meeting the specified sag and twist requirements. A further method for achieving this span increase is to make the trussing continuous across each receiver instead of discontinuous as in the reference design. However, the trussing must be modified to ensure that it does not interfere with the actuating mechanism at each receiver end. Although the receiver actuating arm and jackscrew will be heavier duty as a consequence of this span increase, economies will be realized in the fewer number of parts and field assembly labor.

The receiver is considered relatively complex. While not yet identified, design simplifications should be possible that reduce the number of parts to assemble and/or reduce the fabrication/installation labor.

Heat Transport Subsystem. Design improvements to the HTS are projected from a variety of means including the following:

1. System reoptimization (in view of the high cost of electric trace heating and trenching, factors not appreciated fully in the original studies).
2. Simplified trenching (elimination of concrete lining and checker plate covering).
3. Simplified insulation (straight runs buried in insulation instead of being wrapped).
4. Simplified electric trace heating.
5. Simplified trenching for electric lines.

6. Possibly shared wiring/power supplies for receiver motor drives and electric trace heating since they are not normally used at the same time.

Most of the above items reflect that the reference conceptual design is a starting point for the preliminary design. Some items have turned out to be more costly than anticipated and need to be re-engineered to lower their cost. In addition to the above direct changes, further improvements can be expected if longer collector rows are utilized and if a reduced number of rows are realized from the projected performance improvements. The latter point is a reflection of the fact that the HTS piping cost grows faster than linearly with linear increases in the collector field area.

7.1.3 Improved Manufacturing/Assembly

Modest learning was incorporated into the costs of the receiver, which is a factory-fabricated unit. No learning was incorporated into the concentrator panel fabrication because it was performed at a local factory adjacent to the site. By using a shippable lightweight panel, fabrication continuity can be achieved and productivity improvements can be forecasted. Incorporation of a 90% learning factor permits substantial cost reductions to be realized for an Nth plant.

The higher throughput that accompanies a repetitious collector field component design provides the motivation for higher degrees of automation and justifies the associated capital investment. This further supports the above proposed 90% learning factor.

Other means for reducing manufacturing and assembly costs are given in Table 7.1-1. Improved materials (i.e., lower direct cost or lower indirect cost through improved ease of working) are a further source of learning, though not yet identified. Two possible sources include the use of non-standard thickness for the wall of the heat transfer duct and the sheet steel shell of the receiver. Standard thicknesses were employed in the reference design.

A further possibility is the use of lower-grade steel for the hot piping and receiver duct which has been internally coated so as to have a tenacious oxide coating that is not attacked or removed by the molten draw salt.

7.2 PLANT LIMITATIONS

The basic design configuration of the LFSCPS, which utilizes both hot storage tanks and warm storage tanks, provides excellent flexibility for startup, for continued operation during transients of solar insolation, and for utilization of the full output of the solar collector and receiver at all times.

Potential limitations of the LFSCPS may relate to the areas described below: ;

1. Land Requirements

The principal limitation facing any utility considering solar generation is suitable land availability and land cost.

The plant rating selected will be limited by the amount of land available in many cases. Land requirements for locating an LFSCPS in SPS's service area would be similar to those throughout the Sun Belt. Serious consideration would need to be given before locating a solar facility near any concentration of residential dwellings. Because large, remote land areas are available, which would also lend themselves to connecting into existing transmission facilities, remote siting would be preferred. Site selection requirements therefore would have to be examined for land use, possible restrictions because of mineral development, agricultural activities, specific problems with wildlife or endangered species, and availability due to ownership. Some study would be required of socioeconomic and demographic considerations, but these probably would not significantly limit the siting of the LFSCPS.

2. Scaling

The size of the utility grid and the sizes of generating units in service must be taken into consideration in selecting the size of a solar generating unit which could be utilized efficiently, and these may be a limiting factor.

Within SPS's service area, which is similar to most of the semiarid land areas available in the Sun Belt, the major consideration of scaling up would deal with the flexibility of maintaining, and operability of, the collector system with special emphasis placed on operating and maintaining all of the heat transfer system.

With SPS's limited experience in assessing this type of problem, it is estimated that between 200 and 250 MW(e) would be the maximum limit to scaling up. The minimum for scaling down would be in the range of 50 to 75 MW(e). These estimates are based on GA's knowledge of the economics arrived at in this study and experience in selecting generating capacity for SPS's operating system, and the need to still be competitive with all types of energy alternatives.

3. Environmental Limitations

In those areas where a large potential for solar energy exists, there are some environmental limitations. Consideration would need to be given concerning other parameters in the climatology; for example, in SPS's service area, where the wind speed ranges from an annual average of 4.47 to 6.71 m/s (10 to 15 mph) to peaks as high as 31.3 to 44.7 m/s (70 to 100 mph), extra structural strengths will be required in the design of a collector. Special consideration would have to be given to heat losses due to wind. During certain periods of the year, in the spring

especially, the area experiences dust storms which could present problems of damage to the mirror surfaces, keeping the mirrors clean, and reduction of sunlight during some of the daylight period. Another possible limiting factor is very cold, or below-freezing, temperatures which could be experienced in the winter.

Even though these limitations have been identified here, it is evident from the total study of the LFSCPS that there are numerous advantages to locating such a system in the semiarid areas of the Sun Belt.

4. Development Potential

The limitations of development potential are related to overcoming the development problems with the receiver, handling a heat exchange fluid in a large number of linear feet of piping, overcoming the limitations of draining the system at night, and some of the other problems mentioned in Section 7.3.1.

5. Financial

The economics of the particular utility situation and the funds available for such a unit will also impose stringent limitations on solar generating installations.

If the technical problems can be resolved and engineering economic evaluations are favorable, the step from conceptual study to commercialization would require that the utility industry be assured of a cost effective project which would minimize any negative financial impact. Capital requirements risk, long equipment life, and other proper financial benefits to the investor-owned and investor-regulated electric utility

would require the support of the consumer and stockholder. The financial limitations at this time of an LFSCPS might require Government support in the form of total funding of research, development, and demonstration.

Another alternative to the financial limitations could be a sharing by the whole utility industry and Government of the cost and risk associated with new technology.

Still another solution would be to provide attractive financial incentives by Government which would encourage the use of an LFSCPS so the national benefits of energy independence could be realized. The financial limitations of solar energy alternatives to the private sector of the electric utility industry (which includes the stockholder and consumer) have been examined carefully and in detail in the Southwest Project four-volume report (June 1978, prepared by a group of southwest utilities and Stone & Webster for DOE). This examination deals with typical financial limitations and recommends some potential solutions. An individual case-by-case financial evaluation would be required for future development projects because of the variables from one utility to another.

7.3 UTILITY PERSPECTIVE OF PLANT

7.3.1 Performance

From the user's viewpoint, there are numerous factors that have an effect on the value of the energy from a solar power plant and must be constantly considered in the daily operation of a utility system:

1. The daily load curve pattern aids the dispatcher in predicting, on an hourly basis, what generation will be needed to ensure the availability of required generation capability. This forecasting aid will be of significant benefit in optimizing the use of solar energy and in making the best use of any storage available.
2. Spinning reserves are required to back up any type of generating equipment. If a significant amount of solar generation were installed on a utility system, the amount of spinning reserves required would increase owing to the non-firm nature of solar generation.
3. The time required to start up units that are on stand-by as well as the probable duration of the projected daily peak will determine whether additional units will need to be placed into service in addition to the solar generator unit, and will thus influence the actual daily savings possible from the use of the solar generator.
4. Fuel availability and contractual minimum take or pay arrangements have more influence on scheduling of individual units than unit efficiencies. All of these factors influence the day-by-day worth of the energy savings to be derived from solar generation.

5. Provisions must be made for unit maintenance outages by utilizing reserve capacity or interconnections. A solar generator cannot provide this type of reserve requirement without the addition of a salt heater or a stand-by boiler to permit continuous operation.

The typical daily load curves for SPS have been compared with the typical daily output of the proposed LFSCPS to determine the best utilization of the system. Energy storage of both more than and less than the base design of 420 MW(e)-hr has been evaluated to determine the benefits of each.

The LFSCPS is compatible with the requirements of the utility network from a generation standpoint, i.e., to reduce the quantity of fossil fuel consumed.

The capacity benefit of a solar generation unit is limited because of the non-firm nature of the energy source. The addition of a backup fossil-fired salt heater would allow the plant full capacity credit. Although a direct steam boiler might be desirable from an operating standpoint (i.e., allowing the plant to operate even though the salt system were unavailable), the direct salt heater has many advantages.

Since the salt heater is smaller and has less auxiliary equipment than a steam boiler, the initial capital cost as well as the O&M costs are less. The thermal storage allows the salt heater to have a lower duty cycle (40% to 50%) than the steam boiler. The buffering effect of the storage might allow the salt heater to run at night and possibly take advantage of any take-or-pay fuel supplies.

The modification to the solar collector (decreasing the tilt angle to 25 deg) to obtain maximum output during the peak electrical demand (July or August) resulted in less fuel savings than the base system (tilt angle of 35 deg).

The escalation of fuel prices during the year, coupled with the fact that the larger and more economical coal units undergo annual overhauls during the winter, made the December fuel cost higher than the July cost. Thus, the base system with the maximized annual generation, and hence more output in the winter, is preferred.

The SPS service area experiences many meteorological conditions that will have a serious impact on the performance of the solar collector. The most serious is the reduction in annual kilowatt-hours due to cloud cover. Figures from the National Weather Service indicate that within SPS's service area, only about three-quarters of the total sunlight is available annually to the solar plant (see Table 7.3-1). The effect of cloud cover was taken into account in the economic assessment.

Another local meteorological condition that affects the solar plant operation is the occurrence of high gusting winds. The base design receiver structure and drive mechanism should be designed to allow operation in winds above 15.6 m/s (35 mph). National Weather Service data indicate the average wind velocity in SPS's service area is about 5.81 m/s (13 mph), with the "fastest mile" (fastest observed 1-min value when the direction remains within 10 deg) being 31.3 m/s (70 mph). Damage from high winds is a common occurrence within the area, especially during the spring. However, data from SPS's wind turbine-generator candidate site, near Amarillo, indicate the wind is only above 13.4 m/s (30 mph) slightly over 1% of the time. Since some of these high winds occur at night and during periods of cloud cover (frontal passages and thunderstorms), the impact of high winds on availability is not serious.

A secondary effect caused by high winds and dry weather is blowing dust. Dust storms, under the worst conditions, could block insolation from the collectors 7 to 10 days a year. However, the real impact of the blowing dust is on the increased maintenance of the collector field. Erosion of the glass surfaces by blowing dust has not been quantified, but would also tend to increase maintenance. Although these problems exist, their impact on the availability of the plant will be minimal.

TABLE 7.3-1
PERCENT AVAILABLE SUNSHINE IN SPS SERVICE AREA

Month	Monthly Percent Available Sunshine ^(a)	Seasonal Test Period	Seasonal Percent Available Sunshine	Weeks in Season	Weeks of Possible Operation per Season
February	73	Spring	75	12	9
March	78				
April	73				
May	78	Summer	80	12	9.6
June	82				
July	74	Peak	74	4	2.96
August	79	Fall	76	12	9.12
September	72				
October	76				
November	76	Winter	74	12	7.4 ^(b)
December	75				
January	70				

(a) National Weather Service Climatological Data Summary for Lubbock, Texas, 1977.

(b) Turbine generator unavailable for 2 weeks during this period for annual overhaul.

The effectiveness of "wind barriers" erected around the collector field and the shielding of the center collectors by the outside rows to keep out dust is questionable. Observations have shown that houses in the center of town suffer the same effects from wind and dust that houses on the edge experience.

Since damaging hailstorms occur every 5 to 7 yr, the collectors must be constructed to survive such impact with no cracking or breaking of glass. General Atomic's test shows that adequate quality control during mounting of the glass will prevent such damage; hence, hailstorms will not affect plant availability.

The problems associated with clearing snow and ice from the collectors are valid. However, the area experiences less than 3 days a year with snowfall amounts greater than 25.4 mm (1 in.), so the reduction in availability would be only a few days per year.

