SOLAR POWER SYSTEM
AND COMPONENT
RESEARCH PROGRAM
(January 15 to November 15, 1974)

Supported by the National Science Foundation
Research Applied to National Needs (RANN)

Any opinions, findings, conclusions or recommendations expressed in this publication are those of the author(s) and do not necessarily reflect the views of the National Science Foundation.

Prepared by

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The Solar Power System and Component Research program is authorized under National Science Foundation RANN Grant AER 74-07570, with a period of performance from Jan 15, 1974 to Jan 31, 1975. NSF program manager is Mr. George M. Kaplan of the Advanced Energy Research and Technology Division. The program includes both system analysis and component design for a 100-MWe solar energy conversion power system. The component design phase is concentrated on the boiler/superheater steam generation equipment. Martin Marietta Aerospace (as prime contractor) and Georgia Institute of Technology (as major subcontractor) lead the team of cooperating organizations performing the program. The final report covers the system definition and performance analyses performed during the program. Additionally, results from the user application analyses, meteorological analysis, economic analysis, and boiler superheater component design are included.
The Solar Power System and Component Research program is authorized under National Science Foundation Grant AER 74-07570, having a period of performance of January 15, 1974 to January 31, 1975. The NSF program manager is Mr. George M. Kaplan of the Advanced Energy Research and Technology Division. The program includes both system analysis and component design for a 100-MWe solar-energy-conversion power system with the component design phase concentrated on the boiler/superheater steam-generation equipment. Martin Marietta Corporation (as prime contractor) and Georgia Institute of Technology (as major subcontractor) lead the team of cooperating organizations performing the program. Mr. Floyd A. Blake of Martin Marietta, principal investigator, and Mr. Jesse D. Walton of Georgia Tech lead the technical program segments.

Program activities for the initial six months are covered in the Semi-Annual Progress Report No. 1, NSF/RANN/SE/GI-41305/PR/74/2, dated July 1974. This final report covers the activities of the final four months of the program in detail, as well as sufficient highlight information from the prior six months to provide perspective on the total program.

Activities during the final four-month period included:

Refinement of the performance analysis of the 100 MWe system;

Expansion of the data base for the solar/hydroelectric application;

Expansion of the meteorological modeling of solar insolation;

Completion of the detail thermal analysis of the bench model steam generator;

Detail design of the bench model steam generator;

Layout design of the test installation in the CNRS furnace for the bench model steam generator, the circulation and control system, and the instrumentation console;

Preliminary design of the small heliostat array, the 1-5 MWth Solar Facility, and the 12.5 MWe proof-of-concept experiment envisioned to implement the NSF five-year plan.
The Martin Marietta activity was supported by:

- Mr. T. R. Tracey
- Dr. M. T. Howerton
- Mr. C. J. Chocol
- Mr. M. L. Clevett
- Mr. H. C. Hunter
- Mr. S. W. Reusser
- Miss S. A. Stadjuhar
- Mr. R. M. Ballantyne
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- Mr. G. A. Mentgen
- Dr. A. N. Silver
- Mr. P. J. Grosser
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The Georgia Tech activity was supported by:

- Dr. S. H. Bomar, Jr.
- Dr. C. W. Gorton
- Dr. C. T. Brown
- Dr. J. H. Murphy

Collaborating organization participants to date have included:

- Professor F. Trombe and M. Claude Royere, Centre National de la Recherche
- Mr. Russell Humphreys, Salt River Project
- Mr. Harry Garretson and Mr. Hugh Kasai, Bonneville Power Administration
- Mr. Don von Fossen and Mr. Robert Lee, Babcock & Wilcox
- Mr. John I. Yellott, Yellott Engineering Associates
## CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Pages</th>
</tr>
</thead>
<tbody>
<tr>
<td>I. INTRODUCTION</td>
<td>I-1</td>
</tr>
<tr>
<td>A. Program Objectives</td>
<td>I-1</td>
</tr>
<tr>
<td>B. Program Plan--Work Breakdown and Schedule</td>
<td>I-4</td>
</tr>
<tr>
<td></td>
<td>thru</td>
</tr>
<tr>
<td></td>
<td>I-6</td>
</tr>
<tr>
<td>II. 100-MWe SOLAR-ENERGY-CONVERSION POWER SYSTEM</td>
<td>II-1</td>
</tr>
<tr>
<td>A. Baseline Features of Solar-Energy-Conversion Plant Subsystems</td>
<td>II-1</td>
</tr>
<tr>
<td>B. Design Configuration of the 100 MWe Solar-Energy-Conversion Power Plant</td>
<td>II-6</td>
</tr>
<tr>
<td>C. Projected Performance of 100 MWe Solar Power System</td>
<td>II-13</td>
</tr>
<tr>
<td>D. References</td>
<td>II-51</td>
</tr>
<tr>
<td>III. USER APPLICATION ANALYSES--A DIRECT SOLAR AUGMENTATION BASED ON SALT RIVER PROJECT OPERATIONS</td>
<td>III-1</td>
</tr>
<tr>
<td>A. Background Discussion of System Composition and Constraints</td>
<td>III-1</td>
</tr>
<tr>
<td>B. Solar Conversion Plant Performance, Referenced to Operating System</td>
<td>III-3</td>
</tr>
<tr>
<td></td>
<td>thru</td>
</tr>
<tr>
<td></td>
<td>III-10</td>
</tr>
<tr>
<td>IV. USER APPLICATION ANALYSES--INTERTIE AUGMENTATION BASED ON BONNEVILLE POWER ADMINISTRATION MODEL</td>
<td>IV-1</td>
</tr>
<tr>
<td>A. Study of Candidate Solar Plant Technical Compatibility</td>
<td>IV-1</td>
</tr>
<tr>
<td>B. Pacific Northwest--Pacific Southwest Interies</td>
<td>IV-7</td>
</tr>
<tr>
<td></td>
<td>thru</td>
</tr>
<tr>
<td></td>
<td>IV-11</td>
</tr>
<tr>
<td>V. ECONOMIC ANALYSIS</td>
<td>V-1</td>
</tr>
<tr>
<td>A. Construction Costs</td>
<td>V-1</td>
</tr>
<tr>
<td>B. Operating Costs</td>
<td>V-1</td>
</tr>
<tr>
<td>C. Basis for Cost Estimates</td>
<td>V-3</td>
</tr>
<tr>
<td>D. Construction and Operation Cost Itemization</td>
<td>V-6</td>
</tr>
<tr>
<td>E. Air Cooled Condenser Size Versus Number of Heliostats Cost Tradeoff</td>
<td>V-17</td>
</tr>
<tr>
<td>F. References</td>
<td>V-19</td>
</tr>
<tr>
<td>VI. STEAM GENERATOR DESIGN AND ANALYSIS</td>
<td>VI-1</td>
</tr>
<tr>
<td>A. Overall System Analysis</td>
<td>VI-1</td>
</tr>
<tr>
<td>B. Performance Analysis of the 100-MWe Cavity</td>
<td>VI-12</td>
</tr>
<tr>
<td>C. Design of the Bench Model System and Cavity</td>
<td>VI-22</td>
</tr>
<tr>
<td></td>
<td>thru</td>
</tr>
<tr>
<td></td>
<td>VI-58</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>---------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>I-1</td>
<td>Program Team for Solar Power System and Component Research</td>
</tr>
<tr>
<td>I-2</td>
<td>Program Schedule for Solar Power System and Component Research</td>
</tr>
<tr>
<td>II-1</td>
<td>Position of Concentrating Heliostat Along Solar Concentration/Temperature Spectrum</td>
</tr>
<tr>
<td>II-2</td>
<td>Martin Marietta Concentrating Heliostat Optical Accuracy Development Prototype, in Operation</td>
</tr>
<tr>
<td>II-3</td>
<td>Cavity Configuration Central Receiver Steam Generator of Prof. G. Francia's Solar Pilot Plant at Genoa, Italy, in Operation</td>
</tr>
<tr>
<td>II-4</td>
<td>1/32-Scale Model of Solar Power System Steam Generator Cavity</td>
</tr>
<tr>
<td>II-5</td>
<td>100 MWe Solar Plant Layout</td>
</tr>
<tr>
<td>II-6</td>
<td>Heliostat Field Elevation Profile</td>
</tr>
<tr>
<td>II-7</td>
<td>Heliostat Field Plan, 12.5 MWe Solar Collector Field</td>
</tr>
<tr>
<td>II-8</td>
<td>Conceptual Layout of 100-MWe Receiver</td>
</tr>
<tr>
<td>II-9</td>
<td>Heat Balance, 100 MW System</td>
</tr>
<tr>
<td>II-10</td>
<td>Specular Reflectivity Test Rig Using Normal Incidence Pyrheliometers, Profile View</td>
</tr>
<tr>
<td>II-11</td>
<td>Specular Reflectivity Test Rig Using Normal Incidence Pyrheliometers, Front View</td>
</tr>
<tr>
<td>II-12</td>
<td>Reference Pyrheliometers Used for Solar Insolation Monitoring during Specular Reflectivity Tests</td>
</tr>
<tr>
<td>II-13</td>
<td>National Climatic Center Annual Sunshine Hours Map with &quot;Sun Bowl&quot; Accented</td>
</tr>
<tr>
<td>II-14</td>
<td>Heliostat R38-28L, 8 a.m., March 21, Cosine = 0.91211 (24.20°)</td>
</tr>
<tr>
<td>II-15</td>
<td>4x4-Foot Mirror, Masked Except for Center and Corners to Enable Optical Adjustment</td>
</tr>
<tr>
<td>II-16</td>
<td>Five Solar Discs from Partially Adjusted 4x4-Foot Mirror</td>
</tr>
<tr>
<td>II-17</td>
<td>Five Solar Discs Overlapped after Final Optical Adjustment</td>
</tr>
<tr>
<td>II-18</td>
<td>Heliostat R38-28L, 10 a.m., March 21, Cosine = 0.92662 (22.14°)</td>
</tr>
<tr>
<td>II-19</td>
<td>Heliostat R38-28L, 12 Noon, March 21, Cosine = 0.8850 (27.31°)</td>
</tr>
<tr>
<td>II-20</td>
<td>Heliostat R38-28L, 2 p.m., March 21, Cosine = 0.8018 (36.69°)</td>
</tr>
<tr>
<td>II-21</td>
<td>Heliostat R38-28L, 4 p.m. March 21, Cosine = 0.67697 (47.39°)</td>
</tr>
<tr>
<td>II-22</td>
<td>Optical Ray Tracing Pattern Referenced for Warp Accuracy Correlation</td>
</tr>
<tr>
<td>II-23</td>
<td>Image of 19.5-in. Curved Heliostat Sample Mirror at a 65-ft Focal Distance (2-in. Grid Spacing)</td>
</tr>
</tbody>
</table>
II-24 Optical Accuracy Development Concentrating Heliostat; Seven 4x4-ft Second Surface Glass Mirrors Warped for 108-ft Focal Distance  
II-25 Wheel Curvature Control Structure Installed on 4x4-ft Second Surface Glass Mirror, 108-ft Focal Length  
II-26 Optical Image from Wheel Structure Controlled 4x4-ft Mirror, Concentration = 5.32:1  
II-27 Seven Mirror Concentrating Heliostat Masked for Center Ray Reflection Optical Pattern Testing to Study Effective Spherical Aberration  
II-28 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 9 a.m. MDT (8 a.m. TST) May 29, 1974  
II-29 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 10 a.m. MDT (9 a.m. TST) May 29, 1974  
II-30 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 11 a.m. MDT (10 a.m. TST) May 29, 1974  
II-31 Overlay of Test Observation Images on Computer Pattern, Noon MDT (11 a.m. TST) May 29, 1974  
II-32 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 1 p.m. MDT (Noon TST) May 29, 1974  
II-33 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 2 p.m. MDT (1 p.m. TST) May 29, 1974  
II-34 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 3 p.m. MDT (2 p.m. TST) May 29, 1974  
II-35 Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 4 p.m. MDT (3 p.m. TST) May 29, 1974  
III-1 Salt River Project Hydroelectric Installation  
III-2 Salt River Project and Potential Solar Plant Profiles, July 2, 1973  
III-3 Solar Plant Operating Time, Total Sunshine Hours and Monthly Operating Days for a Solar Plant for a 1-yr Period in the Phoenix, Arizona, Area  
III-4 Summary of Sample Weeks Power Generation and Operating Periods Against Year-Long Time Scale Referenced Against Daily Temperature Pattern  
IV-1 Electric Power Plant Locations in Pacific Northwest  
IV-2 John Day Navigation Lock, Spillway, and Power Plant  
IV-3 Interior View of First Eight Generator Units at The Dalles Power Plant  
IV-4 Grand Coulee Dam and Power Plants  
IV-5 Major Electrical Transmission Network of Western U.S.
Table

II-1 Parameters for 100-MWe Solar Power Plant .......... II-11
II-2 Alternative Thermodynamic Cycles and Resultant Solar Collector Field Sizing Parameter .......... II-12
II-3 Time-of-Year Effect on Performance Factors - 100-MWe Solar Power System ............. II-14
II-4 Area Loss Performance for All-North Field Solar Collector Configuration, Equinox Date, at 33.6°N, 111.3°W (Horse Mesa, Arizona) Location .......... II-18
II-5 Area Loss Performance for All-North Field Solar Collector Configuration, Summer Solstice Date (Day 172), at 33.6°N, 111.3°W (Horse Mesa, Arizona) Location .......... II-19
II-6 Area Loss Performance for All-North Field Solar Collector Configuration, Winter Solstice (Horse Mesa, Arizona) Location .......... II-20
II-7 Specular Reflectivity Measured by Normal Incidence Pyrheliometer Test Rig .......... II-24
II-8 Comparison of North- and South-Field Heliostat Efficiencies for Central Tower Collector Field Average Daily Values for Equinox for 33.6°N, 111.3°W Location (Horse Mesa, Arizona) .......... II-35
III-2 Summary of Sunshine Monitor Data from Salt River Project, August 1973-1974 .......... III-7
III-3 Performance Summary of 13-Week Operational Data for 100-MWe Solar Conversion Power Plant Associated with Salt River Project (Phoenix, Arizona) Power System .......... III-9
IV-1 Major Hydroelectric Installations Associated with Bonneville Power Administration in U.S. Pacific Northwest .......... IV-1
<table>
<thead>
<tr>
<th>VI-1</th>
<th>100-MW Cavity Performance</th>
<th>VI-12</th>
</tr>
</thead>
<tbody>
<tr>
<td>VI-2</td>
<td>Full-Scale Receiver Insolation, Nodal Distribution</td>
<td>VI-14</td>
</tr>
<tr>
<td>VI-3</td>
<td>System Energy Balance, Full-Scale Receiver</td>
<td>VI-16</td>
</tr>
<tr>
<td>VI-4</td>
<td>Components</td>
<td>VI-26</td>
</tr>
<tr>
<td>VI-5</td>
<td>Instrumentation List and Equipment Required</td>
<td>VI-28</td>
</tr>
<tr>
<td>VI-6</td>
<td>Temperature-Dependent Properties of Water</td>
<td>VI-37</td>
</tr>
<tr>
<td>VI-7</td>
<td>Total Incident and Absorbed Solar Energy per Node for Superheater, kW</td>
<td>VI-53</td>
</tr>
<tr>
<td>VI-8</td>
<td>Total Incident and Absorbed Solar Energy per Node for Boiler, kW</td>
<td>VI-53</td>
</tr>
<tr>
<td>VI-9</td>
<td>Total Incident and Absorbed Solar Energy per Node for Preheater, kW</td>
<td>VI-54</td>
</tr>
<tr>
<td>VI-10</td>
<td>System Energy Balance</td>
<td>VI-57</td>
</tr>
</tbody>
</table>
I. Introduction
I. INTRODUCTION

The Solar Power System and Component Research program is authorized under National Science Foundation Grant AER 74-07570, with a period of performance of January 15, 1974, to January 31, 1975. The NSF program manager is Mr. George M. Kaplan of the Advanced Energy Research and Technology Division. The program includes both system analysis and component design for a 100-MWe solar-energy-conversion power system with the component design phase concentrated on the boiler/superheater steam-generation equipment. Martin Marietta Corporation (as prime contractor) and Georgia Institute of Technology (as major subcontractor) lead the team of cooperating organizations performing the program (Fig. I-1). Mr. Floyd A. Blake of Martin Marietta, principal investigator, and Mr. Jesse D. Walton of Georgia Tech lead the technical program segments.

This final report covers the total program activities with detail emphasis on the final four months of activities that have not been previously reported.

A. PROGRAM OBJECTIVES

Two broad objectives were established for the program at the outset.

1) To evaluate the potential utilization of solar-thermal-conversion central receiver-hydroelectric hybrid power systems.

2) To define requirements for a bench-model central receiver test program consistent with the CNRS Solar Energy Laboratory's 1-MW thermal furnace at Odeillo, France.

3) An additional objective to define program and facility specifications necessary to accomplish technology development leading to the proof-of-concept-experiment pilot plant for central receiver thermal power systems was established at the customer's request during this quarter.
Martin Marietta Corporation
Denver, Colorado
System Analysis
Thermal Analysis & Design
Bench Models

Georgia Institute of Technology
Atlanta, Georgia
Structural Analysis &
Physical Design of Bench
Models
Test Installation Design
Test Plan

Consultant
John I. Yellott, Phoenix, Ariz

Cooperating Organizations
Salt River Project
Phoenix, Arizona
Application Data
Program Advice/Review

Bonneville Power Administration
Portland, Oregon
Application Data
Program Advice/Review

Southern California Edison
Rosemead, California
Program Advice/Review

Collaborating Organizations
Centre National de la Recherche Scientifique Solar Laboratory
Odeillo, France
Counseling on Bench Model
Test Installation & Test Plan

Babcock and Wilcox
Alliance, Ohio
Consulting Critique in Areas of Boiler Code Adherence & Manufacturing Practice

Fig. I-1 Program Team for Solar Power System and Component Research
The solar/hydroelectric hybrid power system was selected for analysis in this program because it is a promising near-term energy displacement and load-following utilization of solar thermal power systems. The primary factors supporting this choice are the increased relative importance of hydroelectric installations in the western states most richly endowed with solar energy, the potential for increased generation capacity without an increase in water usage, and the desirability of increasing the range of power systems being evaluated in the NSF/RANN solar thermal program.