7.3.2 Economics

The economics studies have been based on operation at full load for a reduced number of hours for various periods of the year due to average cloud cover based on a 5-yr average cloudiness factor as determined from the U.S. Weather Bureau data for this area. An economic evaluation of the LFSCPS is given below.

The annual kilowatt-hour output of the LFSCPS has been calculated by GA using the SUNPOW computer code for each hour of the day for five representative days: equinoxes, solstices, and a representative peak electrical demand day.

The hourly generation of the solar plant was input to a proprietary SPS fuel budgeting program that distributes the system load among the turbine-generator units expected to be in operation during the particular data

point and calculates the cost of fuel used by each unit. The program used the actual 1978 load demand and fuel supply costs to arrive at the daily calculated fuel cost. The weekly system fuel supply costs are shown in Table 7.3-2.

A weekly system fuel cost was calculated by running the fuel budgeting program with typical weekday and weekend daily loads and summing the resulting cost. This was done for each of the five test points. An annual fuel cost was calculated from the five developed test weeks and the number of each type of test week per year.

The validity of this method was verified by comparing calculated values of system load and fuel cost for the Base Case (no solar generation) with the actual 1978 values. The generation values agreed within 4.2% and the fuel costs within 0.3%.

The fuel budgeting program was run for various configurations of the line focus solar plant (1/4, 2, 4, and 6 hr of storage, 25- and 35-deg angle of tilt, etc.), and weekly fuel costs were determined for each of the five test periods.

The effects of cloud cover were also considered in the assessment by determining the average percent available sunlight for each of the test periods. The monthly percent available sunlight for the last 5 yr was used to calculate the number of weeks that the solar plant could possibly operate during each of the test periods. The monthly averages were obtained from a 1977 National Weather Service Climatological Data Sheet for Lubbock, Texas. The results are shown in Table 7.3-1.

The differences in weekly fuel costs for the Base Case (no solar generation) and the solar test case (with solar generation) were summed for the weeks the solar plant would operate in each of the five test periods. This value represents the annual fuel savings the solar plant would provide (see Table 7.3-3) and ranged from \$3,111,000 for the 1/4-hr storage to \$5,784,000 for the 6-hr storage.

TABLE 7.3-2
SOUTHWESTERN PUBLIC SERVICE COMPANY 1978 AVERAGE
WEEKLY SYSTEM FUEL COST FROM BASE CASE RUN

Week of	System Fuel Cost [\$/10 ⁶ kJ (\$/MMBtu)]
March 21	1.451 (1.375)
June 21	1.569 (1.487)
July 21	1.515 (1.436)
September 21	1.692 (1.604)
December 21	1.620 (1.536)
Annual average	1.578 (1.496)

TABLE 7.3-3
UTILITY'S ECONOMIC ASSESSMENT OF GENERAL ATOMIC LFSCPS

Hours of Storage	Annual Output from Solar Plant (MW-hr)	Capital Invest. Total Plant (1st Plant Cost) (\$)	Annual O&M Cost (5.0 mills/kW-hr)	Cost of Electricity (mills/kW-hr)	Annual Fuel Savings (\$)	Invest. Worth to Utility Based on Fuel Savings (\$)	Capacity Credit (\$)	Total Worth to Utility (\$)	Ratio of Total Worth to Total Investment (%)
1/4	170,060	208,000,000	850,300	250	3,111,000	29,150,000	4,300,000	33,450,000	16.1
2	218,700	239,000,000	1,093,500	224	3,849,000	33,575,000	5,500,000	41,075,000	17.2
4	282,573	287,000,000	1,412,900	208	4,902,000	45,250,000	7,200,000	52,450,000	18.3
6	337,590	320,000,000	1,688,000	195	5,784,000	53,000,000	8,600,000	61,600,000	19.3

Operation and maintenance costs for each solar plant configuration were calculated based on a 5-mill/kW-hr cost. This value agreed with estimates by GA for the collector and by Bechtel National for the balance of the plant.

The cost of electricity shown in Table 7.3-3 was calculated using a 20% fixed charge rate, capital plant investment, projected O&M cost, and the calculated annual megawatt-hours produced by the solar plant. First plant costs as reported in Section 5 were utilized.

Using the annual fuel savings and O&M cost along with SPS's normal financial parameters, the SPS Discounted Cash Flow Model calculated the maximum investment worth to the utility for the solar plant. A value for capacity credit for each solar plant was determined by comparing the amount of generation produced from the 100-MW(e) solar plant with that of a conventional 100-MW(e) fossil plant. The total value to the utility provided by inclusion of the solar plant is the sum of the capacity credit value and the investment worth of the fuel savings, less O&M cost, taken over the 30-yr life of the plant.

This calculation was done for solar plants with 1/4, 2, 4, and 6 hr of storage and is shown in Table 7.3-3. The ratio of total utility worth to total investment indicates that 6 hr of storage is better than the lesser amounts, although the capital cost is higher.

The solar plant optimized to collect the most annual energy (tilt angle of 35 deg) had a 5% greater savings than the same plant optimized to collect most energy during the summer (tilt angle of 25 deg) when the SPS peak occurs. Because the program used the actual 1978 fuel cost (see Table 7.3-1), which escalated about 6% from January 1978 to December 1978, the winter fuel supply costs were higher than the peak (July). Also, the large coal units were not available during the winter due to annual overhauls. More expensive gas generation replaced the coal generation capacity. These two factors combined to give a higher fuel cost in

December [$\$1.620/10^6$ kJ ($\$1.536/\text{MMBtu}$)] than in July [$\$1.515/10^6$ kJ ($\$1.436/\text{MMBtu}$)]. Although the 25-deg tilt angle displaced more generation than the base design during the summer, the latter produced more electricity in the fall and winter when the fuel costs were higher.

The greatest savings occurred when the plant was run as follows. When the morning solar flux was high enough, the plant was brought on line at minimum load. The load was increased with the solar flux until maximum output was reached, and the excess solar energy, above that required for maximum output, was put into storage. During the afternoon decrease in solar flux, the stored energy was used to hold the output at maximum until the storage was exhausted. The 4 and 6 hr of storage allowed the output to be held at maximum well into the daily peak.

7.3.3 Technical

The design arrangement of the LFSCPS provides for good flexibility of operations, without adverse effects from solar insolation transients. The ability to operate the turbine at rated conditions beyond the end of the solar day from storage (or during brief interruptions due to a passing cloud) is a very desirable benefit of the LFSCPS.

The amount of storage can be adjusted upward if future operational considerations make this advantageous. This possibility should be considered in the layout of the collector and receiver system to provide space for possible additional tanks.

It may be desirable to utilize more than one tank for each portion of the receiver system. Even though smaller tanks will increase the cost slightly, they would allow flexibility for tank maintenance and minimize the size of any spare tank capacity requirements.

The potential for salt freezing if its temperature drops below 494 K (430°F) has been given considerable attention. This problem is of particular concern from a utility viewpoint, also. The design approach of using electric heat tracing to protect the field piping is very desirable from the standpoint of operation. It may be more efficient thermodynamically, and hence require less energy, to arrange to circulate salt at a very low rate from storage for short outages or overnight. Draining the system for extended outages will obviously be necessary. It might be advantageous to have provisions for injecting pressurized air or inert gas into the salt headers to reduce the drainage time. This would be a very minor addition to the proposed design.

All components, i.e., piping, heat exchangers, and especially the turbine-generator, must be designed to accommodate the daily thermal cycling which will be imposed on them. Standard turbine-generator designs of presently supplied steam turbines have stringent limitations on thermal cycling, particularly at higher pressures and temperatures owing to the thick castings involved. The control system should include adequate metal temperature monitoring and rate of change of temperature monitoring plus alarms to prevent abnormal thermal stresses during startup and shutdown operations. With proper precautions it is anticipated that the turbine can be started up and brought to full load from a hot start (or from the previous day's operation) in about 1 hr and therefore should not impose undue limitations on daily operation schedules.

All solar energy collected each day would be used. To aid in predicting desired load level and operating time, and to ensure maximum utilization, a small on-line digital computer may be required.

The most serious failure which can be envisioned is a leak in a hot salt storage tank. It would be anticipated that the operator, upon detecting such a leak, would transfer the contents through the steam generators and into the warm tank, or into other spare hot tanks in the field. After the hot tank was repaired, operation could resume, and the major problem would be the cleanup of the spilled salt.

A major advantage of the solar plant over conventional generation is licensability. A review of the existing regulations in Oklahoma, Texas, and New Mexico (SPS's service area) indicates there would be no problem in siting and no licensing requirements to locate a 100-MW(e) LFSCPS in the non-urban areas. In fact, indications from the Texas Public Utility Commission and the Texas Governor's Energy Advisory Council encourage the utilization of economical solar applications.

A regulatory problem faced by most electric utilities concerning the use of large amounts of capital is the preconstruction authorization by utility commissions to spend such monies (money which would be placed in the utility rate base). There would be some concern of the utility commissions for the rate payer in authorizing the use of capital money for a project that did not produce the most economical kilowatt for the rate payer. Indirect benefits such as national energy independence would be secondary in many utility commission evaluations; therefore, obtaining approval for construction of a solar facility might be a limitation.

The application of an LFSCPS would be much easier to secure permits for construction for than the now-planned coal-fired facilities. The solar system would not require the lengthy study and permit period to determine "prevention of significant deterioration," because the solar system would not be emitting contaminants as coal-fired facilities might. Therefore, there would be no requirement for ambient air background study, expensive air quality control devices, continuous monitoring of stack emissions, or continuous monitoring of air quality to assure that the source would not exceed the Environmental Protection Agency Class II ambient air requirements. In addition, there would be no troublesome records to be maintained or reporting to environmental control agencies.

For the required ambient air monitoring for coal-fired plants, the capital cost of the land, equipment, instruments, and data processing devices would range from \$500,000 to \$1,500,000 and annual O&M expenses as high as

\$250,000 could be required. Costs for air quality control emission devices such as fabric filters, scrubbers, SO₂ removal equipment, etc., could be from \$10,000,000 to \$30,000,000 depending on the quality and type of coal being used. There could be an annual cost of operating such environmental emission control equipment from \$1,000,000 to \$5,000,000 per year.

The water requirements for the solar system and fossil fuel system for use in cooling and boiler make-up are similar. Waste water discharge problems for both plants are also similar.

The impact on terrestrial resources of the solar facility would be somewhat greater than for the fossil fuel facilities because of the larger land area required for the solar collectors. Even with this greater impact in the semiarid area of the Sun Belt, it would be minimal to vegetation, wildlife, endangered species, etc. More detailed studies of specific sites may be necessary to take into consideration the unique terrestrial resources of endangered species. There would be a requirement for fencing to protect large animals from the solar system. Detailed studies may be necessary to investigate a specific site's impact on birds and flying insects, because they may be attracted to the reflective surfaces or be harmed by the concentrated solar rays.

A plan for recovery and cleanup of any release of heat transfer salt leakage from storage tanks, or receivers, would have to be formulated. To prevent contamination of groundwater because of the solubility of the heat transfer salt, it is assumed that such a leak into the atmosphere and the ambient temperatures would solidify the material so it could be easily collected (perhaps re-used). Such a plan would have to involve prompt action so rainfall runoff and percolation would not result in an adverse impact.

As the development of the LFSCPS progresses, consideration will be required to address the problems of safety to individuals who have the occasion to be near or visit the facility and to personnel involved in the handling of heat transfer fluids, exposure to concentrated solar rays, and other hazards associated with the type of system being discussed in this report.

7.3.4 Operability

Startup and shutdown of the collector-receiver system must be monitored, alarmed, and controlled through the data acquisition system to minimize inadvertent freezing of salt. The system as conceived meets this criterion. With the hot and warm storage provided for startup and shutdown, the EPGS is independent of the operation of the collector-receiver system. This flexibility permits each system to operate within its own parameters.