The cooperating organizations for the user application analyses include the Salt River Project of Phoenix, Arizona, whose system is only partially hydroelectric, and the Bonneville Power Administration of Portland, Oregon, whose system is dominantly hydroelectric. A 100-MWe solar energy conversion power plant was evaluated relative to its contribution when integrated into the Salt River Project system and a 1000 MWe solar plant was evaluated as a part of the Bonneville system. The potential application for the Salt River Project presumes a geographical location within that system's territory, whereas that augmenting the Bonneville Power Administration is assumed to be sited in a favorable solar energy location and connected via the Pacific inter-ties.

The second objective focuses the component-design effort in this program on the boiler and superheater energy-conversion subsystem. Conceptual design of the full-scale steam generator to be used at the focal zone of each module of the 100 MWe solar plant established guidelines for the detail design of a bench scale steam generator sized for operation in the CNRS Solar Laboratory furnace. Collaborating organizations in this segment of the program include the Centre National de la Recherche Scientifique Solar Energy Laboratory of Odeillo, France, and Babcock and Wilcox Research Laboratory of Alliance, Ohio.

Activities during the final segment of the system analysis performed in support of objective 1 (above) included:

1) Refinement of the performance analysis of the 100 MWe system;

2) Expansion of the data base for the solar/hydroelectric application;

3) Expansion of the meteorological modeling of direct solar insolation.

Activities during the final segment of the 1 MWth Bench Model Cavity Receiver Steam Generator component design accomplished in support of objective 2 included:

1) Completion of the detail thermal analysis of the bench model steam generator;
2) Detail design of the bench model steam generator;

3) Layout design of the test installation in the CNRS furnace for the bench model steam generator and supporting circulation, control, and instrumentation systems.

Elements of the long-range planning study carried out within the guidelines of objective 3 were:

1) Preliminary design of the small heliostat array, the 1-5 MWth solar facility, and the 12.5 MWe proof-of-concept-experiment envisioned to implement the NSF five-year plan;

2) Definition of key objectives for the program stages needed to accomplish the individual program stages of the NSF five-year plan.

This report presents the results of these specific activities together with pertinent results obtained during the initial six-month segment of the grant-supported program.

B. PROGRAM PLAN--WORK BREAKDOWN AND SCHEDULE

An overview of the program plan tasks and schedule is presented in Fig. 1-2. Specific discussions of the elements currently underway will be presented in subsequent chapters.

The system-analysis segment of the program has four major tasks.

1) System definition and performance analysis of a 100-MWe solar power system.

2) User application analyses of direct hydroelectric integration, tie-line integration, and industrial augmentation systems.

3) Economic analyses to establish both capital cost and operating-cost characteristics of the solar energy conversion system.

4) Meteorological analyses to model direct vs total isolation data for Phoenix, Arizona, and to relate the Inyokern/China Lake data from the Aerospace Corporation program to the performance of the baseline design system.
## Program Schedule for Solar Power System and Component Research

<table>
<thead>
<tr>
<th>Task</th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>Jun</th>
<th>Jul</th>
<th>Aug</th>
<th>Sep</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1.0 Program Management</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>2.0 System Analysis</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.1 100-MWe Definition &amp; Performance</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>2.2 User Application Analysis</td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>2.2.1 Direct Hydro</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>2.2.2 Tie-Line Hydro</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>2.2.3 Industrial</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td><strong>2.3 Economic Analysis</strong></td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<td><strong>2.4 Meteorological Analysis</strong></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
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<tr>
<td><strong>3.0 Boiler &amp; Superheater Design</strong></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>3.1 Analysis</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<tr>
<td>3.1.1 Heat-Exchange Interface</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<tr>
<td>3.1.2 Secondary Collector</td>
<td></td>
<td></td>
<td></td>
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<td>3.1.3 Boiler Tube Bundle</td>
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<td>3.1.4 Superheater</td>
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<td>3.1.5 Flow Hardware</td>
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<td><strong>3.2 Design, Structural</strong></td>
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<td>3.2.1 Boiler &amp; Superheater Stress Analysis</td>
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<td>3.2.2 Flow Hardware</td>
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<td>3.2.3 Boiler &amp; Superheater Drawings</td>
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<tr>
<td><strong>3.3 Boiler Code &amp; Practice Adherence</strong></td>
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<td><strong>3.4 Test Installation Drawings</strong></td>
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<td><strong>3.5 Test Plan</strong></td>
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<td>3.5.1 French Furnace Adaptation</td>
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<tr>
<td>3.5.2 Bench &amp; Power Tests</td>
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</tbody>
</table>

**Legend:**
- Planned Period of Performance
- ▲ Data from Martin Marietta Available
- ▼ Task Complete: Data Package to Martin Marietta
- ▽ Support Data to Martin Marietta Required

*Fig. I-2 Program Schedule for Solar Power System and Component Research*
Work on the steam generator designs was divided along thermophysics analysis and mechanical design analysis lines with the responsibility for the segments assigned to Martin Marietta and Georgia Tech, respectively. Boiler code and practice adherence was analyzed by Georgia Tech with support from Babcock and Wilcox during the conceptual design stage, and Martin Marietta during the detail design stage.

Test installation engineering and test planning was also Georgia Tech's responsibility, and was undertaken in collaboration with the CNRS Solar Energy Laboratory. Elements of this activity were performed both in France and in the United States, during visits of Mr. Walton and Dr. Bomar to Odeillo, and visits of Professor Trombe and M. Royere to Atlanta and Denver.
II. 100-MWe Solar-Energy-Conversion Power System Configuration and Performance
II. 100 MWe SOLAR-ENERGY-CONVERSION POWER SYSTEM CONFIGURATION AND PERFORMANCE

A. BASELINE FEATURES OF SOLAR-ENERGY-CONVERSION PLANT SUBSYSTEMS

Selection of the basic system characteristics to be optimized in implementing the "point design" central receiver/optical-transmission solar energy conversion power system resulted from the in-house study and proposal analyses that led to the granted program. These included establishing the following baselines:

1. Solar Collector/Concentrator

The concentrating heliostat was established as the baseline solar collector to accomplish both the large-area energy collection and the necessary concentration required to convert the solar energy to steam energy at temperatures consistent with modern power plants. The relative optical performance of the heliostat is shown in Fig. II-1, referenced against the demonstrated state of the art for alternative concentrators (Ref II-1 thru II-7). Considerable design flexibility exists with the heliostat, in that concentrations as low as 200 could be used without jeopardizing the 1000°F steam-generation capability. The experimental prototype concentrating heliostat built by Martin Marietta to demonstrate the concept and develop optical accuracy cost effectively is shown in Fig. II-2.

2. Central Cavity Configuration Heat Receiver

Historical precedent and the potential for attaining highly efficient thermal energy conversion combined to establish the cavity configuration for the heat receiver. All the solar thermal programs undertaken during the power research programs of the space program used the cavity for its performance benefits. Additionally, the only solar-powered steam generators used to obtain temperature-pressure conditions suitable for modern turbines---those of Prof. Francia at Genoa, Italy---used cavity configurations (Ref II-6), as shown in Fig. II-3. Among the most desirable features of the cavity are its near-black-body absorption, the flexibility of designing the interior to attain nearly uniform flux at the heat-conversion interface, and its ease of insulation. Figure II-4 shows a 1/32-scale model of the boiler cavity.
Fig. II-1 Position of Concentrating Heliostat Along Solar Concentration/Temperature Spectrum
Fig. II-2
Martin Marietta Concentrating Heliostat Optical Accuracy Development Prototype, in Operation (Illuminated Smoke from Igniting Canvas)
Fig. II-3
Cavity Configuration Central Receiver Steam Generator of Prof. G Francia's Solar Pilot Plant at Genca, Italy, in Operation (Photo Courtesy of Georgia Tech)
Fig. II-4 1/32-Scale Model of Solar Power System Steam Generator Cavity
3. **Air-Cooled Condenser**

To maximize the environmental desirability of the solar-conversion power plant, and in acknowledgement of the extreme water shortage normally accompanying abundant solar isolation sites, the baseline design will use an air-cooled condenser and carry the economic and performance penalties that result. (This does not preclude the use of a water-cooled condenser should industrially allocated water, such as that of the Central Arizona Project, be available at the solar site.)

4. **Single-Expansion Steam Turbine**

Based on a preliminary evaluation of the turbines suitable for cycling operation, and the problems attendant with out-of-plant steam transmission, the thermodynamic cycles considered were limited to those featuring a single expansion from superheated steam. As will be discussed later, the efficiency improvement from reheating would largely be negated by transmission losses.

5. **Sizing of 100-MWe Solar-Conversion Power Plant**

The sizing of the solar plant for the study was established based on information fed back from electrical utility representatives on the NSF Review Board and present at presentations covering the concept. For smaller plants the operational cost per kWh increases sharply due to the higher manpower costs.

Based on the above guidelines, we established a group of goals that were continually referenced in the design decision process. These goals were primarily driven by the desire to achieve economic optimization and had a direct influence on the resulting design configuration. The following sections of this chapter cover the 100 MWe solar plant configuration, its projected performance, detailed discussion of the design goals, and substantiating data on the performance of individual elements of the collection/concentration subsystem.

### B. DESIGN CONFIGURATION OF THE 100 MWe SOLAR-ENERGY-CONVERSION POWER PLANT

The overall layout plan for the 100 MWe solar-energy-conversion power plant is shown in Fig. II-5. Eight solar collector fields are located around a central steam-turbine-generation power plant and control center. The power plant would be similar to the air-cooled condensing plant of the Black Hills Power and Light Company at Wyodak, Wyoming, shown in the inset. Each solar collector field is 1400x1400-ft square (45.0 acres) on a site having a
Fig. II-5  100 MWe Solar Plant Layout
natural slope to the south of 10 to 13° to accommodate terracing with minimum excavation. The contour profile is shown in Fig. II-6.

Conversion of the solar energy into superheated steam at 1250 psig, 950°F is accomplished in the cavity-mounted boiler/superheater atop the 400-ft tower at the center point of the south border. High density mirror packing, attainable with minimum shadowing, due to the terracing, is defined in Fig. II-7.

The criterion for minimum shadowing was based on zero row-to-row shadowing for solar elevation angles above 20°, which corresponds to attaining the threshold illumination established by the Phoenix, Arizona, test data.

The design parameters of the 100-MWe solar plant are shown in Table II-1. The most striking of the features is the 14,720 individual heliostat mirrors required to collect and concentrate the incident solar energy. Each of the mirror fields contains 1840 heliostats, the current baseline design, but could accommodate up to 2002 within the current boundaries, should growth be necessary.

For perspective, the CNRS solar furnace has operated a coordinated field of 63 similar-sized mirrors since 1970.

Based on recommendations of turbine manufacturers, three thermodynamic cycles were evaluated during the preliminary system configuration design phase. The initial overriding requirement during the evaluation was to demonstrate the capability to perform in the cycling operations attendant with the solar application.

Table II-2 shows the identifying points of the cycles, the resultant heat rates, and the resultant sizing parameters imposed on the solar collector field. The sensitivity of the thermodynamic cycle is evident in that the reflecting surface of the mirror field ranges between 8.22 million ft² for the lowest (600 psig, 750°F) efficiency cycle and 5.87 million ft² for the highest (1250 psig, 950°F) efficiency cycle.

The 100-MWe system selected uses a single 100-MWe regenerative nonreheat turbine-generator. The steam inlet is at 1250 psig, 950°F and the thermodynamic efficiency is 35% \( \frac{\text{net electrical power out}}{\text{heat into steam}} \). The steam temperature and pressure are lower than used in present conventional steam plant design practice in order to ensure that the turbine can be designed to be compatible with the thermal cycling required in a solar plant application.
Fig. II-6 Heliostat Field Elevation Profile

Typical Cosine Values (Midday)

<table>
<thead>
<tr>
<th>Row</th>
<th>Summer</th>
<th>Winter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>.855</td>
<td>.994</td>
</tr>
<tr>
<td>15</td>
<td>.906</td>
<td>1.000</td>
</tr>
<tr>
<td>16</td>
<td>.910</td>
<td>.999</td>
</tr>
<tr>
<td>38</td>
<td>.991</td>
<td>.955</td>
</tr>
</tbody>
</table>

Fig. II-6 Heliostat Field Elevation Profile
**Data:**

- **Dimensions:** 1400' x 1400'
- **Area:** 1,960,000 ft.² = 45.0 acres
- **No. Heliostats:** 2002
- **Mirror Surface:** 800,800 ft.²
- **Max. Azimuth Delta (Summer Solstice)**

<table>
<thead>
<tr>
<th>Position</th>
<th>Degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>105</td>
</tr>
<tr>
<td>38</td>
<td>82.75</td>
</tr>
<tr>
<td>602</td>
<td>100</td>
</tr>
<tr>
<td>1975</td>
<td>105</td>
</tr>
<tr>
<td>2002</td>
<td>110.5</td>
</tr>
</tbody>
</table>

**Fig. II-7 Heliostat Field Plan, 12.5 MWe Solar Collector Field (East Half Shown) (West Half Opposite)**
<table>
<thead>
<tr>
<th>Area</th>
<th>Parameters for 100-MWe Solar Power Plant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Energy Collection&lt;br&gt;Modules (8)</td>
<td>405 Acres (1.64 km²)</td>
</tr>
<tr>
<td>Power Plant (Central)&lt;br&gt;Site</td>
<td>45 Acres (426 x 426 m = 0.182 km²)</td>
</tr>
<tr>
<td>1400 x 1400 ft = 45 Acres (426 x 426 m = 0.182 km²)</td>
<td></td>
</tr>
<tr>
<td>Turbine Generators&lt;br&gt;One Unit in Central&lt;br&gt;Plant</td>
<td>100-MWe (141,000 SHP) Rating</td>
</tr>
<tr>
<td>Inlet Conditions</td>
<td>1250 psig, 950°F (510°C) Steam</td>
</tr>
<tr>
<td>Solar Collector System&lt;br&gt;Heliostat Assemblies</td>
<td>14,720 (400 ft² (37.2 m²) Mirror on Each, Total Mirror Area of 5,870,000 ft² (545,000 m²))</td>
</tr>
<tr>
<td>Functional Module Fields</td>
<td>8</td>
</tr>
<tr>
<td>Rated Solar Insolation&lt;br&gt;Threshold</td>
<td>1.234 Langleys; 80 W/ft² (861 W/m²)</td>
</tr>
<tr>
<td>Heat Exchange System&lt;br&gt;[Atop 400-ft (121.9-m) Towers]}</td>
<td>8</td>
</tr>
<tr>
<td>Boiler/Superheater Units Integrated into Cavity Configuration</td>
<td>Unit Superheater Area: 850 ft² (78.97 m²)</td>
</tr>
<tr>
<td>Unit Boiler Area: 1,890 ft² (175.6 m²)</td>
<td></td>
</tr>
<tr>
<td>Unit Preheater Area: 3,020 ft² (280.6 m²)</td>
<td></td>
</tr>
<tr>
<td>Steam Rate (54,793 kg/hr)</td>
<td>120,800 lb/hr</td>
</tr>
<tr>
<td>Outlet Steam Conditions</td>
<td>1,300 psig, 955°F (513°C)</td>
</tr>
<tr>
<td>Cycle</td>
<td>Peak Cycle Pressure, psig</td>
</tr>
<tr>
<td>-------</td>
<td>--------------------------</td>
</tr>
<tr>
<td>1</td>
<td>600</td>
</tr>
<tr>
<td>2</td>
<td>850</td>
</tr>
<tr>
<td>3</td>
<td>1250</td>
</tr>
</tbody>
</table>

*Applicable for a solar insolation of 80 W/ft² (1.234 Langley) and a solar energy thermal collection/conversion efficiency of 60.6%.
Converting the concentrated solar energy into thermal energy in the working fluid occurs in eight tower-mounted cavity configuration boiler/superheaters. Each of the eight cavities for the 100-MWe system has a 20x20-ft opening, 30-ft deep, 40-ft wide and 40-ft high, as shown in Fig. II-8. An insulated door protects the cavity from the environment during shutdown periods and drastically reduces the temperature drop during shutdowns. For example, during a one-hour shutdown due to cloud cover the cavity average temperature will only drop about 10°F. During an overnight shutdown of 16 hours, the cavity temperature will drop about 160°F. The overall efficiency of the cavity design is 95% without using glass in the aperture. The extremely high efficiency of the cavity approach, along with the ability to drastically reduce temperature changes during shutdown periods, makes it a far superior design than "open" heat receivers.

C. PROJECTED PERFORMANCE OF 100 MWe SOLAR POWER SYSTEM

The thermal collection efficiency for the solar-energy-conversion power system has been defined as "the output from the heat exchange subsystem divided by the solar isolation intercepted by the full mirror area." Because this efficiency directly contributes to the sizing of the mirror field for any design, it is of primary importance that each factor contributing to it be optimized. The factors currently established for the optical loss group (i.e., the cosine of the tracking angle, surface reflectivity, and optical accuracy) and the thermal group (i.e., the convection, IR, and reflected radiation through the aperture, and insulation) are shown in Table II-3. The data are shown for the winter solstice, equinoxes, and summer solstice. The two factors affected by time of year are the area loss cosines and the optical accuracy, which is partially dependent on the cosines. Attainable thermal collection efficiency values range from 0.614 at the summer solstice to 0.763 at the winter solstice.

The waterfalling effect of the energy collection losses is shown in Fig. II-9.

1. Area Loss (Cosines) Factor

The initial factor of the optical group, the area loss factor, varies with both time of day and time of year. The values shown in Table II-3 and Fig. II-9 are the daily averages for the respective three days illustrating the time-of-year effect. A finer breakdown showing the time-of-day variation for the same three days is shown in Tables II-4, II-5, and II-6. Also shown is the varied performance of the mirror rows, taken at intervals across the field.
Table II-3  Time-of-Year Effect on Performance Factors - 100-MWe Solar Power System

<table>
<thead>
<tr>
<th></th>
<th>Day 355 Winter Solstice</th>
<th>Day 80/264 Equinox</th>
<th>Day 172 Summer Solstice</th>
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</thead>
<tbody>
<tr>
<td>Threshold Potential</td>
<td>0.944</td>
<td>0.895</td>
<td>0.779</td>
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<tr>
<td>Area Loss (Cosines)</td>
<td>444.5 th</td>
<td>421.6 th</td>
<td>367.2 th</td>
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<tr>
<td>Reflectivity</td>
<td>0.85</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>Optical Accuracy</td>
<td>1.0</td>
<td>1.0</td>
<td>0.975</td>
</tr>
<tr>
<td>Aperture Radiation</td>
<td>372.1 th</td>
<td>352.9 th</td>
<td>299.7 th</td>
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<tr>
<td>Convection Loss</td>
<td>0.986</td>
<td>0.968</td>
<td>0.968</td>
</tr>
<tr>
<td>Insulation Loss</td>
<td>359.5 th</td>
<td>340.9 th</td>
<td>289.6 th</td>
</tr>
<tr>
<td>Turbine-Alt Mech Eff</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
</tr>
<tr>
<td>Thermal Collector</td>
<td>0.763</td>
<td>0.724</td>
<td>0.614</td>
</tr>
<tr>
<td>Efficiency, Output</td>
<td>359.5 th</td>
<td>340.9 th</td>
<td>389.6 th</td>
</tr>
<tr>
<td>Potential System</td>
<td>0.267</td>
<td>0.253</td>
<td>0.215</td>
</tr>
<tr>
<td>Efficiency, Output</td>
<td>125.8 E</td>
<td>119.3 E</td>
<td>101.3 E</td>
</tr>
</tbody>
</table>
Steam Outlet to Superheaters

36-in. Dia Separator Drum

Boiler in Sections 1-in. OD Tubes Side by Side in Parallel

Superheater 1, 73 3/4-in. Tubes in Parallel

Boiler Sections 1-in. OD Tubes Side by Side in Parallel

12-in. Dia Header 6-in. Dia (4 Places)

8-in. Dia (4 Places)

Tube and Header Arrangement No Scale

Note: All cavity internal surfaces not shown for the boiler or superheater are covered with a ceramic sheet.

Fig. II-8 Conceptual Layout of 100-MW Receiver

11-15 and 11-16
<table>
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<tr>
<th>Loss Type</th>
<th>Value 1</th>
<th>Value 2</th>
<th>Value 3</th>
<th>Value 4</th>
<th>Value 5</th>
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<tbody>
<tr>
<td>Cosine Loss</td>
<td>377.8</td>
<td>372.1</td>
<td>360.2</td>
<td>359.5</td>
<td>358.3</td>
</tr>
<tr>
<td>Optical Accuracy</td>
<td>471.8</td>
<td>444.5</td>
<td>367.2</td>
<td>358.3</td>
<td>358.3</td>
</tr>
<tr>
<td>Reflected Loss</td>
<td>312.1</td>
<td>304.3</td>
<td>290.2</td>
<td>289.6</td>
<td>289.6</td>
</tr>
<tr>
<td>Turbulent Loss</td>
<td>125.8</td>
<td>119.3</td>
<td>115.4</td>
<td>117.6</td>
<td>118.8</td>
</tr>
<tr>
<td>Convection Loss</td>
<td>400.0</td>
<td>386.7</td>
<td>372.1</td>
<td>360.2</td>
<td>340.0</td>
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<tr>
<td>Mechanical Loss</td>
<td>536.5</td>
<td>538.3</td>
<td>532.2</td>
<td>529.7</td>
<td>529.7</td>
</tr>
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</table>

**Fig. II-9: Heat Balance, 100 MW System**

- A - Cosine Loss
- B - Reflectivity Loss
- C - Optical Accuracy
- D - Convection Loss
- E - Turbulent Loss
- F - Mechanical Loss

**Legend:**
- Potential Energy to System
- To Mirror Surface
- To Focal Zone
- Into Cavity Walls
- Into Cavity Walls
- Through Cavity Walls
- Into Steam
- Busbar Electricity

**Graph:**
- Heat Balance graph showing various energy losses.
<table>
<thead>
<tr>
<th>Heliostat Row</th>
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<tbody>
<tr>
<td></td>
<td>8 am</td>
<td>9 am</td>
<td>10 am</td>
<td>11 am</td>
<td>Noon</td>
<td>1 pm</td>
<td>2 pm</td>
<td>3 pm</td>
<td>4 pm</td>
<td></td>
<td></td>
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<tr>
<td>38</td>
<td>0.797</td>
<td>0.838</td>
<td>0.869</td>
<td>0.888</td>
<td>0.895</td>
<td>0.891</td>
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<td>0.909</td>
<td>0.891</td>
<td>0.861</td>
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<td>0.816</td>
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<td>0.903</td>
<td>0.926</td>
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<td>0.930</td>
<td>0.910</td>
<td>0.877</td>
<td>0.830</td>
<td>0.888</td>
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<td>16</td>
<td>0.824</td>
<td>0.877</td>
<td>0.917</td>
<td>0.941</td>
<td>0.951</td>
<td>0.945</td>
<td>0.924</td>
<td>0.889</td>
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<td>0.833</td>
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<td>0.924</td>
<td>0.947</td>
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<td>0.840</td>
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<td>0.959</td>
<td>0.967</td>
<td>0.959</td>
<td>0.935</td>
<td>0.895</td>
<td>0.840</td>
<td>0.914</td>
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</tr>
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<td>1</td>
<td>0.854</td>
<td>0.911</td>
<td>0.952</td>
<td>0.977</td>
<td>0.985</td>
<td>0.977</td>
<td>0.952</td>
<td>0.911</td>
<td>0.854</td>
<td>0.930</td>
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<tr>
<td>Average of Field</td>
<td>0.824</td>
<td>0.875</td>
<td>0.912</td>
<td>0.935</td>
<td>0.943</td>
<td>0.941</td>
<td>0.916</td>
<td>0.881</td>
<td>0.832</td>
<td>0.895</td>
<td></td>
</tr>
</tbody>
</table>
### Table II-5

Area Loss Performance for All-North Field Solar Collector Configuration, Summer Solstice Date (Day 172), at 33.6°N, 111.3°W (Horse Meas, Arizona) Location

<table>
<thead>
<tr>
<th>Heliostat Row</th>
<th>Average of Cosines of All Heliostats in Designated Row</th>
<th>Average of Cosines for Day</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>6 am</td>
<td>7 am</td>
</tr>
<tr>
<td>38</td>
<td>0.580</td>
<td>0.642</td>
</tr>
<tr>
<td>31</td>
<td>0.591</td>
<td>0.658</td>
</tr>
<tr>
<td>23</td>
<td>0.607</td>
<td>0.680</td>
</tr>
<tr>
<td>15</td>
<td>0.629</td>
<td>0.706</td>
</tr>
<tr>
<td>8</td>
<td>0.653</td>
<td>0.732</td>
</tr>
<tr>
<td>1</td>
<td>0.698</td>
<td>0.775</td>
</tr>
<tr>
<td>Average of Field</td>
<td>0.627</td>
<td>0.698</td>
</tr>
<tr>
<td>Heliostat Row</td>
<td>Average of Cosines of All Heliostate in Designated Row</td>
<td>Average of Cosines for Day</td>
</tr>
<tr>
<td>---------------</td>
<td>-------------------------------------------------------</td>
<td>---------------------------</td>
</tr>
<tr>
<td></td>
<td>9 am</td>
<td>10 am</td>
</tr>
<tr>
<td>38</td>
<td>0.917</td>
<td>0.942</td>
</tr>
<tr>
<td>31</td>
<td>0.921</td>
<td>0.948</td>
</tr>
<tr>
<td>23</td>
<td>0.922</td>
<td>0.952</td>
</tr>
<tr>
<td>16</td>
<td>0.919</td>
<td>0.951</td>
</tr>
<tr>
<td>8</td>
<td>0.905</td>
<td>0.940</td>
</tr>
<tr>
<td>1</td>
<td>0.893</td>
<td>0.931</td>
</tr>
<tr>
<td>Average of Field</td>
<td>0.913</td>
<td>0.944</td>
</tr>
</tbody>
</table>
2. **Specular Reflectivity**

The reflectivity parameter of concern to the designer of a concentrating power system is the specular reflectivity, rather than the reflectivity normally obtained with spectrophotometers. Additionally, the reflectivity needs to be referenced to a terrestrial solar spectrum in the geographical zone where the installation is to be located. In an effort to obtain comparative data on several candidate mirror surfaces with specularity collimation similar to that of the solar monitoring instrumentation, the reflectivity test rig shown in Fig. II-10 and II-11 was assembled. The reflected beam from the test mirror is monitored by the reverse-mounted pyrheliometer. To avoid shadowing, the mirror sample is tested with a 10-deg tilt ($\cos = 0.98481$). The reference insolation input to the mirror is monitored by the pair of independently tracked pyrheliometers shown in Fig. II-12.

![Specular Reflectivity Test Rig Using Normal Incidence Pyrheliometers, Profile View](image-url)
Fig. II-11
Specular Reflectivity Test Rig Using Normal Incidence Pyrheliometers,
Front View
Fig. II-12
Reference Pyrheliometers Used for Insolation Monitoring during Specular Reflectivity Tests
Results of specular reflectivity tests completed to date are shown in Table II-7. The design specular reflectivity used in establishing the system performance is 85%. From the above samples tested it would appear that this is attainable with laminated mirrors made of glass with improved transmissivity. The increase in transmission required is attainable by use of water white glass or a thinner mirror in the lamination by 1 mm.

Table II-7
Specular Reflectivity Measured by Normal Incidence Pyrheliometer Test Rig

<table>
<thead>
<tr>
<th>Description of Mirror Sample</th>
<th>Test Date</th>
<th>Specular Reflectivity, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>High</td>
</tr>
<tr>
<td>First Surface Aluminized</td>
<td>10-1-74</td>
<td>86.5</td>
</tr>
<tr>
<td>Teflon (Reference Surface)</td>
<td>10-8-74</td>
<td>86.6</td>
</tr>
<tr>
<td>Second Surface Silvered</td>
<td>9-20-74</td>
<td>83.1</td>
</tr>
<tr>
<td>Commercial Float Glass</td>
<td>9-23-74</td>
<td>83.9</td>
</tr>
<tr>
<td>Laminated Mirror, 3 mm</td>
<td>9-23-74</td>
<td>84.1</td>
</tr>
<tr>
<td>Front to Mirrored Surface</td>
<td>10-8-74</td>
<td>79.3</td>
</tr>
<tr>
<td>Second Surface Silver on</td>
<td>9-23-74</td>
<td>79.3</td>
</tr>
<tr>
<td>Acrylic, 3.2 mm Thick</td>
<td></td>
<td>79.3</td>
</tr>
<tr>
<td>Second Surface Silvered</td>
<td>9-20-74</td>
<td>72.2</td>
</tr>
<tr>
<td>Commercial Float Glass, 6 mm Thick</td>
<td>10-8-74</td>
<td>73.3</td>
</tr>
</tbody>
</table>

3. Optical Performance

To develop detailed optical performance data for the solar collector field of the various configurations being traded off during the preliminary design, we used the Martin Marietta PAGOS (Analysis of General Optical Systems) computer program.

The optical patterns formed in the focal plane, by each of 54 parabolically curved heliostats distributed throughout the field of 2002 possible locations, were obtained and form the basis for the cosine and optical accuracy performance predictions. The data were obtained for equinox, summer solstice, and winter solstice solar conditions throughout the day at 1-hour intervals for a plant location at Horse Mesa, Arizona (33.6°N, 111.3°W). The location is geometrically typical of the bulk of the southwestern U.S. Sun Bowl and is illustrated on the solar weather map (Fig. II-13).
Fig. II-13
National Climatic Center Annual Sunshine Hours Map with "Sun Bowl" Accented
(Hours in Hundreds)
The clusters of 396 central rays and 396 right-side edge rays for 8 am, March 21 (row 38/position 28, left), are shown in Fig. II-14 against a background of the 20x20-ft aperture. The solar discs associated with the center and edge zones (right, left, top, and bottom) are also shown. The program traces the ray pattern from each of 396 zones on a single optical surface, in this case a 20x20-ft square heliostat. The five solar discs shown are similar to those from a masked mirror (Fig. II-15). Figure II-16 illustrates the pattern from the masked, only partially warped mirror, while Fig. II-17 shows the same discs after final adjustment. The reference disc on the target is the theoretical size of the sun's image for the specific adjusted focal distance. These pictures are from another program, but are included to illustrate the effects being studied with the computer program.

The image pattern variation with time of day caused by the spherical aberration is illustrated by the ray clusters and associated images of Fig. II-18 thru II-21 for 10 a.m., 12 noon, 2 p.m. and 4 p.m. for the heliostat position in row 38/lef t 28.

At 8 a.m. all of the solar discs are well inside the aperture and the fringe band is relatively narrow. This general condition is maintained and even slightly improved at 10 a.m. (Fig. II-18). The noon focal pattern (Fig. II-19) is again nearly a match for 8 a.m. Significant aberration begins to set in in the afternoon, with the 2 p.m. pattern (Fig. II-20) nearly reaching the edge of the aperture, and with the 4 p.m. pattern (Fig. II-21) impinging on the secondary concentrator.

Optical image patterns have been the controlling influence in sizing the cavity apertures of all boiler, superheater, and combined boiler/superheater cavities. Overall, the mirror curvature compensates for the divergent angle of the incoming sunlight rays, enabling use of an aperture the same size as the mirror for fields as large as 1400x1400 ft. An example of this influence is the body of optical data led to the major design modification, which moved the tower location from the center of the field to the south border. Table II-8 shows a set of cosine and geometric efficiencies for corresponding heliostat locations in the north and south fields for the equinox, and the resulting heliostat efficiency, which is the product of the two. The north field heliostats were more effective, optically, than the south field heliostats in all cases by factors ranging between 1.51 and 2.01. It was evident that the area loss and optical efficiency performance allocations of Fig. II-9 would be compromised if the configuration were not changed.
Fig. II-14 Heliostat R38-28L, 6 am, March 21, Cosine = 0.91211 (24.20°)
Fig. II-15
4x4-Foot Mirror, Masked Except for Center and Corners to Enable Optical Adjustment
Fig. II-16
Five Solar Discs from Partially Adjusted 4x4-Foot Mirror
Fig. II-17
Five Solar Discs Overlapped after Final Optical Adjustment
Fig. II-18  Heliostat R28-28L, 10 am, March 21, Cosine = 0.92662 (22.14°14)
Fig. 11-13 Heliostat R38-28L, 12 Noon, March 21, Cosine = 0.8860 (27.3°)
Fig. II-20  Heliostat H39-B8L, 2 pm, March 31, Cosine = 0.8018 (36.66°)
Fig. II-21 Heliostat H38-28L, 4 pm, March 21, Cosine = 0.07637 (47.3°)
Table II-8
Comparison of North- and South-Field Heliostat Efficiencies for Central Tower Collector Field Average Daily Values for Equinox for the 33.6°N, 111.3°W Location (Horse Mesa, Arizona)

<table>
<thead>
<tr>
<th>Heliostat Position</th>
<th>North Field Heliostat</th>
<th>South Field Heliostat</th>
<th>Relative Performance for Corresponding Heliostats, North/South</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>%Area (Cosine)</td>
<td>%Geom (Aberration)</td>
<td>%Area (Cosine)</td>
</tr>
<tr>
<td>Row 12B, Position 1, Left</td>
<td>0.850</td>
<td>1.00</td>
<td>0.850</td>
</tr>
<tr>
<td>Row 12B, Position 12, Left</td>
<td>0.837</td>
<td>1.00</td>
<td>0.837</td>
</tr>
<tr>
<td>Row 12B, Position 24, Left</td>
<td>0.806</td>
<td>0.964</td>
<td>0.785</td>
</tr>
<tr>
<td>Row 3NH, Position 1, Left</td>
<td>0.893</td>
<td>1.00</td>
<td>0.893</td>
</tr>
<tr>
<td>Row 3NH, Position 10, Left</td>
<td>0.842</td>
<td>0.950</td>
<td>0.798</td>
</tr>
</tbody>
</table>

4. Optical Accuracy Attainable with Warping Glass Mirrors

The optical pattern from a warped glass mirror exceeds the size of the solar image that would be obtained by "on axis tracking" due to the combined effects of spherical aberration and manufacturing or assembly tolerances. While correlation of the patterns available from various mounting arrangements is beyond the scope of this contract, an indication of the magnitude of the two effects is obtainable from the feasibility sample mirror (19x19-in.²). Referring to Fig. 11-22, a computer-derived aberration pattern for a centrally located heliostat in the early design configuration, it will be noted that the sun centers occupy an image grouping very close to that of the complete 32-minute sun image. This would result in doubling the diameter of the resulting image due to aberration alone.

The approximate doubling of the image due to the spherical aberration is also evident in the small sample mirror built during the proposal activity. Figure II-23 shows the image of the sample mirror at a focal distance of 65 ft, referenced against a solar image for the same distance. (Grids on the target are spaced 2 in. apart.) An optical-error analysis of this image shows that the error is equivalent to an overall error of 11.7 minutes, with a probable distribution of 8 minutes for spherical aberration and 3.7 minutes for tangential error of the surface.

5. Optical Accuracy of Heliostat Assembled with Warping Glass Mirrors

The optical accuracy values in Table II-3 are based on the analyses of the computer derived patterns from the Martin Marietta PAGOS (Analysis of General Optical Systems) as previously described. Empirical correlation data were obtained from an independent heliostat optical accuracy development program undertaken with Martin Marietta corporate research support to determine the attainable performance from the warped glass mirrors currently envisioned.
Fig. II-22
Optical Ray Tracing Pattern Referenced for Warp Accuracy Correlation
Fig. II-23
Image of 19.5-in. Curved Heliostat Sample Mirror at a 65-ft Focal Distance
(2-in. Grid Spacing)
for the baseline design. These results and the correlation between the observed and the computer-derived patterns are presented here because they form the basis for confidence in this approach.