Drain provisions of the heat transport system (HTS) for maintenance purposes or extended periods of no insolation appear adequate. However, draining of the HTS nightly during normal operation increases the exposure to salt contamination, accelerated corrosion, inadvertent freezing in receivers, and vent valve malfunction. Malfunctions of this nature would be minimized if circulation in the HTS were maintained. Freeze protection of the receivers would be required. Perhaps additional receiver insulation and heat tracing in the stowed position would resolve this problem.

The steam generator configuration should be modified to allow control of steam pressure independent of superheat and reheat temperature. This is particularly important during startup when temperature matching in the steam turbine is critical. Normal steam conditions for a cold start of the EPGS are 3.45 MPa (500 psi) throttle [515 K (467°F) sat.] with 56 K (100°F) superheat. For a hot start of the EPGS following an overnight shutdown, throttle steam conditions would be dictated by turbine first-stage metal temperature. Typically, this throttle temperature would be 644 to 672 K (700° to 750°F). These conditions could be adequately met so long as the hot storage temperature is maintained above 728 K (850°F).

Thermal cycling of a turbine-generator especially designed for cycling service presents no particular problems. Consideration of thermal cycling can be given in the design and manufacture of the turbine and generator, such as use of thin sections and increased clearances between rotating and stationary parts.

Through the use of the thermal storage, the output of the solar plant matches daily system load level. Heat energy going into hot storage during mid-day would be utilized during the evening peak hours. A minimum energy level in hot storage would be established from operating data such as (1) energy required for a normal cold startup of the EPGS, (2) overnight system heat loss, and (3) weather forecast.

The matching of generation to energy availability (dispatch strategy) will necessarily be determined by local conditions. All factors related to the actual hourly thermal output of the solar collector and receiver system, together with the thermal storage inventory, will dictate how much generation is available. The dispatcher may elect to reduce either the output of the generator or the number of hours which the unit will operate in order to match the thermal demand to the supply and thus to utilize the resulting generation to the best advantage.

The control system for the EPGS should provide for both a "boiler follow turbine" mode and a "turbine follow boiler" mode. System dispatch criteria will change from day to day. Operation at constant electrical output (boiler follow turbine mode) might be desirable during the daily peak or under system disturbances. Normally, the solar boiler will produce steam utilizing a standard turbine follow boiler mode philosophy; i.e., turbine valves admit steam to produce desired kilowatt output, and the resulting steam demand and steam pressure act to dictate boiler output and hot salt supply requirements. Variances from actual solar output will be taken care of by addition to or removal from hot salt storage tanks.

Collector field maintenance estimates seem reasonable. The modularity of the LFSCPS allows a good preventive maintenance program; i.e., small increments of the collector-receiver field may be removed from service for maintenance without radical effects on availability.

7.3.5 Reliability

The LFSCPS has a high degree of redundancy inherent in its design. This redundancy increases the overall reliability of the system because the failure of any one line of collectors and/or receivers will have a very small effect on the total system output. Repairs can be made while the balance of the system is in operation with little or no hazard to personnel.

The reference HTS piping arrangement should have high reliability if proper expansion provisions are made. Heat tracing design must be given proper attention since poorly designed heat tracing could contribute to poor reliability of the HTS. Insulation weatherproofing is also important.

Based on years of experience in fossil fuel plants with sealed pumps operating at high temperatures, the HTS pumps will probably have the poorest reliability of any component (piping, valves, motors, etc.) in the system. Careful design and good maintenance will produce the highest attainable reliability.

The overall reliability and availability of the LFSCPS will be enhanced if circulation can be maintained in the HTS during overnight shutdowns, thereby avoiding drainage of the HTS.

7.3.6 Summary

The advantages of the LFSCPS (i.e., modularity, ease of fabrication, and ease of maintenance) warrant the continued development of methods and materials for construction of line focus collectors, special high-temperature

receiver tube coatings, and high-temperature salt handling techniques. Funding for engineering studies to develop the necessary materials and techniques to solve the technical and economic problems addressed in this assessment is justified.

7.4 PRELIMINARY MARKETING ANALYSIS

7.4.1 Introduction

This section describes the investigation of the market potential of the reference LFSCPS which is described in Section 4 and whose cost and performance are presented in Section 5.

Surveys of utilities in the U.S. Sun Belt have already been performed (Refs. 7.4-1, 7.4-2) and indicate that there is an interest in solar energy to provide repowering of existing fossil-fired units. Reference 7.4-2 indicated a potential repowering market of 5189 MW(e). It was therefore felt that instead of repeating this type of survey, it would be more appropriate to perform a generation planning exercise to determine the economic conditions under which a solar plant would penetrate the electric utility market.

A basic methodology for performing this generation planning was provided by the Aerospace Corporation. However, the methodology did not indicate clearly the strategy for plant addition. Consequently, a power unit dispatching and acquisition function was prepared by GA.

A solar power plant generally does not have a high capacity factor because the sun does not shine 24 hr a day. Although it is possible to achieve high capacity factors through the use of large thermal storage capacity and through backup fossil heaters, it seems more likely that the solar plant, being a low-operating-cost plant, will be used to displace high-operating-cost plants placed in service only during the daily power peaking period. Consequently, the typical solar plant is expected to be designed inherently for power peaking or intermediate load use and thus cannot be compared directly with a base-loaded plant.

It follows that the economic assessment of a solar plant requires determination of how the plant is integrated into a utility grid, how much high-cost fossil fuel is displaced over the year, and how this compares with alternate power sources over a suitable generation planning interval (say 3 to 5 yr). These calculations require manipulation of large quantities of data and thus have been programmed for the computer. Section 7.4.2 describes this computer code and Section 7.4.3 presents the results obtained.

7.4.2 Marketing Analysis Formalism

The structure of the marketing analysis code, called PLOY, is illustrated in Figure 7.4-1. The base data are the Electric Power Research Institute synthetic utility data (Ref. 7.4-3) for Case C with Case E as an alternative. Case C is believed to approximate the Western Systems Coordinating Council (WSCC) region, while Case E is believed to approximate the Southwest Power Pool (SPP). Thus, consideration of the two cases encompasses characteristics of most utilities in the U.S. Southwest. In both cases considered, the scaled-down power scenario was employed so as to reduce the number of plants to be tracked in the computer code. The entire utility grids are taken as point models; no account is made of the actual distributed nature of the electric generating sources and load demands.

The base data include electrical load by the hour for each day of the week for winter, spring/fall, and summer. These data are put in the form of characteristic weeks for each month of the year using the monthly peak values given in Ref. 7.4-3 for normalization purposes. Efficiency (heat rate) and availability data as a function of plant size are also given in Ref. 7.4-3 for various types of power equipment, nuclear, coal, oil, gas, gas turbine, and hydroelectric. These efficiency and availability data are curve fitted so as to be selected for any required plant size. Minimum ratings are also stated. Availabilities are utilized continuously as year-average values and serve to reduce the net plant capacity. Since

Fig. 7.4-1. Generation planning and dispatch method analysis

availabilities are queried in the code at each hour when capacity is dispatched to the demand, it would be possible to include a time-dependent availability in further development of the code.

The base data include an existing power plant mix, each plant of which is assigned a year of service introduction and a cost determined by discounting at 7% per annum compounded present purchase prices of new plants. Table 7.4-1 gives the given reference cost (to the owner) of the various new plants and their fuels.

Average fuel cost data are somewhat difficult to ascertain for utilities which have established long-term contracts with minimum provision for escalation. Thus, the data in Table 7.4-1 were utilized as a reference set together with the fuel escalation data of Table 7.4-2. The oil escalation values are fairly high and are based on oil escalation continuing at the current rate until demand subsides. Gas escalation rates behave similarly. Nuclear fuel costs remain relatively low until the demand is revitalized, which occurs later than for most fuels in view of the larger time it takes to put the plant in service after placing the order. Coal has a modest escalation rate to start with, but the rate climbs rapidly as the more-expensive-to-operate oil- and gas-fired plants are 'retired'. Eventually all fuel escalation rates settle down to a standardized inflation rate of 7%. The inflation rate is held above 7% for a few years, reflecting its institutionalization in the U.S. before it is reduced to the more acceptable norm. The power increase rate does not increase much; it actually declines during the anticipated recession of 1980 and then recovers to a rate of 4% per year. This is a lower rate than has been observed over the last 20 yr owing to the fact that electricity will cost much more and there will be more conservation. However, electricity generation by utilities is usually more efficient than locally generated power and is often more economical because it uses coal rather than oil or gas. Thus, switching to utility electric supplies will occur, leading to the power increase rates shown.

TABLE 7.4-1
1979 REFERENCE COST INFORMATION

Plant Type	Capital Cost [\$ /kW(e)]	Fuel Cost [\$ /10 ⁹ kJ (\$ /10 ⁶ Btu)]	Average Heat Rate [kJ/kW-hr (Btu/kW-hr)]
Nuclear	1200	0.664 (0.70)	10,550 (10,000)
Coal	1000	1.14 (1.20)	9,495 (9,000)
Gas turbine	180	5.31 (5.60)	14,770 (14,000)
Solar	2600	--	--

TABLE 7.4-2
ESTIMATED FUEL-ESCALATION-INFLUENCED POWER INCREASE RATES (%)

Year	Fuel Escalation Rate				Inflation Rate	Power Increase Rate
	Coal	Oil (a)	Gas	Nuclear		
1979	7	15	12	-1	10	2
1980	6	14	12	-2	11	0
1981	8	12	13	0	8	2
1982	8	10	14	3	7	4
1983	10	11	14	6	6	7
1984	12	12	14	8	7	9
1985	14	12	13	8	7	7
1986	16	12	14	8	7	5
1987	16	12	14	10	7	4
1988	14	11	14	12	7	4
1989	12	11	14	14	7	4
1990	12	11	13	14	7	4
1991	10	10	13	14	7	4
1992	10	10	13	14	7	4
1993	10	10	13	12	7	4
1994	9	9	13	11	7	4
1995	9	9	12	10	7	4
1996	8	8	11	9	7	4
1997	8	8	10	8	7	4
1998	7	7	9	7	7	4
1999	7	7	8	7	7	4
2000	7	7	7	7	7	4

(a) Gas turbine fuel same as oil.

The code then picks a set of 10 independent variables, which describe the generating capacity addition for nuclear, coal, gas turbine, and solar plants for the next 3 yr. The amount of generation capacity is also allowed to be varied in the first of these 3 yr. Then successively for 3 yr, the code eliminates plants which are obsolete in each year, dispatches the plants against the load for each month, and computes the cash flow for that year. The discounted total for the 3 yr is then determined, and the independent variables are then automatically adjusted with a packaged optimization algorithm so as to minimize this total. When the total is minimized, the selected plants for year No. 1 are added to the reference set, obsolete units are retired, and the process is continued by advancing the reference year.

The dispatching algorithm is not sophisticated since that was considered unnecessary for this type of investigation. Plants are dispatched in reverse priority of their operating cost at full rating. No plant is dispatched at less than its minimum rating. Solar plants are presumed to have the minimum operating costs among all types of plants and are always automatically dispatched over the peak demand in each day.

The solar plants are considered to be characterized by a daily output which varies for each month of the year to reflect the seasonal variation in heat collection. For instance, the FMSC plant collects most heat on the spring and fall equinoxes. The effect of inclement weather is treated simply by means of an availability factor of 80%.

The yearly operating cost of the power plants is calculated with the methodology given in Appendix I and used to calculate BBCOE for the solar power plant described earlier in this report. One significant difference is that these studies utilize a solar power plant cost to the utility instead of the cost to the manufacturer. For the case of the 100-MW(e) plant, an installed owner cost of \$260 million is used. Solar plants

of larger and smaller capacities are scaled in cost according to their capacity ratio to an 0.8 exponent. The cash flow in each year of the generation planning interval is discounted back to year No. 1 with the total cumulative inflation factor for that particular year.

When new plants are added, the following restrictions are employed. It is assumed that oil- and gas-fired plants will not be added due to Federal requirements to restrict the use of these expensive fossil fuels. It is also assumed that no hydroelectric capacity additions are possible. Thus, only coal, nuclear, gas turbine, and solar plants may be added. (The code may be modified to permit a broader spectrum of plants if desired.) Coal plants may not be smaller than 50 MW(e) or larger than 1000 MW(e). Nuclear plants may not be smaller than 600 MW(e) or larger than 1200 MW(e). Gas turbine units may not be larger than 50 MW(e) at any one installation and are added in 10-MW(e) increments. Solar plants must be at least 10 MW(e) but not greater than 500 MW(e). Multiple plants or combinations of these plants may be added in any one year.