The seven-mirror concentrating heliostat used in the Martin Marietta corporate research program is shown in Fig. II-24. The highest accuracy curvature control structure of the candidate methods evaluated during the program was the "wheel" shown in Fig. II-25, which resulted in a concentration of 5.32 : 1, as shown by the optical image on the test target of Fig. II-26.

The spherical aberration correlation testing was performed with the seven mirrors of the concentrating heliostat masked as shown in Fig. II-27 so that the reflected rays to the target were in effect central ray traces such as the computer model generates in its initial stage. The primary difference between the computer model and the test model is that the computer model is based on a perfect parabolic curvature, which in the off-axis tracking concentrating heliostat application never reaches a condition of zero error for the constellation of the test, while the test heliostat was zeroed out for noon Mountain Daylight Time (11 a.m. true solar time). This enabled some of the patterns in the central portion of the day to be tighter than the "theoretical" pattern, while those at the ends of the day were in close agreement with the computed pattern. Figures II-28 thru II-35 show the optical target photographs and the overlayed computer patterns for a full day of testing, May 29, 1974.

6. **Thermal Energy Retention Losses**

The discussion of the reflected radiation, IR radiation, convection, and insulation losses is included in Chapter VI, which covers the results of the steam generator design analyses.
Fig. II-24
Optical Accuracy Development Concentrating Heliostat; Seven 4x4-ft Second Surface Glass Mirrors Warped for 108-ft Focal Distance (Martin Marietta Corporate Research Program)
Fig. II-26
Wheel Curvature Control Structure Installed on 4x4-ft Second Surface Glass Mirror, 108-ft Focal Length (Martin Marietta Corporate Research Program)
Fig. II-26
Optical Image from Wheel Structure Controlled 4x4-ft Mirror, Concentration = 5.32 : 1
(Martin Marietta Corporate Research Program)
Fig. 11-27
Seven Mirror Concentrating Heliostat Masked for Center Ray Reflection Optical Pattern
Testing to Study Effective Spherical Aberration (Martin Marietta Corporate Research
Program)
Fig. II-28
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 9 a.m. MDT (8 a.m. TST) May 29, 1974
(a) Optical Target Photograph

(b) Overlay

Fig. II-29
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 10 a.m. MDT (9 a.m. TST) May 29, 1974
(a) Optical Target Photograph

(b) Overlay

Fig. II-30
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 11 a.m. MDT (10 a.m. TST) May 29, 1974
Fig. II-31
Overlay of Test Observation Images on Computer Pattern,
Noon MDT (11 a.m. TST) May 29, 1974

II-46
Fig. II-32
Optical Test Photograph and Overlay of Test Observation Images on Computer Pattern, 1 p.m. MDT (Noon TST) May 29, 1974
Fig. II-33
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 2 p.m. MDT (1 p.m. TST) May 29, 1974
(a) Optical Target Photograph

(b) Overlay

Fig. II-34
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern,
3 p.m. MDT (2 p.m. TST) May 29, 1974
(a) Optical Target Photograph

(b) Overlay

Fig. II-35
Optical Target Photograph and Overlay of Test Observation Images on Computer Pattern, 4 p.m. MDT (3 p.m. TST) May 29, 1974


III. User Application Analyses—A Direct Solar Augmentation Based on Salt River Project Operations
III. USER APPLICATION ANALYSES--A DIRECT SOLAR AUGMENTATION BASED ON SALT RIVER PROJECT OPERATIONS

The first user application analysis, aimed at evaluating the use of a solar-conversion power plant to directly augment an existing hydroelectric system, is, in a broader sense, an evaluation of the load-following application for a utility having both thermal and hydroelectric generating resources. The Salt River Project of Phoenix, Arizona, is cooperating in this study and is the source of generating resource data, electrical load pattern data for both the system and individual generating units, water usage constraints, and onsite sunshine data. The framework of the study, worked out jointly with personnel from the Salt River Project, is to define the potential contribution of a 100-MWe solar-energy-conversion power plant over a one-year period, referenced against the actual load-generation and insolation patterns. The period selected for the study is July 1, 1973 to June 30, 1974.

A. BACKGROUND DISCUSSION OF SYSTEM COMPOSITION AND CONSTRAINTS

The generating equipment that supplies electrical power to the Salt River Project falls generally into the category of baseload steam-generating units (approximately 64% of the system), hydroelectric generating units, which are dominantly used for peak load-following (approximately 18% of the system), and combustion turbines capable of burning either natural gas or fuel oil, which are used for peak load-following (approximately 18% of the system). The operation of the hydroelectric units is closely controlled by the overriding constraint of water consumption in the Phoenix area.

The hydroelectric system consists of four dams on Salt River--Stewart Mountain, Mormon Flat, Horse Mesa, and Theodore Roosevelt, as shown in Fig. III-1. The flow through the Stewart Mountain Dam, the lowest dam in the system, is indicative of water orders, and ranges from a full flow of approximately 3200 acre-feet/day during the summer months to zero flow during the winter months. Mormon Flat and Horse Mesa have recently been equipped with large pumpback units of 50- and 97-MWe capacities, respectively, which can operate approximately 7.5 hr/day at the rated capacity of 3200-acre-feet for the summer months. During the winter months, these units must pump back a volume equivalent to that used for generation. Theodore Roosevelt Lake is the main storage reservoir and can supply the peak water-order volume in approximately 17.5 hr of operation.
Fig. III-1 Salt River Project Hydroelectric Installations
The data reduction to relate the available solar energy, and operating characteristics of an urban electric generating application in the southwestern United States was completed using data supplied by the Salt River Project of Phoenix, Arizona. Operating profiles for each day of every fourth week for the one year period from July 1, 1973 through June 30, 1974 are included in Appendix D. These show the system load for the Salt River Project in the Phoenix area, the operating periods for the hydroelectric generating units and the combustion turbine units, and the potential output of the solar plant based on the available sunshine. By taking every fourth week, it was believed that the sample results could be multiplied by four with confidence to obtain yearly operation. With regard to the solar operating hours for which the complete yearly data permit cross checking, this was largely substantiated. The total for the 13 weeks was 751.2 hrs, which when scaled up by a factor of 4 resulted in 3004.8 hr. The summation of the full year was 3086.1, a variation of 2.4%.

A sample profile, for July 2, 1973, is shown in Fig. III-2 to illustrate the elements of the SRP generating system and the potential solar plant that were correlated. The system load represents the power supplied to Salt River Project customers. This pattern follows a somewhat sine-wave pattern with the low point occurring near 6 a.m. and a peak near 6 p.m. The operation of the Horse Mesa No. 4 hydroelectric generator is plotted immediately below the system load curve. Note that the heavy generation from this unit occurred for periods of 7 to 8 hr, which included the system's peak periods. The remaining hydroelectric units are plotted in the band below the remaining net curve.

The operation of the combustion turbines is shown below the net curve remaining after "peak shaving" by the hydro units. For the sample day in July, these units carried a portion of the load for the full 24 hr, but this was not typical of the bulk of the days observed for the year.

The potential output from the solar plant, based on the weather existing on the sample day, is shown in the separate plot at the top of the figure and is overplotted on the main profile to illustrate the potential energy displacement. For the July day, the availability and magnitude of solar-plant power was such that the solar plant could have displaced the combustion turbines during the period of peak load buildup from 6 a.m. to 6:30 p.m. The equivalent oil savings of the 1248 MW-hr output from the solar plant on this day was 2714 barrels of fuel oil.
POTENTIAL SOLAR PLANT OUTPUT POWER
1246 MW-HRS, 2714 BBL OIL EQUIV.

SYSTEM LOAD

HORSE MESA HYDRO NO. 4
BALANCE OF HYDRO UNITS

COMBUSTION TURBINES

POTENTIAL SOLAR PLANT

PHOENIX TEMPERATURE RANGE -- 115/87 °F

MONDAY, JULY 2, 1973
1. Solar Data

Solar data taken from August 15, 1973 to August 14, 1974 was reviewed and compiled to establish the potential solar power plant operating hours attainable during the period. Results are shown on a daily basis in Table III-1. The overall potential operation of the solar plant was 3086.1 hr. The range of monthly operating periods was from a high in May of 327.9 hr to a low in November of 197.25 hr.

Solar data for the operating profiles shown for every fourth week in Appendix D, were established from direct intensity data for similar days of the year in 1962, with modifications for both cloud cover from the 1973 data and for fine grained day-to-day differences based on U.S. Weather Bureau charts of total intensity for the 1973 days of interest.

The question of cloud interruptions imposed on solar power system operation can be partially answered by the results of a review of the daily sunshine charts (Table III-2). A cloud interruption was established to be an interruption that would have forced the solar plant to drop load following a period of operation of at least 1 hr. For cloud interruptions more frequent than 1 hr, practical experience in solar facilities has shown that the clouds are in sight of the operator and the plant would not be restarted until sufficient clear sky conditions were established to get at least an hour's operation. Eighty such cloud interruptions were observed in the sunshine monitor data review. These, together with the 350 daily cycles, would indicate that the plant cycling would be 430/yr.

The relationship between solar plant operating hours and total sunshine hours recorded by the weather bureau is shown on a daily basis for the sample year in Fig. III-3. Roughly 2.5 hr/day the ends of the day are not included in the operating time totals although they would be used for warmup and shut-down operations. The operating days per month potential is illustrated in the upper set of curves of Fig. III-3. June, September, October, and December are fully used because of excellent solar weather, while January, March, and November are the poorest, each losing approximately a week to the stormy weather.
<table>
<thead>
<tr>
<th>Day of Month</th>
<th>August</th>
<th>September</th>
<th>October</th>
<th>November</th>
<th>December</th>
<th>January</th>
<th>February</th>
<th>March</th>
<th>April</th>
<th>May</th>
<th>June</th>
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<th>August</th>
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<td>10.50</td>
<td>9.83</td>
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<td>4.25</td>
<td>6.28</td>
<td>11.58</td>
<td>11.21*</td>
<td>11.00</td>
<td>7.50</td>
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</tr>
<tr>
<td>2</td>
<td>10.83</td>
<td>9.67</td>
<td>9.00</td>
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<td>5.25</td>
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<td>0.58</td>
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<td>11.75</td>
<td>10.50</td>
<td>10.00</td>
<td></td>
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<td>11.63</td>
<td>12.00</td>
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<td>9.50</td>
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<td>0.00</td>
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<td>9.67</td>
<td>9.16</td>
<td>7.50</td>
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<td>7.55*</td>
<td>8.25</td>
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<td>7.85</td>
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* Sunshine Monitor Data Missing - Operating Hours Based on Days Having Similar Temperature Profile in Same Month
Table III-2
Summary of Sunshine Monitor Data from Salt River Project, August 1973-1974

<table>
<thead>
<tr>
<th>Observation Period</th>
<th>Operating Days for Solar Plant</th>
<th>Operating Hours Above Threshold Solar Intensity</th>
<th>Cloud Interruptions Following Periods of 1 hr Minimum</th>
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<td>Aug 15-31, 1973</td>
<td>17</td>
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<td>31</td>
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<td>27</td>
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<td>Feb 1-28, 1974</td>
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<td>Apr 1-30, 1974</td>
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<td>Yearly Total</td>
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<td>3086.1</td>
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2. Solar Plant Operation

Potential solar power plant operating periods and output variation patterns within those patterns are shown in detail in the profiles of Appendix D. A summary of the power generated on each of the 93 sample days and its fuel oil equivalent (based on the performance of the most efficient of the combustion turbine installations of the Salt River Project) is shown in Table III-3. Weekly generation subtotals ranged from highs of 8362 and 8121 MW-hr for the June 2-8, 1974 and July 1-7, 1973 weeks to lows of 2893 and 5004 MW-hr for the November 18-24, 1973 and August 26-September 1, 1973 weeks. For prospective, the daily power generation and operating periods are plotted against a year long time scale in Fig. III-4. The days lost to weather within the sample weeks are clearly evident as are the relatively constant operating periods of May-June-July and September-October, the early summer and "indian summer" periods. Daily high and low temperatures for the full year are also included on Fig. III-4 for reference and correlation.

The sample weeks total power generation was 82,787 MW-hr, which scales up to 331,148 MW-hr for the year. The fuel equivalent, was 180,036 barrels of fuel oil for the 13-week sample period and 720,147 barrels of fuel oil for the year.

Results analyzed to date are "first look" in that they relate the potential solar plant to the operational pattern of the utility "as run." It is recognized that further analysis to examine the potential benefits resulting from modification of the hydroelectric dispatching pattern within constraints of water consumption order requirements and using periods of solar powered pump back operation is needed to round out the application analysis. This expanded analysis should be carried out in the subsequent phase of the program.
Fig. III-3
Solar Plant Operating Time, Total Sunshine Hours and Monthly Operating Days for a Solar Plant for a 1-yr Period in the Phoenix, Arizona, Area
<table>
<thead>
<tr>
<th>Day</th>
<th>Potential Solar Plant Output</th>
<th>Operation, hr</th>
<th>Equivalent Barrels of Oil</th>
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<td>Week 1--July 1-7, 1973</td>
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<td>1.251</td>
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<td>July 2</td>
<td>1,248</td>
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<tr>
<td>July 3</td>
<td>1,248</td>
<td>1.274</td>
<td>11.55</td>
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<tr>
<td>July 4</td>
<td>1,248</td>
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<td>July 6</td>
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<td>July 7</td>
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<td>8,160</td>
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<td>757</td>
<td>0.782</td>
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<tr>
<td>Aug 3</td>
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<td>Aug 4</td>
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Fig. III-4
Summary of Sample Weeks Power Generation and Operating Periods Against Year-Long Time Scale Referenced Against Daily Temperature Pattern.
IV. User Application Analyses--Intertie Augmentation Based on Bonneville Power Administration Model
IV. USER APPLICATION ANALYSIS - INTERTIE AUGMENTATION OF HYDROELECTRIC POWER SYSTEM BASED ON BONNEVILLE POWER ADMINISTRATION PROJECTED CONFIGURATION AND REQUIREMENTS

A. STUDY OF CANDIDATE SOLAR PLANT TECHNICAL COMPATIBILITY

During meetings held at Bonneville Power Administration Headquarters (May 7 and 8, 1974), data were supplied on the generating resources, the ac and dc interties used for energy exchange between the Pacific Northwest and Pacific Southwest, and the projected operating schedule patterns used for planning. Location of the resources for the entire region, including those of the federal system, public agencies, and private utilities are shown on the map of Fig. IV-1. Bonneville Power Administration dispatches the power generation and transmission of the dominantly hydroelectric federal system. The major power generation resources of this system are on the Columbia and Snake River systems. Table IV-1 lists the major installations and their nameplate ratings as of December 31, 1973. Typical features of these installations are illustrated in Fig. IV-2 (John Day navigation lock, spillway, and power plant), IV-3 (Generator Room of The Dalles Power Plant), and IV-4 (New Construction, Dual Power Plants, and Spillway of the Grand Coulee Installation).

Table IV-1
Major Hydroelectric Installations Associated with Bonneville Power Administration in U.S. Pacific Northwest

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<th>Columbia River Installations</th>
<th>Nameplate Rating, MWe December 31, 1973</th>
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</thead>
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<td>518.4</td>
</tr>
<tr>
<td>The Dalles Dam</td>
<td>1,807.0</td>
</tr>
<tr>
<td>John Day Dam</td>
<td>2,160.0</td>
</tr>
<tr>
<td>McNary Dam</td>
<td>980.0</td>
</tr>
<tr>
<td>Chief Joseph Dam</td>
<td>1,024.0</td>
</tr>
<tr>
<td>Grand Coulee Dam</td>
<td>2,195.0</td>
</tr>
<tr>
<td>Hungry Horse Dam (Flathead River feeding Columbia)</td>
<td>285.0</td>
</tr>
<tr>
<td>Subtotal, Major Columbia Hydroelectrics</td>
<td>8,969.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Snake River Installations</th>
<th>Nameplate Rating, MWe December 31, 1973</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ice Harbor Dam</td>
<td>270.0</td>
</tr>
<tr>
<td>Lower Monumental Dam</td>
<td>405.0</td>
</tr>
<tr>
<td>Little Goose Dam</td>
<td>405.0</td>
</tr>
<tr>
<td>Subtotal, Major Snake Hydroelectrics</td>
<td></td>
</tr>
<tr>
<td>Combined Major Units Subtotal</td>
<td>10,049.4</td>
</tr>
</tbody>
</table>
Fig. IV-1 Electric Power Plant Locations in Pacific Northwest
(Courtesy of Bonneville Power Administration)

Fig. IV-2 John Day Navigation Lock, Spillway, and Power Plant
(Courtesy Bonneville Power Administration)

Fig. IV-3 Interior View of First Eight Generator Units at The Dalles Power Plant
Fig. IV-4
Grand Coulee Dam and Power Plants (L to R, Construction for Six New Units of Third Power Plant, Left Power Plant, Spillway, and Right Power Plant)
The consensus of the coordination was that the initial study should quantify the degree of compatibility of a solar power plant with the hydroelectric generating complex of the Pacific Northwest. The potential problem with technical compatibility is the possibility that river flow constraints modified to accommodate the influx of solar plant power could cause water to be spilled or loss of head at individual installations. Such effects would largely negate the benefits of the solar energy generated power. Guidelines for the study to establish technical compatibility were:

1) Solar plant size to be 1000 MWe;

2) Solar power schedule on a monthly basis derived from 1973-1974 Salt River Project sunshine data;

3) Basis for assessment would be results of Bonneville Power Administration's Resources Management Planning Model, using 1982 data on system configuration and projected load with river flow in accordance with the 42.5 month minimum model.

The effects on the intertie exchange and daily power generation schedules would be evaluated separately after the initial evaluation of results from the Resources Management Planning Model.

Table IV-2 shows the solar plant operation used as input to the Resources Management Planning Model. Two runs of the model are required to establish the overall energy credit from a potential power resource, a baseline run with the system configuration as currently planned and a second run with the candidate plant's contribution included. The 1000-MWe candidate solar plant's contribution efficiency is established by the ratio of the "system benefit" to the "average power generation." The system benefit shown by the resources model was 350.8 MWe, while the average power of the solar plant was 352.2 MWe, indicating that 99.6% of the solar generated power contributed to the system output without adversely affecting hydroelectric management. Additional firm peak power capacity of approximately 468 MWe (350.8/0.75) would be needed to attain full capacity credit for the solar plant addition to the system. This could be from low usage combustion turbines or additional hydroelectric generators added to existing installations.
Table IV-2
Monthly Schedule of Solar Power Plant Operation for Use in Bonneville Power Administration Resources Management Planning Model

<table>
<thead>
<tr>
<th>Month</th>
<th>Average, hr/day</th>
<th>Total for Month, hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>July</td>
<td>9.75</td>
<td>302.25</td>
</tr>
<tr>
<td>August</td>
<td>9.215</td>
<td>285.67</td>
</tr>
<tr>
<td>September</td>
<td>10.032</td>
<td>300.96</td>
</tr>
<tr>
<td>October</td>
<td>9.114</td>
<td>282.53</td>
</tr>
<tr>
<td>November</td>
<td>6.720</td>
<td>201.61</td>
</tr>
<tr>
<td>December</td>
<td>7.159</td>
<td>221.92</td>
</tr>
<tr>
<td>January</td>
<td>6.452</td>
<td>200.02</td>
</tr>
<tr>
<td>February</td>
<td>7.168</td>
<td>200.70</td>
</tr>
<tr>
<td>March</td>
<td>6.877</td>
<td>213.19</td>
</tr>
<tr>
<td>April</td>
<td>9.018</td>
<td>270.54</td>
</tr>
<tr>
<td>May</td>
<td>10.577</td>
<td>327.88</td>
</tr>
<tr>
<td>June</td>
<td>10.461</td>
<td>313.84</td>
</tr>
</tbody>
</table>

B. PACIFIC NORTHWEST–PACIFIC SOUTHWEST INTERTIES

Energy exchange between the Pacific Northwest and the heavy population centers of the Pacific Southwest is accomplished by use of one dc and two ac interties, shown in the map of Fig. IV-5. Rated capacities are 1400 MWe for the 800 kV dc intertie and 1000 MWe each for the 500 kV ac interties. Important features of the 800 kV dc intertie are its length, 846 miles, and its transmission efficiency, 89% from the 230 kV ac input to the sending converter station to the 230 kV ac output of the receiving converter station. Only two converter stations provide access to the power on the dc intertie. The Celilo converter station, northern terminal of the 800 kV dc line near The Dalles Oregon, is shown in Fig. IV-6 and IV-7.

The end to end efficiency of the ac interties is 86% but intermediate access distribution results in a substantial fraction of the power being delivered at shorter distances with correspondingly higher efficiency.
Potential Siting Zones for Bonneville-Associated Solar Power Plants

Potential Siting Zones for 200X-MWe Solar Power Installation (Dual-Purpose Southern California Load-Following & Bonneville Augmentation)

Potential Site for 300-kWe Solar Pilot Plant

Potential Siting Zone for 30 to 236-MWe Solar Powerplants for Salt River Project Augmentation

(Courtesy of U.S. Bureau of Reclamation)

Fig. IV-5 Major Electrical Transmission Network of Western U.S.
(Courtesy of Bonneville Power Administration)

Fig. IV-6 Celilo Converter Station, Northern Terminal of 800 kV dc Line Near The Dalles, Oregon
Fig. IV-7
Inside View of Celilo Converter Station (Valve Hall, Housing RF Chokes, Mercury Arc Valves, and Insulating Transformers)
The detail analysis of the impact on intertie operation and on energy exchange planned for 1982 has been deferred. General impressions from the initial coordination indicated that solar operation with its relief of energy exchange loads in the daylight periods would be beneficial to the Pacific Northwest system. The exchange pattern is generally structured for power to flow south at approximately 1250 MWe during the period from 9 a.m. to 9 p.m. weekdays and be returned north at approximately 1550 MW from 11 p.m. to 7 a.m. weekdays and around the clock on weekends. Superimposed on this general pattern is a period of constant power being supplied to the southwest during periods of heavy runoff, which results in excess power capacity in the Northwest.
V. Economic Analysis
The usefulness of the 100 MWe solar conversion power plant depends on its ability to produce electrical energy at a cost competitive with existing conventional systems. This chapter presents initial estimates of construction and operating costs for 100 MWe pilot plant located in Central Arizona in close proximity to an appropriate high-voltage transmission line. Also included are initial efforts to compare those data with similar information pertaining to fossil fuel plants. Section A presents the latest data obtained for construction costs. Section B presents the initial effort to estimate operating costs for the solar conversion power plant. Section C presents the approach and the method of data manipulation used in compiling cost estimates.

A. CONSTRUCTION COSTS

The latest estimate of construction cost data is presented in Table V-1. For ease of comparison, those solar plant costs in common with, and those not in common with, fossil fuel plant costs, are given separate columns. All costs listed are current (1974) costs. When comparing the $894/kw solar plant cost with current new coal and nuclear power plant capital costs that range from $400 to $750 per kw, it should be kept in mind that the solar value is for the initial point on the production-construction learning curve. If a 90% learning curve were to be achieved as solar plants go into production, this cost would drop to $724/kw for the fourth plant and to $528/kw for the thirty-second plant. It should also be kept in mind that the estimates are for preliminary designs that should yield to cost reduction engineering during the development program.

B. OPERATING COSTS

Table V-2 presents current estimates of operating costs for the solar energy system with comparable data for the fossil fuel plants. All entries have been applied against an appropriate annual cost index so they reflect current (1974) costs. Detailed basis of costs is given item-by-item in Section D.
Table V-1 100-MW Solar Energy System Capital Costs (1974 Prices)

<table>
<thead>
<tr>
<th>Item</th>
<th>Costs Common to Fossil Fuel Plant&lt;sup&gt;a&lt;/sup&gt;</th>
<th>Costs Not Common to Fossil Fuel Plants&lt;sup&gt;a&lt;/sup&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Land and Land Rights</td>
<td>$271,000</td>
<td></td>
</tr>
<tr>
<td>General Cut and Fill</td>
<td>252,000</td>
<td></td>
</tr>
<tr>
<td>Roads, Walks, and Parking</td>
<td>269,000</td>
<td></td>
</tr>
<tr>
<td>Retaining Walls</td>
<td>222,000</td>
<td></td>
</tr>
<tr>
<td>Fencing &amp; Gates</td>
<td>140,000</td>
<td></td>
</tr>
<tr>
<td>Gatehouse</td>
<td>17,000</td>
<td></td>
</tr>
<tr>
<td>Sanitary Sewage Facility</td>
<td>95,000</td>
<td></td>
</tr>
<tr>
<td>Yard Drainage</td>
<td>137,000</td>
<td></td>
</tr>
<tr>
<td>Yard Lighting</td>
<td>136,000</td>
<td></td>
</tr>
<tr>
<td>Main Power Station</td>
<td>1,415,000&lt;sup&gt;b&lt;/sup&gt;</td>
<td>$8,000,000</td>
</tr>
<tr>
<td>Boiler Towers</td>
<td></td>
<td>2,307,000&lt;sup&gt;c&lt;/sup&gt;</td>
</tr>
<tr>
<td>Steam and Feed-Water Piping</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Switchgear, Transformers, &amp; Switchboard</td>
<td>411,000</td>
<td></td>
</tr>
<tr>
<td>Turbine Instrumentation</td>
<td>131,000</td>
<td></td>
</tr>
<tr>
<td>Wiring, Wiring Structures, &amp; Containers</td>
<td>714,000</td>
<td></td>
</tr>
<tr>
<td>Water Treatment</td>
<td>142,000</td>
<td></td>
</tr>
<tr>
<td>Professional Services, Construction Supervision, Temporary Facilities</td>
<td>3,172,000</td>
<td></td>
</tr>
<tr>
<td>Turbine-Generator Foundation &amp; Support, Lubrication System, &amp; Gas System</td>
<td>138,000</td>
<td></td>
</tr>
<tr>
<td>Boiler Monitor &amp; Control Instrumentation</td>
<td></td>
<td>1,024,000&lt;sup&gt;c&lt;/sup&gt;</td>
</tr>
<tr>
<td>Turbine-Generator</td>
<td>7,386,000</td>
<td></td>
</tr>
<tr>
<td>Condensate &amp; Feed-Water Circulation System</td>
<td>406,000</td>
<td></td>
</tr>
<tr>
<td>Air Cooled Condenser</td>
<td></td>
<td>5,400,000</td>
</tr>
<tr>
<td>Boilers &amp; Superheaters</td>
<td></td>
<td>9,250,000&lt;sup&gt;c&lt;/sup&gt;</td>
</tr>
<tr>
<td>Mirrors</td>
<td></td>
<td>5,979,000</td>
</tr>
<tr>
<td>Heliostat Fabrication &amp; Installation</td>
<td></td>
<td>35,475,000</td>
</tr>
<tr>
<td>Tracker Control &amp; Drive Equipment</td>
<td></td>
<td>6,491,000</td>
</tr>
<tr>
<td>Transmission Lines</td>
<td>55,000</td>
<td></td>
</tr>
<tr>
<td>Subtotal</td>
<td>$15,509,000</td>
<td>$73,926,000</td>
</tr>
<tr>
<td>Total Cost, 100 MWe Solar Energy Conversion</td>
<td></td>
<td>$89,435,000 ($894.35/kw)</td>
</tr>
</tbody>
</table>

Note: <sup>a</sup> See Section E for detailed basis of cost items.  
<sup>b</sup> Reduced from normal station because of deletion of fuel handling equipment and space.  
<sup>c</sup> Itemized as not common to fossil fuel plants because of extensive modification for use in solar energy system.
Table V-2
Annual Operating Costs for 100-MW Solar Energy and Fossil Fuel Plants (1974 Prices)

<table>
<thead>
<tr>
<th>Item</th>
<th>Solar Energy System Cost</th>
<th>Fossil Fuel System Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operations Supervision &amp; Engineering</td>
<td>$20,000</td>
<td>$20,000</td>
</tr>
<tr>
<td>Steam Expenses</td>
<td>4,000</td>
<td>82,000</td>
</tr>
<tr>
<td>Electric Expenses</td>
<td>50,000</td>
<td>50,000</td>
</tr>
<tr>
<td>Miscellaneous Steam Power Expense</td>
<td>4,000</td>
<td>27,000</td>
</tr>
<tr>
<td>Maintenance Supervision &amp; Engineering</td>
<td>14,000</td>
<td>14,000</td>
</tr>
<tr>
<td>Maintenance of Structures</td>
<td>17,000</td>
<td>7,000</td>
</tr>
<tr>
<td>Maintenance of Boiler Plant</td>
<td>105,000</td>
<td>53,000</td>
</tr>
<tr>
<td>Maintenance of Electric Plant</td>
<td>30,000</td>
<td>30,000</td>
</tr>
<tr>
<td>Miscellaneous Maintenance</td>
<td>9,000</td>
<td>4,000</td>
</tr>
<tr>
<td>Transmission Line Expense</td>
<td>1,000</td>
<td>1,000</td>
</tr>
<tr>
<td><strong>Subtotal</strong></td>
<td><strong>$254,000</strong></td>
<td><strong>$288,000</strong></td>
</tr>
<tr>
<td><strong>Fuel</strong></td>
<td>----</td>
<td><strong>$1,701,000</strong></td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$254,000</strong></td>
<td><strong>$1,989,000</strong></td>
</tr>
</tbody>
</table>

**BASIS FOR COST ESTIMATES**

The primary sources for the economic data presented are Ref V-1, V-2, and V-3. Reference V-1 from the Atomic Energy Commission contains a detailed breakdown of construction costs for a hypothetical 1000-MW coal-fired fossil fuel plant. Reference V-2 by the Federal Power Commission contains gross breakdowns of the construction and operating costs of 579 fossil fuel plants with a wide spectrum of power ratings. Reference V-3 from the Department of Interior, Bureau of Reclamation provides a gross breakdown of the construction and operating cost estimates for planning purposes for hydroelectric plants.

Because of the detailed breakdown of construction cost data and commonality of many components of our proposed solar energy system and the fossil fuel system, Ref V-1 was used as a basis to estimate construction cost of the solar conversion system components common with the fossil fuel plant. However, the cost figures from Ref V-1 could not be applied directly because there were differences, primarily, in the power rating and the quantity of land used. When estimating cost data for a given system from similar data for a reference cost system with a different capacity, it is common practice to use the following general equation:
\[
\text{New Cost} = \left(\frac{\text{New Capacity}}{\text{Reference Capacity}}\right)^X
\]

where X can be shown to be the slope of a line drawn between the two data points plotted on log-log paper with capacity as the ordinate and cost as the abscissa. In determining this value of X for the fossil fuel plant, total construction cost data from Ref V-2 was plotted as a function of power rating on log-log paper. The results are shown in Fig. V-1. The resulting slope of the midline of all points is 1.0, indicating that a direct ratio of power plant ratings can be taken in determining total construction costs. Applying the derived exponent to each of the applicable construction cost items as given in Ref V-1 may result in some deviation in their individual costs, but the total system cost is considered to be an acceptable estimate.

Verification of the hypothetical data contained in Ref V-1 was performed by comparing its total construction cost, scaled down from its reference 1000-MW rating to our 100-MW design rating, with those of actual plants listed in Ref V-2. The total estimated construction cost for the 1000-MW plant given in Ref V-1 is $174,115,200. Applying our cost adjustment equation, based on the plant capacity ratio of \((100/1000)^1\) Megawatts, this figure reduces to $17,411,520 for a 100-MW plant. This falls well within the bounds of existing plant data as indicated by the dashed lines in Fig. V-1.

Construction cost data were then obtained from Ref V-1 and adjusted to be representative of the 100-MW plant capacity rating. Where commonality existed between the fossil fuel plant and the solar energy plant, the costs, adjusted for power rating, were used directly. Where differences existed, modification of the capacity adjusted fossil fuel plant costs were made. Section E gives a detailed explanation of all plant component construction costs.

Operating costs were determined from Ref V-2 by determining the average operating costs for numerous plants after first applying the appropriately exponentiated power ratio to obtain data for a 100-MW plant as before. The exponents used for operating costs were not unity and varied for differing items. Because Ref 2 had only gross estimates by category of cost, it was necessary to determine to which category each cost item should be assigned and then determine the percentage of each category allocated to each cost item. It was also necessary to independently estimate the operating costs for some items that were peculiar to the solar energy conversion system. Section E details the rationale and estimate basis for all of the final costs.
Note: These data have been corrected to reflect current costs. The data presented are applicable to approximately the 1968 time period.

Fig. V-1
Federal Power Commission Steam Electric Plant (Fossil Fuel) Construction Costs
Current (1974) estimates of both the construction and operating costs were obtained from the Ref V-1 through V-3 costs that were actually incurred in different years ranging from 1968 to 1972. Because of known trends in technological development, which affected base costs of some items and documented inflation rates, it was necessary to adjust the costs for annual price indices. These were obtained from engineering periodicals and general economic information references and reflected actual historical price trends, not speculative indices for future prices. Unless otherwise noted, all data presented has been indexed to reflect January 1974 prices.

D. CONSTRUCTION AND OPERATION COST ITEMIZATION

This section contains an itemization and detailed description of the source, data, and rationale pertaining to each cost item of the construction and operations costs involved in the solar energy conversion system. Included in these costs are all required materials and labor.

1. Construction Costs

The items discussed in this section will be presented in the same order as found in Table V-1. Where reduction or increase in reference cost is necessary to make the solar plant facility comparable to a reference facility, the method indicated in Section C is used.
a. Land and Land Rights - Data were obtained from the May 10, 1974 issue of the Arizona Republic, the Phoenix newspaper, on the current asking prices for large acreages (primarily farm and ranch land). The cost of these land parcels will yield a conservative cost per acre for the solar energy system as these land sites are fertile areas, some with irrigation. The land required for our system need not have these features. A total of eight large-acreage (117 to 1120 acres) sites were taken at random and the average cost per acre was determined to be $669 per acre. As reference points, the highest and lowest costs were $2750 and $90 per acre, respectively. The proposed solar energy system requires 405 acres; therefore, the total cost of land would be approximately $271,000 at current prices.

b. General Cut and Fill - Although the solar energy plant will require extensive special contouring for the heliostat fields, it is planned to choose a site that provides a maximum of natural desired contours thereby minimizing the amount of special excavating required. For this reason it was estimated that there will be approximately the same amount of cut and fill for the solar energy plant as for a fossil fuel plant of comparable land area.