The plant capacity added each year is allowed to vary from a minimum up to either 7% of the existing total capacity or 1.5 times the minimum required addition. The minimum addition is the larger of either of the following criteria:

1. The difference between peak demand in the year and the sum of plant ratings times their availability.
2. The difference between peak demand increased by a 7% reserve and the total capacity less outage of the two largest plants in the system.

The above two requirements relate to ensuring overall system reliability to meet peak demands. In most cases, criterion (2) dominates. Thus, it is frequently found that capacity is added to meet the system reliability requirements, not to improve the system economics per se. This is manifested in the case when zero plant additions exhibit the most favorable economics.

7.4.3 Marketing Assessment Results

The computer code PLOY described in Section 7.4.2 has been made operational. Operations with the code have yielded the following useful qualitative information:

1. New plants are added more to meet electric grid reliability criteria than for the need to meet load demand. Plant capacity (rating multiplied by average availability) is usually greater than the peak load.
2. Reliability criteria are usually established for discrete sections of a utility grid. Thus, the use of a point model instead of a distributed model in PLOY does not permit the reliability criteria to be handled well and may make the code predict larger plants to be acquired than would be the case in practice.
3. When a new plant is added, it will displace plants of higher operating cost. If the new plant has a higher operating cost than existing units, it will not be used. This acquisition strategy tends to favor plants of low operating costs or low capital cost, i.e., solar plants and turbine units, respectively.
4. In the short run, the most favored plant acquisition strategy is to acquire the bare minimum, which tends to favor smaller, less economic plants. The cumulative effect of such an approach is a higher cost of electricity. Consequently, the acquisition strategy must be evaluated over an interval (3 to 5 yr) sufficient to permit larger, more economic plants to be included and assessed.

5. It is often cheaper for a utility to meet its reliability criteria by coupling into the grid of its neighbor than by purchasing additional capacity.

Unfortunately, in its present form the PLOY code requires about 2 hr of running time to calculate a 30- to 40-yr period for the selection of plants. This has made it too costly a tool to be used for parametric analysis. Therefore, some simplified calculations have been performed based on the observation that the data of Table 7.4-2 show the power increase rate, fuel escalation rate, and inflation rate settling down to asymptotic values after about 20 yr. This suggests that the criterion of whether or not a solar plant would penetrate the utility market can be determined by comparing plants evaluated under these asymptotic conditions. It is then no longer necessary to determine how solar plants compare with older plants. Thus, the use of the PLOY code is obviated.

The equilibrium fuel costs for the plants discounted to 1979 were obtained by multiplying the base fuel costs of Table 7.4-1 by the appropriate escalation factor for the year 2000 and dividing by the corresponding price inflation factor. The capital recovery factor is taken at 15.7% based on the DOE methodology given in Appendix I. A solar plant then competes against other plants to the extent that its output displaces theirs. For the reference solar plant, with 420 MW(e)-hr of storage, the maximum achievable capacity factor is 48.6% for clear day conditions. However, with actual weather conditions, a capacity factor of 40% seems more likely. This corresponds to 3504 hr per year of operation. With these data, the results of Table 7.4-3 are obtained.

It can be seen that for the specific data assumed, coal and nuclear plants have the lowest costs. Both are roughly equal and are much less than the other alternatives. The solar plant has lower yearly capital costs than the gas turbine plant if the latter is operated at ~40% capacity. However,

TABLE 7.4-3
YEARLY EQUILIBRIUM COSTS FOR PLANTS OPERATING AT 3504 HR/YR

Plant Type	Adjusted Fuel Cost [\$/ 10^9 kJ (\$/ 10^6 Btu)]	Yearly Capital Charge (\$/kW)	Yearly Fuel Cost (\$/kW)	Total Yearly Cost (\$/kW)
Nuclear	0.768(0.81)	188	28	216
Coal	1.62(1.71)	157	54	211
Gas turbine	9.79(10.33)	28	507	535
Solar	--	408	0	408

gas turbines are typically used (as the code PLOY shows) at much smaller capacity factors (<10%). In such circumstances, the solar plant is more costly. It would appear that the cost to the utility solar plant must be about half of the assumed value of \$2600/kW if these plants are to compete in the utility market. Factors which would assist their penetration include:

1. Lower tax rates for solar plants.
2. Higher-than-predicted escalation of conventional fuels.
3. Lower cost of capital (e.g., municipality-owned plant).
4. Locally consumed output (plant not burdened by transmission line costs).
5. Repowering of the existing plant, which in turn is allowed to continue operating its existing fossil-fired equipment in a backup mode.
6. Specific fuel unavailability.

Two other observations can be made:

1. There is likely to be a market for cycling coal and nuclear plants (the latter only if nuclear energy is permitted to remain as an available energy option).
2. Smaller-size nuclear and coal plants are likely to become economically competitive with oil and natural gas plants.

REFERENCES

- 7.4-1 Curto, P. A., and Z. D. Nikodem, "Solar Thermal Repowering,"
Mitre Corporation Report MTR-7861, May 1978.

- 7.4-2 "Technical and Economic Assessment of Solar Hybrid Repowering,"
Public Service Company of New Mexico Report SAN/1608-1,
September 1978.
- 7.4-3 Zaininger, H. W., et al.. "Synthetic Electric Utility Systems
for Evaluating Advanced Technologies," Electric Power Research
Institute Report EPRI EM-285, February 1977.

8. PLANT DEVELOPMENT

8.1 DEVELOPMENT ISSUES

This section summarizes briefly the significant development issues for the plant. Significant issues are defined as those where resolution is required to ensure the technical/economical feasibility of the plant. Areas of uncertainty important to plant design which have less significance and whose resolution can reasonably be expected through further design studies are discussed in Section 8.2 as open questions.

8.1.1 Solar Selective Absorptive Coating

It is clear that a selective coating is essential for the absorbing surface of the heat receiver pipe. The coating should have a high solar absorptivity, a low infrared emissivity, and a long-term thermal stability in air. An alternative approach would involve a highly absorptive, nonselective coating on the pipe and an infrared-reflecting coating on the inside of the receiver window. The plant economics are very sensitive to coating properties as discussed in Section 3.2.4, an observation that is generally true for all systems of mid-range concentration factor (50 to 200). Design methods exist to reduce the view factor of the absorbing surface to the outside, which increases the effective absorptivity. However, for the FMSC receiver, the benefits to be gained from design modifications to lower the view factor are expected to be modest. Selective coatings are being developed and several show promise. However, none has been tested in air for extended periods of time. In addition, the chosen surface for this design study has not yet been deposited and tested on a stainless steel substrate.

There is interest in selective coatings for central receiver solar energy systems, and some coating development has been funded as a consequence of that interest. Thus, coating development for line focus receivers should be considered an incremental expense.

8.1.2 Primary Concentrator Fabrication

A rapid low-cost fabrication method for the concrete primary concentrator is described in Section 4.2. While there are some commercial analogies to this type of concrete fabrication, the entire process has not been tested. It is essential to demonstrate that the fabrication method will achieve the desired cost and performance goals. The latter depend principally on achieving the desired angular precision of the mirror strips (3σ of ± 15 min). The following elements are required:

1. Establishing that secondary concrete forms made from a steel master will have the desired precision.
2. Establishing that the concrete panels obtained from the secondary concrete forms fall within the specified angular tolerance.
3. Establishing that the glued-on mirrors meet or exceed the overall angular accuracy requirements.
4. Developing a rapid technique for measuring the angular accuracy of the facets on the concrete panels so as to achieve good quality control.
5. Demonstrating that the cost goals are indeed achievable.

General Atomic expects to initiate these efforts in the near future.

8.1.3 Collector/Receiver Subsystem Performance Verification

The collector and receiver designs presented in Section 4 are substantially advanced over those designs utilized in 260-m^2 (2800-ft^2) FMSC array at

Sandia Laboratories, Albuquerque. An improved performance is also predicted due to reduction in optical and thermal losses and to the increased concentration factor. Verification of the estimated system performance through prototype construction and test is a required development step. The demonstration would confirm the adequacy of the design code, SUNPOW, and provide the necessary confirmation of the numerical values of the energy loss train. General Atomic expects to perform limited testing on such an advanced receiver system in the near future.

8.1.4 Corrosion Data

Type 316 stainless steel has been selected for the hot portions of the salt components [$>700\text{ K}$ ($>800^\circ\text{F}$)] and carbon steel for the low-temperature [$<700\text{ K}$ ($<800^\circ\text{F}$)] portions. The basis for this selection is not well established. Long-term exposure of 316 SS and other candidate materials to molten draw salt at temperatures of 839 K (1050°F) is not well documented. The data in the literature are scarce and incomplete. Corrosion allowances to be used for design purposes are considered arbitrary. Long-term exposure data are needed which encompass crevice corrosion, galvanic effects, stress in the base material, and the effect of impurities in the draw salt. The impurities to be considered should encompass those present in the supply stock as well as those arising from exposure to moist air. A range of candidate materials should be considered, including the austenitic steels and Incoloy 800. A similar requirement exists for carbon steel and its alternatives, such as low-alloy chrome/molybdenum steels, to explore the upper temperature limit at which these materials will perform satisfactorily.

The required development effort can be expected to arise from the interest in draw salt for heat transport and thermal storage in both solar energy systems of the central receiver type and in fossil- and nuclear-heated systems.

8.2 OPEN QUESTIONS

This section presents a list of questions and issues pertinent to the LFSCPS whose resolution would be expected during the normal progression from the conceptual to the final design stage. The list is not necessarily complete but reflects the significant items identified from the investigations described earlier in this report.

1. The long-term resistance of the mirror silvering (mirrors glued to concrete FMSC panel) to environmental exposure needs to be determined. Tests are under way at Sandia Laboratories Environmental Test Chamber in Albuquerque.
2. The receiver requires testing to demonstrate that its design eliminates various mechanical difficulties experienced by the units in the 260-m^2 (2800-ft^2) FMSC array at Sandia Laboratories. Fabrication studies are also needed to verify that the receiver design is amenable to low-cost fabrication.
3. The means for the automated or semiautomated process of gluing the mirrors to the concrete panels needs to be demonstrated.
4. The basis for the O/M cost of the concentrator and receiver subsystems needs to be better established by means of actual data from a large-scale system representative of the design presented herein.
5. The operational limitations of the plant using draw salt as the heat transport fluid require improved definition. The transient data, described in Section 3.6, were generated late in the program and show a relatively long time before full recovery to normal collector field operation following a short-term (13 min) cloud cover. Means for reducing this time need to be examined. In addition, the balance of the plant in response to a range of accident and malfunction conditions needs to be explored to ensure that the disadvantages of salt [strong oxidant, melting point of 494 K (429°F)] do not impair the flexibility of the plant and its responsiveness to load or insolation level changes.

6. An item needed sequential to Item 5 is the development of design criteria that permit a thermal storage management strategy to be established.
7. Demonstration of pumps and valves for continuous operation at the hot salt temperatures is required.
8. The trenching of the heat transport system is expensive. An alternate, which involves burying the pipe in insulation in the ground and which has the virtue of reducing heat losses by a factor of ~ 4 , has been demonstrated for low-temperature [533 K (500°F)] systems but not for the 839 K (1050°F) hot salt temperatures proposed here. Such demonstration is needed to verify performance improvement and trenching cost reduction.
9. The cyclical operation of the plant imposes startup and shutdown limitations (from temperature transients) on the plant components, particularly the turbine and the steam generator. The impact of these limitations on the plant availability and its economics needs to be explored. A related need is the assessment of how important the backup fossil heater is to plant operations in order to determine the most cost effective design efficiency of the heater.
10. The construction sequence and schedule need to be worked out in detail simply because they determine the required allowance for interest during construction (AIDC) as well as the escalation (ESC) amount. The sum of these two can be substantial, e.g., 35% for a 3-1/2-yr construction schedule. At the same time, the construction period is expected to be shorter than for nuclear plants. Thus, quantification of AIDC and ESC can impact the competitiveness of the installed plant relative to other plants using conventional fuels.
11. The development cost and development plan need to be worked out so that cost benefit studies can be made to determine how this plant compares with others in meeting future thermal energy needs.

APPENDIX I

LINE FOCUS SOLAR CENTRAL POWER SYSTEMS
REQUIREMENTS DEFINITION AND REFERENCE
METHODOLOGY FOR CALCULATING THE
BUS BAR COST OF ELECTRICITY

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

LINE FOCUS SOLAR CENTRAL POWER SYSTEMS
REQUIREMENTS DEFINITION
(General Atomic. Only)

December 1, 1978

Prepared by:

The Aerospace Corporation
Energy Systems Group
El Segundo, California

TABLE OF CONTENTS

	Page
1. GENERAL	
1.1 Scope	1
1.2 Description	
1.2.1 Concentrator Subsystem	1
1.2.2 Receiver Subsystem	1
1.2.3 Heat Transport Subsystem	2
1.2.4 Energy Storage Subsystem	2
1.2.5 Electrical Power Generating Subsystem	3
1.2.6 Master Control Subsystem	3
1.2.7 Plant Support Subsystem	3
2. DOCUMENTS	
2.1 Codes and Standards	4
3. REQUIREMENTS	
3.1 System Design Requirements	5
3.1.1 Electrical Power Output	5
3.1.2 Plant Operating Modes	5
3.1.3 Energy Utilization	5
3.1.4 Storage Subsystem	5
3.1.4.1 General	5
3.1.4.2 Charging Rate	6
3.1.5 Collector/Concentrator Subsystem	
3.1.5.1 Control	6
3.1.5.2 Interface	6
3.1.5.3 Field Optimization	6
3.1.5.4 Heliostat/Concentrator	6
3.1.5.5 Special Environmental Consideration	6
3.1.6 Receiver Subsystem	7
3.1.6.1 Modes of Operation	7
3.1.6.2 Control	7
3.1.6.3 General	7
3.1.7 Heat Transport Subsystem	7
3.1.7.1 General	7
3.1.7.2 Working Fluid	8

3.1.7.3	Control	8
3.1.8	Electric Power Generation Subsystem	8
3.1.8.1	General	8
3.1.8.2	Cycles	8
3.1.8.3	Control	8
3.1.9	Master Control Subsystem	8
3.1.10	Heat Rejection	9
3.1.11	Lifetime	9
3.1.12	Plant Availability and Reliability	9
3.1.13	Maintainability	9
3.1.13.1	Maintenance Locations	9
3.1.13.2	Equipment Replacement	9
3.1.13.3	Personnel	9
3.1.14	Plant Operation	9
3.2	Cost Estimates Requirements	9
4.	ENVIRONMENTAL CONDITIONS	
4.1	Baseline Design Environment	10
4.1.1	Reference Site	10
4.1.2	Insolation	10
4.1.3	Dynamic Performance	10
4.1.4	Wind Speed	10
4.1.5	Temperature	10
4.2	Operating Environments	10
4.2.1	Insolation	10
4.2.2	Temperature	10
4.2.3	Wind	10
4.3	Survival Environmental Conditions	12
4.3.1	Temperature	12
4.3.2	Wind	13
4.3.3	Wind Rise Rate	13
4.3.4	Dust Devils	13
4.3.5	Snow	13
4.3.6	Rain	13
4.3.7	Ice	13

4.3.8	Hail	13
4.3.9	Sandstorm Environment	13
4.3.10	Seismic Design Criteria	13
4.4	Lightning Considerations	13
4.4.1	Protection	13
4.4.2	Direct Hit	13
4.4.3	Adjacent Strike	14

1. GENERAL

1.1 Scope This specification defines the design requirements, subsystem characteristics, environmental requirements, and cost and performance evaluation techniques for the Line Focus Solar Central Power System (LFS).

1.2 Description

1.2.1 Concentrator Subsystem. The concentrator subsystem (CS) is composed of an array of fixed, linear focusing concentrators and supporting foundations. Each concentrator row in the array reflects solar radiation onto a moveable heat receiver in a manner which satisfies the receiver subsystem incident heat flux requirements.

1.2.1.1 Interfaces. The concentrator interfaces mechanically and optically with the heat receiver subsystem.

1.2.2 Receiver Subsystem. The receiver subsystem (RS) provides the means for transferring the incident radiant flux energy from the CS into a suitable working fluid, and consists of linear heat receiver assemblies (i.e., having a length much greater than the cross sectional dimensions) which are moveable and positioned in the focal zone of the concentrator row. The RS also includes the supporting power and control elements which interact with the master control subsystem, the superstructure and drive hardware for supporting and positioning each receiver, control valves, risers and downcomers, heat tracing (where applicable), and control systems necessary to regulate the fluid flow, temperature, and pressure and for safe and efficient operation, startup, shutdown and standby.

1.2.2.1 Interfaces. The heat receiver subsystem shall respond to control signals from the master control subsystem. The mechanical and fluid interface with the heat transport subsystem shall be on the heat receiver side of the shut off valves. The heat receiver subsystem, including the heat receivers, superstructures, and drive components, shall interface mechanically and optically with the concentrators.

1.2.3 Heat Transport Subsystem. The heat transport subsystem (HTS) is the means of transporting heat from the RS to the energy storage and electrical power generation subsystems for subsequent generation of electricity. It includes the receiver bypass (isolation) valves, inert gas blanketing and heat transport fluid purge provisions for emptying the heat receivers and downcomers (where applicable), fluid piping, thermal insulation, heat tracing (where applicable), piping supports, valves, fluid pumps, motors, and motor and valve controls, fluid conditioning

and makeup provisions (where applicable), steam generator and reheater (where applicable).

1.2.3.1 Interfaces. The HTS shall respond to control signals from the master control subsystem. Mechanical and fluid interfaces with the RS and energy storage systems shall be on the receiver and storage side of the shutoff valves. The mechanical and fluid interface with the electrical power generation subsystem shall be on the steam generator (and reheater where applicable) side of the steam isolation valves.

1.2.4 Energy Storage Subsystem. The energy storage subsystem (ESS) provides a means of storing the energy which is available from the receiver. Sensible heat storage is utilized. The ESS includes the storage tanks and internals, interconnecting pipes and valves, thermal insulation, heat tracing (where applicable), and the necessary provisions for controlling fluid pressure.

1.2.4.1 Interfaces. The energy storage subsystem shall accept and execute commands from the master control subsystem during normal transient and emergency operations. The mechanical and fluid interface with the heat transport subsystem shall be on the storage side of the shutoff valves.

1.2.5 Electrical Power Generating Subsystem. The electrical power generating subsystem (EPGS) provides the means for converting the thermal output of the working fluid from the RS and the ESS into electric power. The output from the EPGS will be regulated for integration into existing electrical power system networks. It includes the steam turbine and electric generating equipment, boiler feedwater equipment, heat rejection equipment, and power generating subsystem controls.

1.2.5.1 Interfaces. The EPGS mechanical interface with the HTS will be between the steam generator (and reheater where applicable) and the steam isolation valves. The EPGS shall interface with the utility grid.

1.2.6 Master Control Subsystem. The master control subsystem (MCS) will be used to sense, detect, monitor and control all system and subsystem parameters necessary to ensure safe and proper operation of the LFS.

- 1.2.6.1 Interfaces. The MCS shall interface with the individual subsystem controllers of the receiver, heat transport, energy storage, and electric power generation subsystems.
- 1.2.7 Plant Support Subsystems. Various plant support subsystems will be required, including power, water, compressed air, and other auxiliaries.
- 1.2.8 Site/Civil Structures. Site preparations, yardwork, various buildings, equipment foundations, and other civil structures will be required to house and support plant personnel and equipment.

2. DOCUMENTS

The following documents, of the issue in effect on the date of the contract award, form a part of this specification to the extent specified herein. Although the conceptual design effort required of the contractor is not expected to attain a level of detail commensurate with these documents, they are referenced herein to provide guidance in design approach, engineering judgment and cost analysis. To this extent, the system design shall be consistent with the intent of these representative codes.

2.1 Codes and Standards

OSHA Regulations

OSHA Title 29, part 1910 - Occupational Safety and Health Standards

ASME Boiler and Pressure Vessel Code

Section I - Power Boilers

Section II - Materials Specifications

Section V - Nondestructive Examination

Section VIII - Unfired Pressure Vessels

Section IX - Welding and Brazing Qualifications

ANSI B31.1-1977 Power Piping

Uniform Building Code - 1976 Edition by International Conference of Building Officials.

Institute of Electrical and Electronic Engineers (IEEE) Codes, as applicable.

National Fire Protection Association (NFPA) National Fire Codes - 1975.

Human Engineering Design Criteria

MIL-STD - 810C

MIL-STD - 1472

Design, Construction and Fabrication Standards

Standards of AISC (Am. Inst. of Steel Constr.)

Standards of ACI (Am. Concrete Institute)

Standards of TEMA (Tub. Exchanger Mfr. Assoc.)

Standard 650 of API (Am. Pet. Inst.)

- Welded Steel Tanks for Oil Storage

Environmental Legislation

National Environmental Policy Act (NEPA)

3. REQUIREMENTS

3.1 System Design Requirements. The LFS shall meet the design requirements listed herein.

3.1.1 Electrical Power Output. The LFS shall produce approximately 100 MWe net when operating solely from insolation and when subjected to the environments specified in Sections 4.1 and 4.2. This rating is the minimum net output of the plant after subtracting all plant auxiliary power requirements.

The Plant shall be capable of operation at full rated power for a period of at least four hours when operating solely from insolation every clear day of the year.

The LFS system shall provide 420 MWe hours of electrical energy to the utility grid when operating solely from a fully charged energy storage. Net power from storage shall be 70 MWe (minimum).

Power ratings and energy storage capacities other than those specified above should be considered and included if more attractive economic performance results.

3.1.2 Plant Operating Modes. The plant shall be capable of executing the following operating modes:

- 1) CS-RS driving the EPGS (TSS is not involved).
- 2) CS-RS driving the EPGS and charging the TSS (normal solar power operation).
- 3) CS-RS driving the EPGs while the TSS also discharges to share in driving the EPGS (low solar power operation).
- 4) CS-RS charging the TSS and TSS discharging to drive the EPGS (intermittent cloudiness).
- 5) CS-RS charges TSS only (thermal storage charging)
- 6) TSS discharges to drive the EPGS (extended operation)
- 7) Overnight shutdown and hold of all plant subsystems (subsystem conditioning).

3.1.3 Energy Utilization. The plant shall be designed to effectively utilize the collected incident sunlight for conversion to electrical energy. This includes steady-state operation as well as normal transients such as startup, shutdown, and broken sun operation.

3.1.4 Storage Subsystem

3.1.4.1 General. The LFS ESS shall be designed to maximize the economic and safe recovery of

recovery of useful energy from storage and to minimize energy storage losses. Specific size, shape and configuration constraints will be governed by minimizing the cost of electricity and by designing the power plant layout to facilitate efficient and safe operation and maintenance. The ESS shall be designed to provide safe and reasonable access for proper inspection, maintenance and repair of the structure, fluid lines, utilities, instrumentation and controls, etc. It shall accept and execute commands from the master control subsystem during normal transient and emergency operations.

3.1.4.2 Charging Rate. The Storage Subsystem charging rate must be sufficient to insure compatibility with all the plant operating modes and to insure effective solar utilization.

3.1.5 Collector/Concentrator Subsystem

3.1.5.1 Control. The concentrators may be equipped with control systems which employ sensors and drive mechanisms to direct the reflected rays of the sun onto the receiver as well as to provide auxiliary functions for other methods of operation and non-operational positionings.