Reference V-1 indicates that for a 500-acre fossil fuel plant, the cost for this work will be $280,000. Multiplying this by the ratio of land areas, 405:500, the cost of the 405-acre plant would be approximately $227,000. Because these data reflect the cost at the time of publication of Reference V-1 (1972), a cost index must be applied to obtain 1974 costs. Reference V-4 indicates that the construction cost index changed from 143 to 159 from 1972 to 1974. This indicates that the January 1974 excavation costs for the solar energy plant are $227,000 x 159/143 = $252,000.

c. Roads, Walks, and Parking Areas - Reference V-1 indicates that for a 1000-MW, 500-acre plant, $242,000 is required for construction in this category. The solar energy plant being smaller, requires fewer walks and parking areas, but, because of the need for access to each row of heliostats, requires more roads. It is estimated, therefore, that this $242,000 figure will be representative of the cost of roads, walks, and parking areas for the solar energy plant. Applying the construction cost index as indicated previously, the January 1974 cost is $269,000.

d. Retaining Walls - The purpose of these retaining walls was not detailed in Ref V-1, but we conservatively estimated that the cost indicated for the 1000-MW, 500-acre plant would be the same for the solar plant, i.e., $200,000. Applying the construction cost index yields a January 1974 cost of $222,000.

e. Fencing and Gates - The cost per foot for fencing, including gates, was indicated in Ref V-1 to be $7.50 per foot. Recognizing the need to fence the entire perimeter of the installation, a
total of 16,800 ft of fencing is required. This dictates a capital expenditure of $126,000. After applying the construction cost index, the January 1974 cost for fencing and gates is $140,000.

f. Gatehouse - The cost of constructing a gatehouse is indicated in Ref V-1 to be $15,000. There will be no modification of this cost based on differing plant size. Application of the construction cost index to this figure yields a total cost of $17,000.

g. Sanitary Sewage Facilities - This category, which includes the construction of sewage treatment facilities and connections from the plant to the treatment plant, does not readily lend itself to reduction in cost because of reduced plant size. It is therefore estimated that the cost indicated in Ref V-1, $85,000, is valid for the solar energy plant. Application of construction cost indices yields a final January 1974 cost of $95,000.

h. Yard Drainage - Reference V-1 indicates that for its 1000-MW, 500-acre facility, a capital expenditure of $140,000 is required to provide yard drainage. It is realized that the areas in which the solar energy plant will be constructed will be dry with relatively little rainfall. However, these areas are subject to cloud bursts, which in the contoured fields could cause serious damage. It is, therefore, estimated that the cost indicated in Ref V-1 be reduced only by the ratio of plant acreage. The cost for the solar energy plant would therefore be $140,000 x 405/500 = $123,000. After application of the construction cost index the January 1974 cost for this item is $137,000.

i. Yard Lighting - The capital cost for yard lighting is directly proportional to the size of yard to be lighted. Reference V-1 indicates lighting a 500-acre yard costs $150,000. The cost for lighting the 405-acre solar plant yard would therefore be $150,000 x 405/500 = $122,000. Applying the construction cost index provides a January 1974 cost estimate of $136,000.

j. Main Power Station - The cost for constructing a main power station for a 1000-MW fossil fuel plant is estimated to be $16,970,000 in Ref V-1. This cost includes constructing a boiler room, bunker bay, and heater bay. As these items are not needed in our proposed solar energy plant, an appropriate reduction in cost is required, in addition to the reduction due to differing plant capacity (power output). First, application of the plant power ratio to reduce the cost from a 1000-MW to a 100-MW plant results in a main power station cost of $16,970,000 x (100/1000) = $1,697,000. Estimating the reduction in construction cost due to deletion of the previously discussed unneeded items at 25%, the final main power station construction cost for the solar energy system would be $1,697,000 x 0.75 = $1,273,000. Application of the construction cost index yields a January 1974 cost of $1,415,000.
k. Boiler Towers - A rough order of magnitude quotation was obtained from the M. W. Kellogg Co. for the construction of these towers. A rough estimate of $850,000 per tower without foundation, or approximately $1,000,000 per tower including foundation was provided. This results in a total tower cost (8 towers) of 8 x $1,000,000 = $8,000,000. These are current costs. The M. W. Kellogg Co. indicated that when more detailed requirements and design data were obtained a more accurate price estimate could be made.

l. Steam and Feed-Water Piping - There are many possible routes and manifolding schemes for this pipe, each having a different total cost. Forty different routings were analyzed. The least costly routing and manifolding scheme was selected while maintaining the system requirements in pressure drop, etc. After arriving at a reasonable configuration, costs for materials and labor were obtained from Ref V-5. The cost breakdown for this system is as follows:

<table>
<thead>
<tr>
<th>Item</th>
<th>Steam System</th>
<th>Feed-Water System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piping</td>
<td>$1,262,000</td>
<td>$314,000</td>
</tr>
<tr>
<td>Welding</td>
<td>108,000</td>
<td>23,000</td>
</tr>
<tr>
<td>Insulation</td>
<td>190,000</td>
<td>77,000</td>
</tr>
<tr>
<td>Valves</td>
<td>70,000</td>
<td>27,000</td>
</tr>
<tr>
<td>Total</td>
<td>$1,630,000</td>
<td>$441,000</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Separable Items</td>
<td>$2,071,000</td>
</tr>
<tr>
<td>Saddles and Rollers</td>
<td>33,000</td>
</tr>
<tr>
<td>Pipe Support Structures</td>
<td>210,000</td>
</tr>
<tr>
<td>Total Piping System</td>
<td>$2,314,000</td>
</tr>
</tbody>
</table>

Reference V-5 has a publication date of 1973. To convert this cost to 1974 prices, a cost index consisting of a mean of the materials cost indices and the skilled labor indices was derived from Ref V-4. This resulted in an index of 167.5 for 1973 and 167.0 for 1974. The January 1974 cost of the piping system is then $2,314,000 x 167.0/167.5 = $2,307,000.

m. Switchgear, Transformers, and Switchboard - Data from Ref V-1 indicates that the capital cost for these items for the 1000-MW plant is $3,744,800. Reducing this cost to 100-MW for the solar energy plant produces a capital cost of $374,000. The switchyard and substation cost index from Ref V-4 for 1972 and 1974 are 1.33 and 1.45, respectively. Application of these indices yields a January 1974 cost of $374,000 x 1.46/1.33 = $411,000.

n. Turbine Instrumentation - A capital cost of $1,165,000 is indicated in Ref V-1 for this item in a 1000-MW plant. Reducing this cost for the 100-MW plant yields a solar energy system cost of $117,000. Reference V-4 indicates that the cost indices for
accessory electrical and miscellaneous equipment for 1972 and 1974 are 1.47 and 1.65, respectively. Application of these provides a January 1974 capital cost for turbine instrumentation of $117,000 x 1.65/1.47 = $131,000.

c. Wiring, Wiring Structures, and Containers - This category includes the generator bus work, station service power wiring, control wiring, instrument wiring, underground duct runs, cable trays, conduit, and concrete footings for bus supports. Reference V-1 allocates $3,922,000 for basic wiring and $2,439,000 for structures and supports for a total of $6,361,000 for the 1000-MW station. Reducing this cost for application in the 100-MW station yields a capital cost of $636,000. Using the cost index for accessory electrical and miscellaneous equipment, the January 1974 capital cost for this item is $636,000 x 1.65/1.47 = $714,000.

d. Water Treatment - Reference V-1 indicates a cost of $1,280,000 for water treatment in a 1000-MW plant. Reducing this for the 100-MW solar conversion plant, the cost would be $128,000. The cost index for materials for 1972 and 1974 are indicated in Ref V-4 as being 139 and 154, respectively. Application of these to the water treatment costs provides a January 1974 cost of $128,000 x 154/139 = $142,000.

e. Professional Services, Construction Supervision, and Temporary Facilities - The capital cost in this category is indicated in Ref V-1 to be $28,015,000 for the 1000-MW plant. The comparable cost for the 100-MW solar energy plant would be $2,802,000. Reference V-4 provides a cost index for skilled labor in 1972 and 1974 of 159 and 180, respectively. Application of this index yields a January 1974 cost for the solar plant of $2,802,000 x 180/159 = $3,172,000.

f. Turbine-Generator Foundation & Supports, Lubrication System, & Gas System - Reference V-1 provides the following breakdown in this category for the 1000-MW system:

<table>
<thead>
<tr>
<th>Description</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foundation and Support</td>
<td>$876,000</td>
</tr>
<tr>
<td>Lubrication System</td>
<td>256,000</td>
</tr>
<tr>
<td>Gas (Cooling) System</td>
<td>112,000</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$1,244,000</strong></td>
</tr>
</tbody>
</table>

Reducing these costs to represent the 100-MW solar plant, the capital cost is $124,000. Application of the construction cost index from Ref V-4 provides the January 1974 cost of $124,000 x 159/143 = $138,000.
s. Boiler Monitor and Control Instrumentation - A cost for this item of $2,275,000 for the 1000-MW plant is indicated in Ref V-1. Reducing this for a 100-MW plant provides a capital cost of $228,000. There will be eight boilers used in the solar energy system, which will increase the instrumentation required for these additional boilers. However, there will be some commonality and the cost of this category will not increase linearly with the number of boilers. Considering that this is a pilot plant operation and extensive engineering test data will be required for design analysis, we estimate a factor of 4 increase in cost for the solar plant instrumentation. The final capital outlay of this instrumentation is 4 x $228,000 = $912,000. When the cost index for accessory electrical and miscellaneous equipment is applied to this cost, the January 1974 capital cost becomes $912,000 x 1.65/1.47 = $1,024,000. On subsequent operational plants this cost item may be subject to significant reduction as the need for test data decreases.

t. Turbine-Generator - Data obtained from the General Electric turbine-generator catalog indicates that a 100,000-kW nonreheat, turbine-generator with an 8-in. Hg back pressure would cost $6,580,000 in 1972 dollars. Application of the accessory electrical and miscellaneous equipment cost index provides a January 1974 turbine-generator cost of $6,580,000 x 1.65/1.47 = $7,386,000.

u. Condensate and Feed-Water Circulation System - A 1000-MW plant requires a capital outlay of $3,607,000 for these items (Ref V-1). Reduction to the 100-MW solar energy plant size results in a capital cost of $361,000. Reference V-4 provides a cost index for pumps and prime equipment for 1972 and 1974 of 1.45 and 1.63, respectively. Application of these indices to the cost yields a January 1974 required cost of $361,000 x 1.63/1.45 = $406,000.

v. Air Cooler Condenser - A quotation from GEA Airexchangers, Inc. indicates that an air cooled condenser for a turbine-generator, with a back pressure of 6 in. Hg, operating in Phoenix, Arizona, handling a steam flow of 623,000 lb/hr would cost approximately $5,400,000. This is a current price. The 6 in. Hg back pressure indicated for this condenser was chosen by means of an analysis determining the total cost of many items as a function of back pressure. The details of this analysis are presented in Section E.

w. Steam Generator - Reliance on similarities, scalable relationships, and judgment of the impact of solar unique design characteristics provided the basis for the steam generator (tower mounted boiler-superheater units) estimate. From Ref V-1, a reference steam generator cost of $28,675,000 for a 1000 MWe conventional power plant was obtained (specific cost = $28.67/kw). Material cost included in this estimate was $19,925,000 ($19.92/kw) and the combined cost for erection, cleaning, and checkout testing included was $8,750,000 ($8.75/kw).
Compensating characteristics led to the judgment that the material value approximates the material required for the group of small (eight steam generators producing steam for 12.5 MWe segment of the plant output) generators distributed between the solar plant modules. The apparent excessive comparable material, which would be attributed to the conventional plant because of its higher operating pressure and use of large diameter piping to accommodate a re-heat cycle, is balanced by the increased heat exchanger wall area required as a result of the single side heating in the solar steam generators.

Erection costs for a single 100 MWe power plant steam generator would scale to a value of $875,000. Pending a more complete design and analysis of the tower mounted installation, a comparable value should be estimated for each of the eight steam generators in the solar plant without modification for the smaller scale of the individual units. The expected cost benefit from the smaller size units (12.5 MWe) will be largely negated by the added complexity of erection on top of the towers. The resulting material and labor estimate for the steam generator is $92.5/kw, a factor of 3.23 times the steam generator cost of a conventional power plant.

x. **Mirrors** - A quotation from the Gardner Mirror Corporation indicates that a 5x5-ft mirror would cost $25 in 1973 prices. Sixteen of these mirrors would be required for each heliostat; there are 2002 heliostats per boiler field and there are eight boiler fields per installation. This results in a total capital expenditure of $25 x 16 x 2002 x 8 = $6,406,000. The materials cost indices to be applied as obtained from Ref V-4, are 165 for 1973 and 154 for 1974. This results in a January 1974 cost of $6,406,000 x 154/165 = $5,979,000.

y. **Heliostat Fabrication and Installation** - A preliminary design configuration for the structural equipment of the heliostat was established under a separate Martin Marietta-sponsored program in sufficient detail to permit labor and material requirements to be estimated. The total estimated cost for the structural elements of the individual 20x20 ft heliostat was $2410. For a total plant requiring 14,720 heliostats, the estimate is $35,475,200. This forms the largest single cost element of the solar plant (39.6%) and is the most likely to benefit substantially from the cost reduction attainable by use of high volume production tooling.
z. Tracker Control – A preliminary design heliostat tracker control and drive subsystem was established during a separate Martin Marietta-sponsored program and a breadboard model assembled as shown in Figure V-2. The cost allocation goal established for the subsystem was $441 per heliostat, which appears now to be realizable. This extrapolates to $6,491,000 for the 100 MWe plant. As with the heliostat structure, the elements of this subsystem should yield cost reduction benefits when large volume production tooling is available.

aa. Transmission Lines – Reference V-3 indicates that for a line voltage of 13.2 kV and single wood poles with no overhead ground wires, the cost for transmission lines would be $7500 per mile (1968 prices). It is estimated that 5 miles of transmission line will be required for the pilot plant installation, dictating that the total cost will be $7500 x 5 = $38,000. Reference V-3 also provides the cost indices for transmission lines for 1968 and 1974. These are 1.104 and 1.60, respectively. The application of these to the total cost provides the January 1974 transmission line cost of $38,000 x 1.60/1.104 = $55,000.

Operating Costs

a. Operations Supervision and Engineering – Data from Ref V-2 indicate that the production expense for this item will be $18,000 for a 100-MW fossil fuel plant. There should be little or no deviation from this expense for the solar energy system. The data from Ref V-2 are dated 1971. A suitable cost index for skilled labor was obtained in Ref V-4. The index for 1971 was 159, while that for 1974 was 180. The January 1974 expense for this item would then be $18,000 x 180/159 = $20,000.

b. Steam Expenses – Included in this category are the following fossil fuel system items: condenser water and demineralization, feed-water makeup and demineralization, coal stoker or oil or gas pump power, fuel oil viscosity heaters (for oil system) and ash cleaning system power (for coal systems). Only the feed-water makeup and demineralization are applicable to the solar energy system. The annual expense for this category for a fossil fuel system is $73,000 (Ref V-2). It has been estimated that only 5% of this amount will be incurred by the solar energy plant. Therefore the solar energy plant steam expense is: 0.05 x $73,000 = $3650. A cost index made up of the mean of the common labor index and the materials index can be derived from Ref V-4. These are 143.5 for 1971 and 160.5 for 1974. Application of these indices to the solar plant steam expense yields the January 1974 expense: $3650 x 160.5/143.5 = $4000.
Fig. V-2
Breadboard Heliostat Tracker and Drive Used for Evaluation of Tracking Accuracy and Projected Cost Estimation (Built under Martin Marietta Sponsorship)
c. Electric Expenses - Reference V-2 indicates that the electrical expenses for a 100-MW fossil fuel plant would be $44,000. There should be little or no change from this value for the 100-MW solar energy plant. Reference V-6 provides a cost index for electrical expenses for the years 1971 and 1973. These are 113.9 and 125.4, respectively. To bring this expense up to 1974 the construction cost index from Ref V-4 was used. This was 155 for 1973 and 159 for 1974. Application of these indices provided the January 1974 value of electric expense for the solar energy plant: $44,000 x 125.4 x 159/(113.9 x 155) = $50,000.

d. Miscellaneous Steam Power Expense - For a fossil fuel plant, this category would include such items as coal processing for storage, coal pulverizing power, lubricant and coolant for use in turbine-generator equipment, and expense in operating pollution control devices. The only major item in common with the solar energy plant is the expense for the lubricant and coolant used in the turbine-generator equipment. Reference V-2 indicates that $24,000 is the annual operating cost attributable to the miscellaneous steam power expense category for a 100-MW fossil fuel system. It is estimated that only 15% of this expense will be incurred in the operation of the solar energy plant. The expense for the solar energy plant will, therefore, be $24,000 x 0.15 = $3600. Applying the combination materials/labor index, mentioned previously, to this amount yields $3600 x 160.5/143.5 = $4000 as the January 1974 annual expense for this category.

e. Maintenance Supervision and Engineering - An annual expense of $12,000 for this category is the amount indicated in Ref V-2 for a 100-MW fossil fuel plant. The 100-MW solar energy plant should have little or no deviation from this amount. Application of the skilled labor index to this expense yields the annual expense at the solar energy plant for this category: $12,000 x 180/159 = $14,000.

f. Maintenance of Structures - Data from Ref V-2 indicate that a 100-MW fossil fuel plant has an annual expenditure of $6000 for maintenance of structures. The structure of the main plant of the 100-MW solar energy system is smaller than that of the fossil fuel system because of the lack of boiler and fuel handling facilities. A reduction in maintenance expense is therefore called for. However, additional maintenance is required on the 8 boiler towers and 16,016 heliostats. Additions in maintenance expense must be made for these. These changes from the fossil fuel plant expense are reflected in the following estimates:

1) Reduce 15% for lack of boiler and fuel handling facilities:

2) Increase 120% for maintenance of heliostats;

3) Increase 50% for maintenance of boiler towers.
These modifications dictate a total annual expense of $6000 \times 2.55 = $15,300 for maintenance of structures for the solar plant. The combination materials-labor index derived from Ref V-4 can now be applied to determine January 1974 expense for this item: $15,300 \times 160.5/143.5 = $17,000.

g. Maintenance of Boiler Plant — A total annual expense of $47,000 for a 100-MW fossil fuel plant is indicated in Ref V-2 for this category. The solar energy plant has the same total boiler capacity as this plant, but this capacity is distributed among eight boilers. As this necessarily increases the effort in their maintenance due to more materials to replace, travel required between boilers, duplication of effort for each boiler, etc, it is estimated that a factor of 2 must be applied to the fossil fuel plant expense to obtain a reasonable estimate for the solar plant: $47,000 \times 2 = $94,000. Application of the combination materials-labor index derived from Ref V-4 provides the January 1974 expense for this item: $94,000 \times 160.5/143.5 = $105,000.

h. Maintenance of Electric Plant — Reference V-2 indicates that an annual expense of $27,000 can be expected for this item for the 100-MW fossil fuel plant. There will be little or no change in this figure for the solar energy plant. The combination materials-labor cost index derived from Ref V-4 can be applied to determine January 1974 annual expense: $27,000 \times 160.5/143.5 = $30,000.

i. Miscellaneous Maintenance — The miscellaneous maintenance expense for a 100-MW fossil fuel plant amounts to $4000 as indicated in Ref V-2. The 100-MW solar plant would have a comparable expense, plus additional outlay for such items as maintaining the longer pipe runs from the main station to the boiler towers, the periodic cleaning of the heliostat mirrors, and maintenance of the access roads between the heliostats and around the fields. It is estimated that a factor of 2 be applied to the comparable fossil fuel expense: $4000 \times 2 = $8000. Application of the combined materials-labor index derived from Ref V-4 provides the January 1974 miscellaneous maintenance expense: $8000 \times 160.5/143.5 = $9000.

j. Transmission Line Expense — Reference V-3 indicates that the annual expense for transmission lines is $165/mile (1970 prices) for a 34.5-kV line with single wooden poles. The pilot solar conversion plant will have five miles of line indicating a total annual expense of $165 \times 5 = $825. Reference V-4 indicates that the cost indices for 1970 and 1974 for transmission lines are 1.34 and 1.84, respectively. Application of these yields the January 1974 annual expense: $825 \times 1.84/1.34 = $1000.
During the investigation of costs for various equipment to be used in the solar energy conversion system, it was found that extremely high costs will be incurred in the air cooled condenser if this condenser is designed for the desired turbine-generator exhaust back pressure of 3½ in. Hg. Increasing this back pressure will reduce the required condenser size and cost as this will increase the saturation temperature of the exhaust steam allowing a larger temperature differential between the steam and the high ambient daytime temperatures in the U.S. Southwest. However, this increased back pressure reduces the turbine-generator efficiency requiring additional heliostats to compensate for the lower generator output so that the total power output of the system will remain at the required 100 MW. This necessitated a cost tradeoff study, the results of which are shown in Fig. V-3.

To create this figure, equations were derived that related the cost of the condenser to the back pressure and related the increase in the number heliostats required to the back pressure. These are as follows:

\[ C_c = 3.16 \times 10^7 - 8.82 \times 10^6 P + 9.50 \times 10^5 P^2 - 3.46 \times 10^4 P^3 \]

where \( C_c \) = cost of the condenser, dollars,
\( P \) = back pressure of the turbine, in. Hg.

This equation is accurate for 3.5 ≤ \( P \) ≤ 12:

\[ N = 1 - 1.77 \times 10^{-2} P + 6 \times 10^{-3} P^2 - 2.67 \times 10^{-4} P^3 \]

where \( N \) = the factor by which the number of heliostats required for 3½ in.-Hg system must be multiplied to obtain the number required for back pressure, \( P \),

\( P \) = back pressure of the turbine, in. Hg.

This equation is accurate for 3.5 ≤ \( P \) ≤ 8.

If \( C_h \) = cost of the heliostat fields for \( P \) = 3.5 in. Hg, then \( NC_h \) = cost of heliostat fields for back pressure, \( P \). Assuming the cost of the turbine-generator to be \( C_t = $7,300,000 \) (it actually varies from $7,100,000 for 3½ in. Hg to $7,400,000 for 8 in. Hg), then the total cost of the turbine-generator, condenser, and heliostat fields would be

\[ C_s = C_c + NC_h + C_t \]  \[V-3\]
Fig. V-3
Total Cost of Turbine-Generator, Air Cooled Condenser, and Heliostat Fields vs Turbine-Generator Back Pressure
Because a refined value of $C_h$ has not yet been determined (early estimate of maximum value was approximately $39,500,000$), it was varied parametrically in this equation with the results being those shown in the attached graph.

From Fig. V-3, it can be concluded that even though the value of $C_h$ is not yet known, the total cost of these relatively expensive items can be minimized by designing the system for a back pressure of 6 to 7 in. Hg.

Many other items in the system that would be affected by these tradeoffs were not taken into account in this analysis. For instance, the varying quantities of heliostats would require varying quantities of land upon which to construct them, which in turn may alter the quantity of piping running to and from the heliostat fields. Boiler tower height may also be affected by heliostat field size. All of these alterations will modify the overall system cost. These are under study and will be discussed in future reports; however, it appears that the optimum back pressure, from an economic viewpoint, will likely be closer to 6½ in. Hg than the initial 3½ in. Hg value.

F. REFERENCES


VI. Steam Generator Design and Analysis
To make the bench model as meaningful as possible, we first performed a preliminary design of a full-scale (100 MWe) system along with its cavity receiver. We then made the bench model a scale model of the full-scale receiver to the maximum extent practical.

A. OVERALL SYSTEM ANALYSIS

The selected design for the 100-MWe system is shown in Fig. VI-1. This system uses a single 100-MWe turbine-generator with steam inlet conditions of 950°F and 1250 psig. The gross heat rate for the selected turbine-generator is 9077 Btu/kW-hr (see Table A-1 of Appendix A). A correction of 1.05 was used to account for a higher backpressure (6-in. Hg for an air-cooled condenser) and a factor of 1.03 was used to account for auxiliary power, giving a net heat rate of 9077 x 1.05 x 1.03 = 9780 Btu/kW-hr. This is equivalent to an efficiency of 35% (power out/heat into steam).

The assumption of a 6-in. Hg backpressure (at a saturation temperature of 141°F) is very important to the overall efficiency of the system (see Fig. A-1 of Appendix A).

A preliminary system piping analysis was conducted to derive the allowances for the pressure and temperature drops shown in Fig. VI-1. It is particularly important to allow a large (100-psi) pressure drop in the superheater to maintain high heat transfer coefficients to keep the tubes cooled. The selected cycle uses five feedwater heaters that operate with steam extracted from the turbine, resulting in a feedwater temperature of 439°F. The use of a conventional, regenerative feedwater heating cycle improves the thermodynamic efficiency and provides a method of de-aerating the feedwater. Of the energy used 74% is in the boiler/preheater, and 26% is used in the superheater.

1. Rationale for the Selected 100-MWe System

The thermal efficiency of the system should be as high as possible, consistent with safe design practices for the selected materials and type of power plant. This factor is very important because the largest cost in a solar power plant is for the mirror collection system. Its cost tends to be inversely proportional to the plant's thermal efficiency. Thermal cycling of the equipment, particularly the turbine, is also an important consideration in a solar plant. Also, the cost of conversion equipment should be minimum. However, because the collection system is much more expensive than the conversion equipment, increasing expenditures on
Fig. VI-1 100-MWe Power System Schematic
conversion equipment to improve plant efficiency is generally justified. Finally, the equipment should be as simple and reliable as possible.

The single most important variable that affects the thermal efficiency is the temperature at the turbine inlet. Modern steam power plants commonly use a value of 1000°F. However, detailed discussions of possible thermal cycling problems with steam turbine experts led us to select an inlet temperature of 950°F. We also selected a relatively low inlet pressure (1250 psig) to reduce the case thickness and minimize thermal stresses during transient operation (see Appendix A for a detailed discussion of turbine cycling).

A single 100-MWe turbine-generator is significantly more efficient than multiple units. For example, the single unit is about 10% more efficient than four units (see Table A-1 of Appendix A) and costs about 25% less. Moreover, with a single unit, the cost of other equipment such as the condenser, feedwater heaters and deaerators, will be significantly lower. A single unit is also more reliable and requires less maintenance because much less machinery is used. Finally, units much larger than 100 MWe have been successfully used in cyclic operations comparable to those required in a solar plant (see Appendix A).

A reheat cycle is not recommended because it adds significant complexity to a solar plant. The potential 5% gain in efficiency would probably be offset by added costs for heat exchangers, piping, and controls.

Control Concept

The control concept for the 100-MWe system is shown in Fig. VI-2. Pressurized water is supplied to the boilers by a feed pump. A level sensor in each boiler drum then controls a valve that regulates the water level by controlling the flow rate. The superheater outlet temperature is controlled by controlling the heat flux to the cavity—that is, by pointing the mirrors so their flux impinges either on the superheater or the boiler. Typically, during startup, some of the superheater mirrors will be aimed at the boiler cavity, causing the superheater temperature to fall below that required. The superheater temperature will then be increased to the desired value by moving the focusing mirrors from the boiler to the superheater, which will reduce the flow from the boiler and increase the heat to the superheater. The exit pressure is controlled by a pressure control valve. Turbine bypass valving is used during startup to achieve the desired steam conditions before starting the turbine.
Fig. VI-2 Solar Power Plant Control Concept
3. **Design of the 100-MWe Cavity**

The design for the 100-MWe cavity is shown in Fig. VI-3. A 20x20-ft opening is used to collect the solar flux from the mirrors. To minimize the size of the cavity opening, a 5-ft-wide secondary collector is provided to collect the "fringe" energy from the mirrors. An insulated door, integral with the secondary collector, protects the cavity from the environment (e.g., sand, dust, and birds) during shutdown, minimizes the temperature loss during shutdown (due to cloud cover or overnight), minimizes thermal stresses in the hardware, and reduces the startup transient time required. Twelve inches of insulation will be used on all sides, including the door.

A preliminary analysis was conducted to estimate the cavity efficiency as a function of the L/D ratio (see Fig. VI-4). Although the convective loss as a function of L/D ratio is not known, it probably increases with decreasing L/D. As a result of these analyses, we selected a depth of 30 ft, which gives an L/D of approximately 1.5 (30-ft deep/20-ft opening). In our design we have assumed that low solar absorptance (α) and high emittance (ε) coatings compatible with the solar furnace environment (i.e., high solar flux and high temperature in air) will be available. One possibility is a ceramic sheet attached to the steel wall. The boiler tubes are plain 1.0-in. OD and 0.10-in. thick carbon steel. The superheater tubes are 3/4-in. OD and 0.080-in.-thick stainless steel. The same tubes will be used in the bench model. All boiler tubes are in parallel. There are 146 tubes in parallel in the superheater.

The shape of the cavity was selected to minimize the amount of direct flux impinging on the uncooled surfaces. As shown in Fig. VI-3, the sides are at a 45° angle and the top and bottom are at a 221/2° angle. This configuration prevents direct solar energy impingement on the uncooled surfaces and allows direct incident solar only on the boiler/superheater surfaces. To achieve the required L/D ratio without an excessively large heat exchanger, the 45° angle in the plan view was held for only about one-third of the depth. The overall configuration results in an average flux into the boiler/superheater surfaces of about 40,000 Btu/hr-ft².

The high-temperature superheater was located on the rear cavity surface to minimize the IR losses. Also, the heat flux on the back surface is expected to be relatively constant throughout the day. This is important to superheater operation because the tube safety factors depend on the local flux level. Therefore, the more constant the flux, the better the safety factors. The flux values on the back surface are not expected to vary more than 10% with time of day, whereas the flux on the side surfaces could vary up to 25% with time of day.

VI-5
Superheater 1, 73
3/4-in. Tubes in Parallel

Boiler Sections
1-in. OD Tubes
Side by Side
in Parallel

6-in. Dia (4 Places)

Note: All cavity internal surfaces not shown for the boiler or superheater are covered with a ceramic sheet.

Secondary Collector/Door in Closed Position

Downcomers

50 ft 10 in.

Ion Pump

VI-3 Conceptual Layout of 100-MWe Receiver

VI-7 and VI-8
A steam drum is used to separate the water from the steam, provide a reliable measure for the water level, provide an adequate volume to prevent water from flowing into the superheater during transients, and house the steam-separator equipment. A unique requirement for the solar boiler is that its axis must be tilted downward to properly collect the solar energy. The configuration shown in Fig. VI-3 has the drum parallel to the tilt axis to provide a reliable water-level measurement regardless of the tilt angle.

Boiler Circulation by natural convection is preferred. Preliminary analysis indicates that velocities in excess of 1 fps in the riser tubes can be obtained. However, some potential advantages of forced circulation should be considered. A schematic of a forced circulation system is shown in Fig. VI-5. Note that the flow is always upward during heat addition. To ensure that the fluid is in the nucleate boiling heat transfer regime, all tubes can be orificed to match the flow with the heat flux to maintain...
Fig. VI-5 Schematic of Circulation System

Note: Dashed lines are downcomers.
a minimum of 30% water (by volume) at all times. Strainers must be installed upstream of all orifices to prevent the orifices from becoming plugged and burning out the tubes.

Force (i.e., controlled) circulation has the following potential advantages over natural convection:

1) Forced circulation eliminates some of the questions about flow stability in a low-velocity natural convection system;

2) Forced circulation enables positive control (by orificing) to be used to match flux patterns, even after the boiler is built. In addition, the flow paths can be changed by using series-parallel combinations;

3) Forced circulation makes it possible and desirable to use smaller diameter tubes. This has advantages in addition to reducing the flow through the circulating pump,
   a) A given surface area covered with 1-in.-diameter tubes weights about half as much as an equal surface area covered with 2-in.-diameter tubes. Weight may be an important cost factor considering that the boilers are located on the top of a 400-ft-high tower,
   b) There will be much less water in the boiler, which improves the transient response,
   c) Smaller downcomers can be used,
   d) A lower overall recirculation ratio (weight of steam generated/weight of water circulated) can be used because the circulation can be controlled to match the heat flux variation throughout the boiler.

These advantages may outweigh the disadvantages of the additional pumps required, the slight efficiency loss (<0.5%) due to the power required to drive the pump, and the additional welding involved with smaller diameter tubes. The bench model will be designed so both forced and natural circulation methods can be tested and evaluated.
B. PERFORMANCE ANALYSIS OF THE 100-MWe CAVITY

Table VI-1 summarizes the results of our performance analysis.

**Table VI-1 100-MW Cavity Performance**

<table>
<thead>
<tr>
<th>Losses</th>
<th>Percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reflected Solar</td>
<td>0.8</td>
</tr>
<tr>
<td>IR</td>
<td>1.2</td>
</tr>
<tr>
<td>Convection</td>
<td></td>
</tr>
<tr>
<td>No Wind</td>
<td>2.2</td>
</tr>
<tr>
<td>10-mph Wind</td>
<td>3.2</td>
</tr>
<tr>
<td>15-mph Wind</td>
<td>3.7</td>
</tr>
<tr>
<td>Insulation</td>
<td>0.2</td>
</tr>
<tr>
<td>Cavity Efficiency (Heat in Steam/Energy into Cavity)</td>
<td></td>
</tr>
<tr>
<td>No Wind</td>
<td>95.6</td>
</tr>
<tr>
<td>10-mph Wind</td>
<td>94.6</td>
</tr>
<tr>
<td>15-mph Wind</td>
<td>94.1</td>
</tr>
</tbody>
</table>

The results are based on detailed analysis of the receiver cavity, subdivided into thermally coupled nodes as shown in Fig. VI-6, and the scale-model convection test results described in Appendix C.

The full-scale cavity performance was determined with the same analytical techniques described later in the detail discussion of the bench model design. The solar energy distribution over the cavity nodes is shown in Table VI-2. The surface temperatures resulting from the thermal analyzer computer program are shown in Fig. VI-7.

A temperature profile relating the node temperatures and the steam temperatures along the flow path of the superheater is shown in Fig. VI-8. For this analysis, all uncooled surfaces were assumed to have a solar absorptance of 0.3 and an emittance of 0.8. These properties are typical of a white ceramic surface. The resulting overall system energy balance is shown in Table VI-3.

Previous analyses indicated that a uniform incident solar flux on the boiler and superheater surfaces produced an exit superheater temperature greater than 1000°F. To reduce the superheater exit temperature to a value of 955°F or less, some of the incident energy was diverted from the superheater section to the boiler section. In this case, a portion of the incident solar energy was removed from nodes 1, 2, 11, 12, 13, 18, 19, and 24 (see Fig. VI-6), and an equivalent amount of energy was added to nodes 25, 26, 31, 32, 38, 40, 43, and 46; the total incident energy was maintained constant. According to the flux distribution shown in Table VI-2,
Fig. VI-6 Cavity Surface Node Numbers

Legend:
- Boiler Surface
- Superheater Surface
- Ceramic Surface

VI-13
<table>
<thead>
<tr>
<th>Nodes</th>
<th>Incident, KW(_{th})</th>
<th>Net Absorbed Solar, KW(_{th})</th>
<th>Nodes</th>
<th>Incident, KW(_{th})</th>
<th>Net Absorbed Solar, KW(_{th})</th>
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<td>4, 10</td>
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<td>330.91</td>
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<td></td>
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<td>412.01</td>
<td>397.12, 399.18</td>
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<td>Total,</td>
<td>28,447.48</td>
<td>27,399.4</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td>Boiler</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Aperture</td>
<td>--</td>
<td>379 (Reflected)</td>
</tr>
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<td></td>
<td></td>
<td></td>
<td>Total</td>
<td>38,389</td>
<td>38,391</td>
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Table VI-2: Full-Scale Receiver Insolation Nodal Distribution
Table VI-3
System Energy Balance, Full-Scale Receiver

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<thead>
<tr>
<th></th>
<th>Btu/hr</th>
<th>Percent</th>
</tr>
</thead>
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<td>IR Aperture Loss</td>
<td>1,366,228</td>
<td>1.04</td>
</tr>
<tr>
<td>Solar Aperture Loss</td>
<td>1,294,419</td>
<td>0.99</td>
</tr>
<tr>
<td>Convection Aperture Loss</td>
<td>2,453,535</td>
<td>1.87</td>
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<td>Total Loss</td>
<td>5,114,182</td>
<td>3.90</td>
</tr>
<tr>
<td>Solar to Preheat</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Solar to Boiler</td>
<td>93,560,558</td>
<td>71.42</td>
</tr>
<tr>
<td>Solar to Superheat</td>
<td>32,325,260</td>
<td>24.68</td>
</tr>
<tr>
<td>Total Incident Solar</td>
<td>131,000,000</td>
<td>100.00</td>
</tr>
</tbody>
</table>

The superheated steam leaving on node 18, (associated fluid node 218, Fig. VI-8) is about 840°F, and the steam temperature leaving nodes 16 and 17 (associated fluid nodes 216 and 217) is about 960°F. Because the steam flow rate is the same in each of the three passes, the steam mixture in the exit manifold is about 920°F.

The nonuniform solar flux distribution on the superheater surface in this example produces an extreme temperature variation in the exit superheater temperatures. Techniques for maintaining a more uniform flux over the entire superheater area would result in more uniform temperature rises in the parallel superheater passes.

The use of a glass window at the aperture is not recommended because:

1) The addition of the glass would result in a solar reflection loss of about 11%, which is greater than the total of radiation and convective losses from our cavity design;

2) The heat flux at the opening could be more than 300,000 Btu/hr-ft², which could result in severe local hotspots if there were contamination on the glass.