3.1.5.2 Interface. The concentrators shall respond to control signals from the Master Control Subsystem.

3.1.5.3 Field Optimization. The field optimization will consider shading, blocking and other collector/concentrator position losses and trade them off with heliostat/concentrator costs, land costs, cabling costs, and piping losses to effect an economic minimum cost per deliverable kilowatt hour. Spacing between collector/concentrator modules must permit access by service vehicles, utility lines, and ground maintenance personnel.

3.1.5.4 Heliostat/Concentrator. Heliostat/concentrator design shall be based on performance, cost, manufacturing, operations, maintenance, and servicing constraints, pointing accuracy requirements, and structural requirements, including wind loading. Heliostat design should provide for stored or safe position for use at night, during periodic maintenance and during adverse weather conditions.

3.1.5.5 Special Environmental Consideration. The heliostats/concentrators shall withstand winds at

periphery on the collector field as specified in Section 4.2. If it is determined that wind loading on interior heliostats/concentrators is significantly altered by peripheral concentrators, this factor should be considered. Adequate protection shall be considered against heliostat degradation in the saltation layer (the boundary layer above the ground within which blowing particles, such as sand are concentrated), and for exposure to humidity, temperature transients and other environmental effects specified in Section 4.0.

3.1.6 Receiver Subsystem

- 3.1.6.1 Modes of Operation. The receiver subsystem shall provide a means of transferring the incident radiant flux energy from the collector/concentrator subsystem into a suitable working fluid to facilitate the operation of the LFS in any or all of the operating modes defined in Section 3.1.2.
- 3.1.6.2 Control. Sensing elements in the receiver subsystem shall be devised to provide the master control subsystem the information necessary to accomplish efficient and safe monitoring and control of the receiver subsystem. The receiver subsystem shall respond to control signals from the Master Control Subsystem. Means must be provided to protect the heat receivers upon over temperature, system malfunction, or upon command.
- 3.1.6.3 General. For concepts utilizing a tower mounted receiver, the tower shall be designed to provide access for maintenance and inspection of the structure, receiver, working fluid, instruments and controls, power conversion equipment that may be located on the tower, utilities, etc. For concepts having a moving receiver, appropriate positioning for necessary access shall be provided. Consideration shall be given to ease of maintenance. Adequate provisions shall be made to insure crew safety at all times for required operations, inspection, maintenance and repair. The receiver design shall be consistent with the intent of appropriate ASME Boiler codes.

3.1.7 Heat Transport Subsystem

- 3.1.7.1 General. The heat transport subsystem shall provide the most cost effective means of transporting heat from the receiver to the energy storage and electrical power generation subsystems.

3.1.7.2 Working Fluid. The working fluid for the heat transport subsystem shall be selected on the basis of performance, total system cost, fluid handling requirements, and component development requirements.

3.1.7.3 Control. The heat transport subsystem shall respond to control signals from the master control subsystem.

3.1.8 Electric Power Generation Subsystem

3.1.8.1 General. The EPGS system shall provide the means for transforming the thermal energy of the heat transport subsystem into electric power. The output from the EPGS shall be regulated for integration into existing electrical power system networks.

3.1.8.2 Cycles. The conventional steam cycle will be used, with the particular cycle features being selected on the basis of performance, total system cost, and component development requirements.

3.1.8.3 Control. The EPGS shall accept and execute control commands from the master control subsystem during normal, transient and emergency operating conditions. Specific size, shape, and configuration constraints shall be governed by minimizing cost of electricity, by designing the plant layout to facilitate efficient and safe operation and maintenance, by minimizing the effects of thermal shock on the turbine power loop, and by providing for safe response to variations in the receiver working fluid outlet conditions.

3.1.9 Master Control Subsystem. The master control subsystem (MCS) will supply all required operational and safety controls and will provide state monitoring of all subsystems. The primary requirements imposed upon the MCS are simplicity, reliability, and cost-effectiveness of both design and operation. The MCS shall be designed using standard control practices and simple, well defined interfaces between the MCS and the other subsystem controls. Primary operation shall be automatic with operator override capability.

Reliability shall be assured by use of proven designs and elimination of single point failures through redundant elements whenever it is cost-effective to do so. Failure of the MCS shall not prevent manual operation of the Plant.

The MCS shall contain independent control within each Plant subsystem with the coordination of control of the subsystem provided by the central MCS.

- 3.1.10 Heat Rejection. The LFS must be equipped with wet cooling type of heat rejection.
- 3.1.11 Lifetime. The system shall be designed for a 30-year operating life.
- 3.1.12 Plant Availability and Reliability. The system shall be designed for 90 percent plant availability based on reliability and maintainability exclusive of insolation conditions. Consideration shall be given in the design to attainment of high reliability by providing design and operating margins and utilizing sound engineering design practices.
- 3.1.13 Maintainability.
 - 3.1.13.1 Maintenance Locations. The entire plant shall be so designed that potential maintenance locations can be easily reached and components such as electronic units, motors, drives, fittings, sensors, etc., readily replaced.
 - 3.1.13.2 Equipment Replacement. Elements subject to wear and damage, such as supporting wheels, gears, etc., should be easily serviced or replaced.
 - 3.1.13.3 Personnel. Personnel required to maintain the plant needing special skills other than those currently employed by the utilities should be identified.
- 3.1.14 Plant Operation. Manual operational requirements for daily plant operation should be identified. Special skills required for an operator should also be identified.
- 3.2 Cost Estimates Requirements. The commercialization of line focus solar thermal power plants is highly dependent on the economics of the plant. To assess the economics requires estimates of plant cost and performance. Accurate comparison of solar plant design concepts requires cost and performance estimates based on similar assumptions. It is the intention of the cost data description that follows to gather the necessary information to verify comparability of the estimates and accurately assess cost/performance.

The cost breakdown structures for plant capital investment costs and operations and maintenance costs (see Tables 1 and 2) are both general structures with a high level of aggregation. Cost estimates are to be provided in 1978 dollars for all items applicable to the design. These costs are to be estimated for the first commercial line focus power plant. The operations and maintenance costs are to be specified for the first year of operation with an estimate of how the costs may change over the plant life.

The analysis of cost estimates is a difficult task. It requires an understanding of the alternate methods of cost estimation as well as the inputs and assumptions used by the particular methodology. To facilitate an equitable comparison of the cost estimates and an accurate assessment of the assumptions, it is necessary to have

explicit descriptions of the methodology used in formulating the estimate. For each item in the cost breakdown structures, a worksheet is to be provided showing the level of aggregation at which the cost estimates were made. The worksheet should include the inputs, assumptions, justification, and methodology used to arrive at the cost shown in the structure. These worksheets need only be provided for the line items in the structures which are at the lowest level of aggregation. The worksheets should take the form and provide the detail shown in examples in Tables 3, 4, and 5 for plant capital investment items, and Tables 6 and 7 for operations and maintenance items. These examples are illustrative and are not intended to demonstrate methodologies of cost estimation.

With a highly aggregated structure, for certain components in the designs, it is not obvious under which component in the cost breakdown structure the cost should be included. These questions should be answered by judgment. With the detailed list and/or description of equipment provided in the worksheets for each cost item, it will be obvious in comparing the costs what is included under each item. It is also requested that a drawing or schematic of the subsystems and plant be included showing the subsystem boundaries (for subsystem piping and wiring boundary definition) and defining nomenclature used for components and parts.

4. ENVIRONMENTAL CONDITIONS

4.1 Baseline Design Environment

- 4.1.1 Reference Site: Barstow, California. Site specific information shall be used as required.
- 4.1.2 Insolation: The annual and daily insolation variation is given in Figure 1.
- 4.1.3 Dynamic Performance: The maximum rate of change of incident flux shall be assumed as that which would result from the passage of an opaque cloud across an otherwise clear sky where the sharp leading or trailing edges of the shadow move across the concentrator field at a velocity of 20 meters per second (45 mph).
- 4.1.4 Wind Speed: 3.5 m/sec (8 MPH) at a reference height of 10 m(30 ft.).
- 4.1.5 Temperature: Wet Bulb - 23°C (74°F)
Dry Bulb - 28°C (82.6°F)

4.2 Operating Environments. The system shall be capable of operating in and surviving appropriate combinations of the environments specified below:

- 4.2.1 Insolation: The plant shall remain operational with insolation values to 1000 Watts/M^2 .
- 4.2.2 Temperature: The plant shall be able to operate in the ambient air temperature range from -30°C (-23°F) to $+50^{\circ}\text{C}$ ($+121^{\circ}\text{F}$).
- 4.2.3 Wind: The wind speed specifications at a reference height of 10 m (30 ft) shall be:

Figure 1: DIRECT SOLAR INSOLATION

<u>Speed, m/s</u>	<u>Frequency, Percent</u>
0 - 2	29
2 - 4	21
4 - 6	19
6 - 8	14
8 - 10	8
10 - 12	5
12 - 14	3
Less than	1

Operational performance specifications shall be met for the most adverse probable combination of wind and temperature conditions which are consistent with minimizing the overall cost of electricity. Maximum operational wind speed shall be selected to be consistent with minimization of the overall cost of electricity production. The wind velocity profile varies exponentially with height to the 0.15 power, where the reference height is taken as 10 m (30 ft.), and shall be determined according to the model show below for calculation of wind speed as a function of receiver elevation.

$$V_H = V_1 \left(\frac{H}{H_1} \right)^C$$

Where: V_H = mean wind velocity at height H

V_1 = reference wind velocity

H_1 = reference height (assume 10 meters)

C = 0.15

- 4.3 Survival Environmental Conditions. The plant shall be designed to survive without damage within the survival conditions specified below. The plant may take measures to protect vulnerable equipment at points beyond the specific operational limits. The plant, however, shall return to operational status after the environment return to within the operational limits.

- 4.3.1 Temperature. The Plant shall survive maximum ambient dry bulb temperatures of 60°F (140°F) and minimum temperatures of -30°C (-22°F).

- 4.3.2 Wind. The Plant shall survive winds with a maximum speed, including gusts, of 40 m/s (90 mph), without damage. An angle of attack of ± 10 degrees to the horizontal shall be assumed for the survival condition.
- 4.3.3 Wind Rise Rate. A maximum wind rise rate of 0.01 m/s^2 (1.3 mph/min) shall be used in calculating wind loads during stowage. However, the Plant should withstand, without catastrophic failure, a maximum wind of 22 m/s (50 mph) from any direction, in any orientation resulting from unusually rapid wind rise rates, such as severe thunderstorm gust fronts.
- 4.3.4 Dust Devils. Dust devils with wind speeds up to 18 m/s (40 mph) shall be survived without damage to the Plant.
- 4.3.5 Snow. Survive 5 lb/ft^2 (240 Pa) of snow at deposition rate of 0.3 m (1 ft) in 24 hours.
- 4.3.6 Rain. Average annual - 750 mm (30 in.)
Maximum 24-hr. rate - 75 mm (3 in.)
- 4.3.7 Ice. Survive freezing rain and ice deposits in a layer 50 mm (2 in.) thick.
- 4.3.8 Hail.
	<u>Any Orientation</u>	<u>Stowed</u>
Diameter	20 mm (3/4 in.)	25 mm (1 in.)
Specific Gravity	0.9	0.9
Terminal Velocity	20 m/s (65 fps)	23 m/s (75 fps)
- 4.3.9 Sandstorm Environment. The Plant shall survive after being exposed to blowing dust comparable to the conditions described by Method 510, of MIL-STD 810 BC.
- 4.3.10 Seismic Design Criteria. All structures shall be designed in accordance with the provisions of the Uniform Building Code, 1976 Edition, Section 2312, for Seismic Zone 3 as defined therein. In addition, all structures shall be designed to resist the concurrent application of horizontal and vertical base response spectra of the shape given by NRC RG 1.60 for 5 percent damping with maximum base accelerations of 0.3 g and 0.2 g, respectively.

4.4 Lightning Considerations

- 4.4.1 Protection. The Plant shall be provided with a lightning protection system which is optimized on a cost-risk basis.
- 4.4.2 Direct Hit. Total destruction of a single concentrator and its controller subjected to a direct lightning strike is acceptable.

- 4.4.3 Adjacent Strike. Damage to a concentrator adjacent to a direct lightning strike must be minimized within appropriate cost-risk limits. The central controller and local controllers of concentrators adjacent to a direct lightning strike must be protected or alternate control methods provided to minimize loss of collect or sub-system.

TABLE 1
SOLAR PLANT CAPITAL INVESTMENT COST BREAKDOWN
STRUCTURE

4000	Solar Plant Cost
4100	Site, Structures and Miscellaneous Equipment
4110	Site
4111	Land
4112	Yard Work
4120	Buildings
4121	Turbine Building
4122	Administration Buildings
4123	Warehouse/Maintenance Buildings
4124	Control Building
4125	Other
4130	Miscellaneous Equipment
4131	Transportation and Lifting Equipment
4132	Communication Equipment
4133	Other
4140	Plant Auxiliary Subsystems
4200	Turbine Plant Equipment
4210	Turbine Generators and Accessories
4220	Heat Rejection System
4230	Condensing Systems
4240	Feed-Heating or Recuperator System
4250	Working Fluid Circulation, Treatment, and Auxiliary Inventory Containment Equipment
4300	Electric Plant Equipment
4310	Switchgear
4320	Station Service Equipment
4330	Protective Equipment
4340	Power Wiring, Electrical Structures, and Wiring Containers

TABLE 1 (Continued)

4350	Master Control Equipment
4351	Hardware
4352	Hardware Design & Engineering
4353	Software Design, Development, and Test
4400	Collector/Concentrator Equipment
4410	Reflective Unit
4420	Drive Unit
4421	Tracking Drive
4422	Focusing Drive
4423	Motors
4424	Position Indicators or Encoders
4425	Power Distribution and Emergency Supply
4430	Control and Instrumentation
4431	Sensor or Calibration Equipment
4432	Field Control Electronics
4433	Control Signal Distribution Equipment
4440	Foundation and Site
4450	Concentrator Support and Protection
4451	Concentrator Support Structure
4452	Concentrator Protective Enclosure (if provided)
4453	Lightning Protection
4460	Field Assembly and Check Out
4470	Design and Engineering

TABLE 1 (Continued)

4500	Receiver Equipment
4510	Receiver Unit
4511	Absorber Unit
4512	Support Structure
4513	Receiver Circulation Equipment
4514	Instrumentation and Control
4515	Transportation, Field Erection, and Installation
4516	Drive Unit
4517	Control and Instrumentation
4520	Riser, Downcomer, and Horizontal Piping
4530	Working Media Cost - Steam No Cost
4540	Towers
4541	Bridges
4550	Foundation
4560	Steam Generator or Working Media Heat Transfer Equipment
4570	Design and Engineering
4600	Thermal Storage Equipment
4610	Media Containment Equipment
4620	Media Circulation Equipment
4630	Working Fluid Circulation Equipment
4640	Discharging Heat Exchanger
4650	Charging Heat Exchanger
4660	Foundation
4670	Design and Engineering
4680	Media

TABLE 1 (Continued)

4700	Heat Transport Equipment
4710	Field Piping and Valves
4720	Thermal Insulation
4730	Pumps
4740	Heat Transport Media Treatment and Makeup Equipment
4750	Steam Generator
4760	Heat Transport Media
4770	Design and Engineering
4800	Distributables and Indirect Costs
4810	Temporary Facilities, Equipment, etc.
4820	Spare Parts
4830	Architectural Engineering Services
4840	Construction Management
4850	Plant Startup and Checkout
4860	Contingency

TABLE 2: REFLECTIVE UNITS

4410 Reflective Unit

Part Description	Quantity	First Unit Cost per Unit	Subtotal	Factors					Total
				Learning Curve Factor*	G&A	Overhead	Quality Control		
Mirror .09"x100"x100" Float Glass	1000 ea.	\$35.00	\$ 35,000		1.15		1.1	\$ 44,275	
Foam 2"x100"x100"	1000 ea.	10.00	10,000	.95	1.15		1.1	12,018	
Steel Backing 100"x100" 25GA	1000 ea.	20.00	20,000		1.15		1.1	25,000	
Adhesive 3M #39	25 gallons	50.00	1,250		1.15		1.1	1,581	
Assembly and Bonding of Components	5000 hours	20.00	100,000	.90	1.25	1.75	1.1	216,563	
Tooling for Bonding Tables			10,000					10,000	
TOTAL FOR CSB 4410								\$309,737	

*Define below

Notes:

o Source of Material Cost Estimates

Mirror - quotes from PPG and Libbey, Owens, Ford

Foam - quotes from Dupont and Monsanto

Steel - quotes from U.S. Steel

Adhesive - quotes from 3M

o Source of Labor Estimate

Based on five hours per reflective unit estimated from time and motion studies.

Setup .5 hrs

Assembly of Mirror and foam with Adhesive 2.0 hrs

Assembly of Steel to Mirror and Foam 2.0 hrs

Remove and Transport to Stock .5 hrs

Notes (continued):

- o Tooling cost is for bonding equipment only; the factory building and miscellaneous handling equipment, etc., is included in the overhead rate. The capital investment in tooling is \$100,000 and has an estimated 30 year lifetime with no expected maintenance and repair costs. The capital recovery factor is 0.10.
- o The learning curve factor expresses the average cost of the units being costed as a fraction of the first unit cost (i.e., unit one). The units in this estimate are unit 2000 to unit 3000. The assumed learning curve rates are .99 on foam and .93 on labor. The first 2000 units are those built for the Barstow pilot plant.
- o Manufacturing labor rates are based on union labor rates in the XYZ City, Utah; the source of the labor rates is the "handbook of Manufacturing and Field Labor - 1977".

TABLE 3: POWER WIRING, ELECTRICAL STRUCTURES,
AND WIRING CONTAINERS WORKSHEET

4340 Power Wiring, Electrical Structures and Wiring
Containers

Estimate as a fraction of the cost estimate for switch-
gear station service equipment, and protective equipment
(CBS 4310 to) based on plants ranging in size from
100 MW_e to 200 MW_e built in the past ten years.

Total estimate for CBS 4310, 4320 and 4330	65,000
Fractional Factor for Estimation	<u>x .15</u>
TOTAL FOR CBS 4340	\$9,800

TABLE 4: TURBINE BUILDING WORKSHEET

4121 Turbine Building

Estimated based on turbine buildings built for similar plants (100-150 MW^e) during 1970 in the Southwestern United States.

Historical Building	100,000
---------------------	---------

Escalation Factor to 1978	<u>x 1.85</u>
---------------------------	---------------

TOTAL FOR 4121	\$185,000
----------------	-----------

TABLE 5
SOLAR PLANT OPERATIONS AND MAINTENANCE COSTS

OM10 Operations

OM200 Maintenance Materials

 OM210 Spare Parts

 OM211 Turbine and Electrical Plant

 OM212 Collector Equipment

 OM213 Receiver Equipment

 OM214 Thermal Storage Equipment

 OM220 Materials for Repair

 OM230 Other

OM300 Maintenance Labor

 OM300 Maintenance Labor

 OM320 Corrective Maintenance

TABLE 6: COLLECTOR EQUIPMENT WORKSHEET

OM212 Collector Equipment

<u>Past Description</u>	<u>Expected Failures per Year per Plant</u>	<u>1st Unit Cost per Units</u>	<u>Subtotal</u>	<u>Learning Curve Factor Years 1-10</u>	<u>G&A</u>	<u>Overhead</u>	<u>Total</u>
Reflective Unit (CBS-4110) (tracking)	1	\$310	\$310	.90	1.10		\$ 307
Drive (focusing) (CBS-4121)	2	450	900	.90	1.10		891
Drive (CBS-4122)	1	500	500	.90	1.10		495
Motors (CBS-4123)	1.5	400	600	.88	1.10		581
TOTAL FOR OM212							\$2,274

Notes:

- o Failure rates are based on manufacturer specifications.
- o The learning curve factor expresses the average cost of the units being costed as a fraction of the first unit cost (i.e., unit one). The units in this estimate start with unit 3000 for first three items and unit 6000 for the motors. The learning curve rate in all cases is 0.95. The learning curve factor is estimated to change as follows for spare parts purchased over the plant life:

<u>Years of Plant Life</u>	<u>Factor for all but Motors</u>	<u>Factor for Motors</u>
1-10	.90	.88
10-20	.89	.87
20-30	.88	.87

These factors are estimated based on the manufacturing facility production rate of 1000 units per year for all but motors and 2000 units per year for motors.

TABLE 7: COLLECTOR EQUIPMENT WORKSHEET

OM212 Collector Equipment

<u>Past Description</u>	<u>Expected Failures per Year per Plant</u>	<u>1st Unit Cost per Units</u>	<u>Subtotal</u>	<u>Learning Curve Factor Years 1-10</u>	<u>G&A</u>	<u>Overhead</u>	<u>Total</u>
Reflective Unit (CBS4410) (tracking)	1	\$310	\$310	.90	1.10		\$ 307
Drive (focusing) (CBS4421)	2	450	900	.90	1.10		891
Drive (CBS4422)	1	500	500	.90	1.10		495
Motors (CBS4423)	1.5	400	600	.88	1.10		581
TOTAL FOR OM212							\$2,274

Notes:

- o Failure rates are based on manufacturer specifications.
- o The learning curve factor expresses the average cost of the units being costed as a fraction of the first unit cost (i.e., unit one). The units in this estimate start with unit 3000 for first three items and unit 6000 for the motors. The learning curve rate in all cases is 0.95. The learning curve factor is estimated to change as follows for spare parts purchased over the plant life:

<u>Years of Plant Life</u>	<u>Factor for all but Motors</u>	<u>Factor for Motors</u>
1-10	.90	.88
10-20	.89	.87
20-30	.88	.87

These factors are estimated based on the manufacturing facility production rate of 1000 units per year for all but motors and 2000 units per year for motors.

TABLE 8 : SCHEDULED MAINTENANCE WORKSHEET

OM310 Scheduled Maintenance

<u>Description</u>	<u>Hours per Year</u>	<u>Unit Cost</u>	<u>Subtotal</u>	<u>G&A</u>	<u>Overhead</u>	<u>Total</u>
Mirror Washing	84	\$15	1,260	1.1	1.5	\$ 2,079
Concentrator Lubrication	500	20	10,000	1.1	1.5	16,500
Receiver Preventive Maintenance	20	20	400	1.1	1.5	660
Thermal Storage Preventive Maintenance	10	20	200	1.1	1.5	330

TOTAL FOR OM313 \$19,559

Notes:

- o Labor rates are based on rates quoted in "Handbook of Manufacturing and Field Labor Rates" for QRS City, Arizona, which is 30 miles from the plant site.
- o Hours per year estimates are based on time and motion studies.
- o G&A and overhead rates are based on rates at comparable size power plants (100-150 MW) located in Arizona.

METHODOLOGY FOR CALCULATING BUS BAR COST OF ELECTRICITY

The methodology below was taken from "Line Focus Solar Central Power Systems," RFP No. ET-78-R-03-2073, July 31, 1978, and was utilized in all calculations of the bus bar cost of electricity (BBCOE) presented in this report. The only change was to make the reference year 1979 and in the present case to use the first plant cost estimate.

1. Planning Factors

- Plant 1st unit
- System lifetime (N) 30 yr
- First year of operations (y_{co}) 1990 (Dec.)
- Construction start time (y_t) 1988 (Jan.)
- Construction time 3 yr
- Site Barstow, California
- Type of ownership Investor owned

2. Cost Input

- Plant capital investment (CI_t) \$287.0 million
- Annual operating and maintenance cost (X_o) 0.01 CI_t
- Plant capacity factor (PCF) 0.456
- Plant power level (MW) 100.52 MW(e)

3. Utility Descriptive Data

- Annual "Other Taxes" as a fraction of CI_{pv} (β_1) = 0.02.
- Annual insurance premiums as a fraction of CI_{pv} (β_2) = 0.0025.
- Income tax rate (τ_1) = 0.50.
- Ratio of debt to total capitalization (D/V) = 0.55.
- Ratio of common stock to total capitalization (C/V) = 0.27.
- Ratio of preferred stock to total capitalization (P/V) = 0.18.
- Debt interest rate (k_d) = 0.09.
- Annual rate of return on common stock (k_c) = 0.15.
- Annual rate of return on preferred stock (k_p) = 0.11.

- Investment tax credit, $(\alpha) = 0.10$.
- Depreciation $(DPF_{m,k,n}) = \text{Straight Line } (DPF_{SL_{k,n}})$.

4. General Economic Conditions

- Rate of general inflation $(g) = 0.050$.
- Escalation plus inflation rate for capital costs $(g_c) = 0.65$.
- Escalation plus inflation rate for operating costs $(g_o) = 0.065$.
- Escalation plus inflation rate for maintenance cost $(g_m) = 0.065$.
- Base year for constant dollars $(y_b) = 1979$.

5. Equations

Cost of Capital (k)

$$k = (1 - \tau) k_d \frac{D}{V} + k_c \frac{C}{V} + k_p \frac{P}{V}$$

Capital Recovery Factor $(CRF_{k,N})$

$$CRF_{k,N} = \frac{k}{1 - (1 + k)^{-N}}$$

Annualized Fixed Charge Rate $(\overline{FCR})$

$$\overline{FCR} = CRF_{k,N} \left(\frac{1 - (\tau)(n \cdot CRF_{k,N})^{-1} - \alpha}{1 - \tau} \right) + \beta_1 + \beta_2$$

Present Value of Capital Investment (CI_{pv})

$$CI_{pv} = (1 + g_c)^{y_{co} - y_b} CI_t \left(\frac{1 + g_c}{1 + k} \right)^{(y_t - y_{co} + 1)}$$

Present Value of Recurrent Costs (X_{pv})

$$X_{pv} = (1 + g_m)^{y_{co} - y_p} (X_o) \left(\frac{1 + g_o}{k - g_o} \right) \left[1 - \left(\frac{1 + g_o}{1 + k} \right)^N \right]$$

Annualized System - Resultant Cost ($\overline{AC}$)

$$\overline{AC} = (1 + g)^{-y_{co} + y_b} \left[(\overline{FCR})(CI_{pv}) + (CRF_{k,N})(X_{pv}) \right]$$

Levelized Bus Bar Energy Cost ($\overline{BBEC}$)

$$\overline{BBEC} = \frac{\overline{AC}}{(MW)(PCF)(8760)}$$

APPENDIX II

FABRICATION SPECIFICATION SOLAR PANELS-BEAMS FOR FREEZE-THAW TESTS

GENERAL ATOMIC COMPANY

GA-1485 (4/77)

TITLE: Solar Panels - Beams for Freeze-Thaw Tests																																	
Discipline S		System 01		Doc. Type SPE		Project 3295		Document No. 904078										Issue A		Date APR 20 1979													
QAL II Non-Cat. I SC-NNS														DV&S Non-Deliverable ISSUE SUMMARY										APRVL LVL 2		904069 Next Indentured Document							
Issue		Date		Prepared by		Approval										Purpose of Issue/ Sections Changed																	
																								Engineering		QA		Project					
A		APR 20 1979		V. Nicolayeff		J. Neill										B. Gregory		For Use															
<div style="border: 1px solid black; padding: 5px; display: inline-block;"> RECEIVED J. R. SCHUSTER APR 26 1979 </div>																																	
Do not write in space below. Continue Issue Summary on GA Form 1485-1.																																	
REV																																	
SH																																	
REV																																	
SH	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56					
REV																																	
SH	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28					
Page 1 of 9																																	

1 RELEASED

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

TABLE OF CONTENTS

1.0 Information and General Requirements

1.1 Scope

1.2 Work Included

2.0 Technical Requirements

2.1 Materials

2.2 Casting of Specimens

2.3 Curing and testing

2.4 Report

2.5 Acceptance Criteria

3.0 Quality Assurance Requirements

List of Tables

Table 1. Summary of Castings

Table 2. Suggested Mix Proportions

List of Figures

Figure 1 Beam and Molds

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

SOLAR PANELS - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

1. INFORMATION AND GENERAL REQUIREMENTS

1.1 SCOPE: This specification covers the work to be done by the concrete laboratory in the preparation of beams for General Atomic's (GA) freeze-thaw tests. The tests are part of a design verification program on concrete panels for solar collectors.

1.2 WORK INCLUDED:

1.2.1 The concrete laboratory shall provide all labor, equipment, materials and facilities required to cast, cure and strip twenty-four 3 by 3 by 50-in. beams in accordance with ASTM C192 and this specification, using molds provided by GA. The size and shape of specimens are given in Figure 1. The summary of castings and the schedule are given in Table 1.

1.2.2 Specifically, the laboratory shall be responsible for: a) purchasing cement, aggregate, admixtures, reinforcing bars, cylinder molds and other supplies needed for casting the specimens and testing the concrete, b) designing one concrete mix, c) unloading six steel molds delivered from GA by truck, d) assembling the molds, sealing joints, coating molds with bond breaking compound and installing the reinforcing bars in accordance with Fig. 1, e) determining specific gravity, absorption, soundness, unit weight, moisture content and gradation of aggregate; batching the concrete constituents; mixing the concrete; determining temperature, slump, unit weight and air content of fresh concrete, casting and curing the 24 beams, casting and curing eight 6 by 12-in. cylinders, testing the cylinders for compressive strength and determining the modulus of elasticity on two of the cylinders, f) stripping the beams and cleaning and reassembling the molds, g) curing the 24 beams and loading them on the GA truck, and h) reporting results of all tests to General Atomic.

2. TECHNICAL REQUIREMENTS

2.1 MATERIALS

2.1.1 The specified compressive strength of concrete shall be 4500 psi at age 14 days, the slump shall be $3 \pm 1/2$ inch and the maximum aggregate size shall be $3/8$ in. and air content shall be $7 \pm 2\%$. Suggested mix proportions are given in Table 2.

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

2.1.2 Portland cement shall comply with ASTM C150, Type III. If Type III is not available, Type I may be used, subject to approval by GA. All of the cement needed for the job shall be purchased in one lot and shall be blended by placing one scoopful at a time from each bag into a suitable weathertight container.

2.1.3 Fine aggregate shall meet the grading of ASTM C33. Coarse aggregate shall meet ASTM C33 size 8 (3/8 in. to No. 8). The sodium sulfate soundness loss of each aggregate (per ASTM C88) shall not exceed 8% after 5 cycles.

2.1.4 Prior to casting aggregate of each size shall be tested for sieve analysis per ASTM C136, for specific gravity and absorption per ASTM C127 or C128 and for 5 cycles of sodium sulfate soundness per ASTM C88. The fineness modulus of the fine aggregate shall be calculated per ASTM C125. The dry rodded unit weight of the coarse aggregate shall be determined per ASTM C29. The moisture content of the aggregate shall be determined per ASTM C566 prior to batching.

2.1.5 The air entraining admixture shall conform to ASTM C260 and the water-reducing admixture shall conform to ASTM C494, Type A. The dosage shall be as recommended by the manufacturer.

2.1.6 The reinforcing bar shall conform to ASTM A615, grade 40, No. 3 or No. 4 and shall be cut into 48-in. lengths.

2.2 CASTING OF SPECIMENS:

2.2.1 Mixing, casting and curing shall be per ASTM C192. Mixing shall be done in a suitable mechanical mixer. Just prior to each casting, the mixer shall be "buttered" with a throwaway batch. The admixtures shall be diluted with a small amount of mixing water and shall be added to the concrete after all other materials have been mixed for a period of 15 to 20 seconds.

2.2.2 Batch weights, aggregate moisture content, concrete temperature, slump per ASTM C143, air content per ASTM C231 and unit weight per ASTM C138 shall be determined and recorded for each batch.

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

2.2.3 Molds for the 6 by 12-in. cylinders shall be provided by the laboratory, shall be leak proof and shall be made of metal per ASTM C193 or C470. Paper or cardboard molds shall not be used.

2.2.4 Molds for the beams will be provided by GA. All joints shall be sealed with a suitable material to be provided by the laboratory. Surfaces of molds except for the embedments, shall be coated with mineral oil or a bond-breaking agent, to be supplied by the laboratory. The reinforcing bar shall be installed within ± 0.5 in. from the center of the specimen and shall be securely tied in place with wire or by other means.

2.2.5 The specimens shall be cast in the number of layers specified in ASTM C192, consolidated by rodding and finished with a wood or metal float. Special care shall be taken to insure that the concrete surface formed by the bottom of the mold is smooth and free from defects.

2.3 CURING AND TESTING:

2.3.1 All specimens shall be stripped and cured per ASTM C192 until they are at least 14 days of age.

2.3.2 One 6 by 12-in. cylinder from each casting shall be tested for compressive strength per ASTM C39 at age 7 and 14 days, as indicated in Table 1. The specimens from casting #1 and 2 shall also be tested for modulus of elasticity per ASTM C469 at age 14 days.

2.4 REPORT:

2.4.1 The laboratory shall submit to GA a report containing the following information:

- a) Reference to GA purchase order number.
- b) Brand and type of cement and admixtures; grading, specific gravity, absorption and soundness of aggregate; and the dry rodded unit weight of coarse aggregate.
- c) Mix proportions, number of batches, casting dates, compressive strengths and moduli of elasticity.

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No.

904078

Issue

A

2.5 ACCEPTANCE CRITERIA:

2.5.1 The job shall be considered complete after all the beams have been set on pallets and loaded on the GA truck, the 6 by 12-in. specimens have achieved a compressive strength of at least 4500 psi based on the average of the four 14-day specimens and the report of paragraph 2.4 has been received and found satisfactory by GA.

2.5.2 Specimens which fail to reach the 4500 psi strength (based on cylinder tests), which are damaged by the laboratory through careless handling, or which, after removal of molds show evidence of major defects shall be recast or repaired by the laboratory at no additional cost to GA, unless accepted as is by GA engineers.

3. QUALITY ASSURANCE REQUIREMENTS

3.1 GENERAL:

3.1.1 All laboratory work relevant to the fabrication of the beams shall be in accordance with applicable ASTM standards and with this specification.

3.1.2 General Atomic shall have the right, at any time, to witness inspections and tests, perform product inspections, witness work operations in progress and review quality documentation and records at the testing laboratory.

3.1.3 All specimens shall be properly marked and shall be traceable to the batch and date of casting.

3.1.4 Measuring devices used for inspection, test, and product acceptance shall be calibrated. Standards used for calibration shall be traceable to the National Bureau of Standards.

3.1.5 The laboratory shall notify GA five days prior to each casting.

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

TABLE 1. SUMMARY OF CASTINGS

Description	Beam No.	No. of 6x12-in. Cylinders	Casting Schedule, Weeks After P. O.
Casting No. 1	1,2,3,4,5,6	2**	2
Casting No. 2	7,8,9,10,11,12	2**	2.5
Casting No. 3	13,14,15,16,17,18	2*	3
Casting No. 4	19,20,21,22,23,24	2*	3.5

*Test one cylinder at age 7 days and one at age 14 days for compressive strength.

**Test one cylinder at age 14 days for modulus of elasticity; use the same cylinder for compressive strength test

TABLE 2. SUGGESTED MIX PROPORTIONS

Material	Quantity lb per cubic yard
Cement, ASTM C150, Type III	755
Water, potable	340*
Fine aggregate, ASTM C33	1440
Coarse aggregate, ASTM C33, 3/8 in. max.	1150
Air-entraining admixture, ASTM C260	**
Water-reducing admixture, ASTM C494, Type A	***

*Adjust to have $3 \pm 1/2$ in. slump.

**Dosage as recommended by manufacturer for $7 \pm 2\%$ air.

***Dosage as recommended by manufacturer.

Notations in this column indicate where changes have been made

GENERAL ATOMIC COMPANY

GA-1484

TITLE:

Solar Panels - Beams for Freeze-Thaw Tests

Document No. 904078

Issue A

FIG. 1A - BEAM FOR FREEZE-THAW TEST

FIG. 1B - STEEL MOLD.

Figure 1. - Beam and Molds

SOLAR PANELS - BEAMS FOR FREEZE - THAW TESTS

Doc. No. 904078 Issue A

General Atomic Company