1. Circulation Analysis

The reasons for considering forced circulation were discussed earlier. Based on a literature search, we have selected the following design criteria for boiler circulation:

1) Maintain a minimum of 30% water by volume in the tubes;

2) Maintain a minimum upward velocity of 2 fps;
Fig. VI-8 Superheater Temperature Profile, Full-Scale Receiver
3) Keep the circulation pump power less than 0.5% of the plant output.

The circulation analysis was done for the boiler tube with the highest expected flux value of 50,000 Btu/hr-ft² (i.e., 40,000 Btu/hr-ft² plus 25%). The percent steam by weight is calculated as follows:

\[ Q_{IN} = (q/A) \frac{D}{12} \times L \]

where

- \( Q_{IN} \) = heat flow into tube, Btu/hr,
- \( (q/A) \) = heat flux, Btu/hr-ft²,
- \( D \) = diameter, in.,
- \( L \) = length, ft.

\[ \dot{W}_s = \frac{Q_{IN}}{H_{f-g}} \]

where

- \( \dot{W}_s \) = steam generated in tube, lb/hr,
- \( H_{f-g} \) = latent heat, Btu/lb.

From the continuity equation, the liquid flow rate into the tube can be calculated as follows:

\[ \dot{W}_{LIN} = \rho_{LIN} A V_{LIN} = \frac{0.785 \ D^2 \ \rho_{LIN} V_{LIN} \times 3600}{144} \]

- \( \dot{W}_{LIN} \) = liquid flow rate in lb/hr,
- \( \rho_{LIN} \) = liquid density, lb/ft³,
- \( A \) = tube area, ft²
- \( V_{LIN} \) = velocity of liquid in, fps.

The percent steam by mass is:

\[ \% (\dot{W}_s) = \frac{\dot{W}_s}{\dot{W}_{LIN}} \times 100 = \frac{(q/A) \ DL \times 144 \times 100}{12 \ H_{f-g} (0.785) \ D^2 \ \rho_{LIN} V_{LIN} \times 3600} \]

\[ = \frac{0.425 \ (q/A) L}{H_{f-g} D \ \rho_{LIN} V_{LIN}} \]
The following values used to calculate the percentage of steam generated in the tube as a function of liquid velocity:

\[ L = 40 \text{ ft}; \]
\[ (q/A) = 50,000 \text{ Btu/hr-ft}^2; \]
\[ H_{f-g} = 556 \text{ Btu/lb}; \]
\[ D = 0.8 \text{ in.}; \]
\[ \rho_{\text{LIN}} = 42.5 \text{ lb/ft}^3; \]
\[ \frac{\% \dot{W}_s}{V_{\text{LIN}}} = \frac{0.425 }{556 \times 0.8 \times 42.5} \times \frac{40}{V_{\text{LIN}}} = \frac{45}{V_{\text{LIN}}}. \]

The maximum allowable % (\(\dot{W}_s\)) can be calculated as follows:

\[ \rho_L = \text{liquid density} = 42.3 \text{ lb/ft}^3; \]
\[ \rho_v = \text{vapor density} = 3.6 \text{ lb/ft}^3; \]

Maximum steam by volume = 70% (to maintain 30% liquid by volume);

\[ \text{Max} \% (\dot{W}_s) = \frac{0.7 \times 3.6}{0.7 \times 3.6 \times 0.3 \times 42.5} \times 100 = 16.5\%. \]

The minimum liquid velocity into the tube is

\[ V_{\text{LIN}} = \frac{\frac{45}{16.5} = 2.73 \text{ fps.}}{16.5} \]

The recirculation ratio (R) is

\[ R = \frac{100}{\% (\dot{W}_s)} = \frac{100}{16.5} = 6.0. \]

The minimum recirculation flow rate is \(\dot{W}_R = R \dot{W}_s\)

\[ 6 \times 120,800 = 725,000 \text{ lb/hr.} \]

However, the total recirculation flow rate must also meet the requirement that all tubes must maintain the minimum velocity of 2.73 fps. There are a total of 553 boiler tubes, in parallel, each with an ID of 0.8 in.:

\[ \dot{W}_R = \rho AV \]
\[ \dot{W}_R = 42.5 \times \frac{553 \times 0.785 \times 0.8^2}{144} \times 2.73 \times 3600 = 805,000 \text{ lb/hr.} \]
Using a pump head rise of 20 psi and a pump efficiency of 70%, the pump power is

\[ \text{Power In} = \frac{805,000 \text{ lb/hr} \times 20 \text{ lb/in.}^2 \times 144 \text{ in.}^2/\text{ft}^2 \times \frac{1 \text{ ft}^3}{42.5 \text{ lb}}}{778 \text{ ft-lb/ft-lb} \times 3413 \text{ Btu/ft-lb} \times \frac{1 \text{ kWh}}{0.7 \text{ (Fluid Power)}}} = 29.5 \text{ kW.} \]

Percentage of total output is

\[ \frac{29.5}{100,000} \times 100 = 0.0295\%. \]

2. **Insulation Performance**

A preliminary analysis of the insulation heat leak is presented below. The heat loss, q, is calculated as follows:

\[ q = \frac{K A \Delta T}{X} = \frac{0.5 \times 5760 \times 600}{12} = 144,000 \text{ Btu/hr,} \]

where

\[ K = 0.5 \text{ Btu/hr-ft-°F/in. for medium-temperature block insulation (from Ref IV-1, page 16-6)}, \]

\[ X = 12 \text{ in.}, \]

\[ A = 5760 \text{ ft}, \]

\[ \Delta T = 650 - 50 = 60\text{°F}. \]

Therefore,

\[ \text{% Loss} = \frac{144,000}{125 \times 10^6 \times 100} = 0.12\%. \]

Allowing for other heat leaks, a value of 0.2% is reasonable.

3. **Temperature Drop Overnight and During Cloud Interruptions**

Using the heat leak previously calculated and the heat capacity of the boiler shown below, we made a conservative estimate to determine the temperature drop after a 16-hr shutdown if an insulated door is used. The estimated drop is about 160° F. We feel that the magnitude of this differential is important, particularly from the standpoint of reducing thermal stresses and extending the life of the system. Also a cloud interruption of 1 hour will result in a temperature drop of only about 10° F. The boiler heat capacity is:
Tubes = 9500 Btu/°F;
Water in tubes = 1000 Btu/°F;
Water in drum = 3000 Btu/°F.

4. Superheater Pressure Drop

In the superheater, there are 146 tubes in parallel. Each tube has an ID of 0.5 in. The specific volume at the outlet is 0.604 ft/lb. The velocity is

\[ V = \frac{120,800 \times 0.604 \times 144}{3600 \times 146 \times 0.785(0.5)^2} = 103 \text{ fps}. \]

A velocity head is

\[ \frac{\rho v^2}{2g} = \frac{103^2}{0.604 \times 64.4 \times 144} = 1.9 \text{ psi}. \]

The pressure losses in velocity heads were estimated as follows:

<table>
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<th>Velocity Heads</th>
</tr>
</thead>
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<tr>
<td>Entrance Loss, 0.5 Each x 2</td>
</tr>
<tr>
<td>Exit Loss, 1 Each x 2</td>
</tr>
<tr>
<td>Pipe</td>
</tr>
<tr>
<td>( f \frac{L}{D} ) = 0.02 \times 80/0.5 \times 12</td>
</tr>
<tr>
<td>Total</td>
</tr>
</tbody>
</table>

Therefore,

\( \Delta P = 41.4 \times 1.9 = 78.5 \text{ psi (allow 100 psi)}. \)

5. Superheater Heat Transfer Coefficient

The superheater heat transfer coefficient was calculated as:

The mass flow rate \( G = \dot{W}/A, \)

\[ [VI-12] \quad G = \frac{120,800}{146 \times \frac{0.785 \times 0.5^2}{144}} = 610,000 \text{ lb/hr-ft}^2. \]
The following method was used:

\[ U_{ct} = U_{ct,pp} F_T t' \]

\[ U_{ct} = 2000 \text{ Btu/hr-ft}^2\text{-°F}, \]

\[ F_T = 0.35 \text{ at 1000°F, and 1300 psi}, \]

\[ F_T = 0.97, \]

\[ U_{ct} = 2000 \times 0.35 \times 0.97 = 630 \text{ Btu/hr-ft}^2\text{-°F}. \]

C. DESIGN OF THE BENCH MODEL SYSTEM AND CAVITY

1. Design of the Bench Model System

The primary criterion for designing the bench model system is to make it a scale model of the 100-MWe plant to the maximum extent practical. It is particularly important to maintain the same steam conditions.

The proposed bench model system schematic is shown on Drawing SK-SESG1000 (Fig. VI-9) along with the operating conditions. The system generates steam at the conditions required for a full-scale plant (955°F, 1300 psi) using feedwater simulating the outlet of the final feedwater heater of the full-scale plant (see Fig. VI-1). The feedwater is pressurized by a positive displacement pump (P1) to 1600 psia and is heated to the required boiler inlet temperature of 439°F by the outlet steam in the feedwater heater. Control of the preheater inlet temperature is achieved by controlling the amount of feedwater that bypasses the feedwater heater through the temperature control valves (TCV-1 and TCV-2). Water level in the stream drum is maintained by the remotely operated valve FCV-1 in the water bypass around the boiler feed pump (P-1). All controls are manual. Pump speed can be controlled by manually changing the pulley diameter ratios between the motor and pump while operating. A backup level sensor, LS-1R, is provided to check the level. The boiler is designed for natural circulation. However, it is planned to buy a pump (P-2 – Table VI-4) and make provisions to install the pump if instability is encountered during testing. The boiler pressure is controlled by PCV-1. Relief valves are provided in the boiler, superheater, and condenser for safety. The superheater pressure is controlled by the orifice between the boiler and superheater. Superheated steam is cooled in the feedwater heater and condensed.

VI-22
at 100 psia in the condenser. Condensate is further cooled in the subcooler to 225°F to keep below the upper temperature limit of 250°F on the pump and to provide a safe net positive suction head (NPSH). An acceptable pressure for commonly available condenser shell structures is 100 psia. Air is bled from the system through the purge valve (CB-1) in the condenser. The cooling water flow rate to the condenser is controlled by the valve CV-1. A level sensor (LS-2) is used in the condenser hot well to ensure that adequate makeup water is provided. A check valve (BCV) is provided at the boiler inlet for safety. Filters are provided at the inlet to the pumps and liquid control valves. Control valves are pneumatically operated from the control console. A list of all components is presented in Table VI-4, along with requirements and potential supplier part numbers. Request for quotes have been sent to potential suppliers. Final selection of parts will be based on our assessment of reliability, cost, and lead time.

Instrumentation is provided to safely operate the system and for post-test analysis. Fluid pressure, temperature, and flow rate measurements are shown in Fig. VI-9 and listed in Table VI-5. Temperatures will be measured by the use of thermocouples. The measurement system for the other parameters consists of a transducer, a pneumatic transmitter that converts the signal to a 3- to 15-psi air signal, and pneumatic tubing to transmit the signal to the control and instrumentation console. Four pneumatic recorders will be used to continually monitor the selected eight variables that will be used to control and monitor the safety of the system. The other two recorders will be used to record all the other measurements on a manually controlled intermittent schedule. One hundred thermocouples will be used to measure the thermal performance of the cavity. They will use high-temperature chromel-constantan wire. Two multipoint recorders, each with 24 measurements, will be used to record the temperature data. Manual switches will be used to select the temperatures to be measured and the time intervals. All measurements required to safely operate the system will have a continuous visual readout on the control console.

Design of the Bench Model Cavity

The bench model cavity design layout is shown in Fig. VI-10. The bench model is a 16.8% scale model of the 100-MWe system except in the following areas:

1) The number of parallel paths is reduced to maintain the heat transfer coefficients equal to the full-scale model;
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<th>Components</th>
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<th>Supplier/Part No.</th>
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<td></td>
</tr>
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<td>1. Boiler Feed Pump (P-1)</td>
<td>Operating Pressure: 85 psig Inlet, 1600 psig Outlet&lt;br&gt;Operating Temperature: 225°F&lt;br&gt;Fluid: Water&lt;br&gt;Flow Rate: 2700 lb/hr ± 20%</td>
<td>Kerr&lt;br&gt;Plunger Pump&lt;br&gt;Model KJ-2250</td>
</tr>
<tr>
<td>2. Boiler Circulation Pump (P-1)</td>
<td>Operating Pressure: 1500 psig&lt;br&gt;Operating Temperature: 600°F&lt;br&gt;Fluid: Water&lt;br&gt;Flow Rate: 100,000 lb/hr&lt;br&gt;Head Rise = 40 psi (Minimum)</td>
<td>Ingersoll-Rand&lt;br&gt;Centrifugal Pump&lt;br&gt;Model BEY 1½ x 1</td>
</tr>
<tr>
<td><strong>Heat Exchangers</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Condenser</td>
<td>Total Heat Transfer: $3 \times 10^6$ Btu/hr&lt;br&gt;Steam Inlet Conditions: 700°F, 100 psia, $H = 1310$ Btu/lb&lt;br&gt;Cooling Water Inlet: 50°F&lt;br&gt;Flow Rate: 40,000 lb/hr</td>
<td>American Heat Reclaiming Corp&lt;br&gt;Spiral Heat Exchanger, Type 1-V</td>
</tr>
<tr>
<td><strong>Valves</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Boiler Pressure Control (PCV-1)&lt;br&gt;Remote Manual</td>
<td>Maximum Operating Pressure: 1500 psig&lt;br&gt;Maximum Operating Temperature: 750°F&lt;br&gt;Working Fluid: Steam&lt;br&gt;Flow Rate: 2700 lb/hr ± 20%&lt;br&gt;Pressure Control: 1000 to 1400 psig&lt;br&gt;Actuator: Pneumatic (Remote Manual)&lt;br&gt;Positive Indicator with Remote Readout</td>
<td>Copes-Vulcan&lt;br&gt;Type D-100&lt;br&gt;Material: ASTM 2 Grade C-5</td>
</tr>
<tr>
<td>2. Feedwater Temperature Control (TCV-1 - TCV-2)&lt;br&gt;Manual</td>
<td>Maximum Operating Pressure: 1600 psig&lt;br&gt;Maximum Operating Temperature: 250°F&lt;br&gt;Working Fluid: Boiler Feed-Water&lt;br&gt;Flow Rate: 0 to 3240&lt;br&gt;Maximum ΔP: 25 psi&lt;br&gt;Maximum ΔP with Valve Closed: 150 psi</td>
<td>Copes-Vulcan&lt;br&gt;Type D-100</td>
</tr>
<tr>
<td>3. Boiler Feed Pump Isolation Manual (V1 and V2)</td>
<td>Maximum Operating Pressure: 100 psig&lt;br&gt;Inlet: 1600 psig&lt;br&gt;Operating Temperature: 250°F&lt;br&gt;Working Fluid: Boiler Feed Water&lt;br&gt;Flow Rate: 2700 lb/hr ± 20%&lt;br&gt;Maximum ΔP at Maximum Flow: 2 psi&lt;br&gt;Inlet: 2 psi&lt;br&gt;Outlet: 5 psi&lt;br&gt;Actuator: Manual</td>
<td>A&amp;N Corporation&lt;br&gt;Model No. 4415</td>
</tr>
<tr>
<td>4. Coolant Water Control (CV-1)&lt;br&gt;Remote Manual</td>
<td>Maximum Operating Pressure: 300 psig&lt;br&gt;Maximum Operating Temperature: 100°F&lt;br&gt;Working Fluid: Water&lt;br&gt;Flow Rate: 50,000 lb/hr&lt;br&gt;Maximum ΔP: 10 psi&lt;br&gt;Actuator: Pneumatic (Remote Manual)</td>
<td>Copes-Vulcan&lt;br&gt;Type D-100</td>
</tr>
</tbody>
</table>

* Supplier lead times were excessive.
<table>
<thead>
<tr>
<th>Components</th>
<th>Requirements</th>
<th>Supplier/Part No.</th>
</tr>
</thead>
</table>
| 5. Boiler Relief Valve (RV-1) (2 Required) | Maximum Operating Pressure 1600 psi  
Maximum Operating Temperature 955°F  
Fluid - Steam  
Flow Rate 3000 lb/hr (Maximum)  
Maximum ΔP 100 psi | Ferris  
Model 2741-T  
3/4 x 1 in. |
| 6. Superheater Relief Valve (2 Required)  | Maximum Operating Pressure 1600 psi  
Maximum Operating Temperature 955°F  
Fluid - Steam  
Flow Rate 3000 lb/hr (Maximum)  
Maximum ΔP 100 psi | Ferris  
Model 2741-T |
| 7. Condenser Relief Valve (RV-3) (2 Required) | Maximum Operating Pressure 125 psi  
Maximum Operating Temperature 700°F  
Flow Rate 3000 lb/hr  
Maximum ΔP 25 psi | Ferris |
| 8. Condenser Bleed (CB-1) Manual          | Maximum Operating Pressure 125 psig  
Maximum Operating Temperature 700°F  
Fluid - Air and Steam | A&N Corp  
Modgl 4410 |
| 9. Flow Control Valve (FCV-1)             | Maximum Inlet Pressure 1600 psig  
Maximum Inlet Temperature 250°F  
Outlet Pressure 100 psig  
Flow Rate 0 to 2000 lb/hr  
Fluid - Water  
Actuator - Pneumatic | Copes-Vulcan  
Type D-100-60 |
| 10. Boiler Inlet Check Valve (BCV)        | Maximum Pressure 1600 psig  
Maximum Temperature 450°F  
Flow Rate 0 to 3200 lb/hr  
Fluid - Water  
Maximum ΔP at Full Flow 25 psi | Conval 2-in.  
Y-Type Piston  
Size 12C4 |
| 11. Coolant Water Valves (CV-1, 2-34) 1 in., Manual | Maximum Pressure 150 psig  
Maximum Temperature 150°F | Copes-Vulcan  
Type D100-60 |
| 12. Boiles Shutoff Valve BSV - Remote Manual | See Fig. VI-9 | |
# Table VI-6 Instrumentation List* and Equipment Required

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Quantity</th>
<th>Measurement Range</th>
<th>Equipment Required</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Temperatures</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>a) Surface Temperatures</td>
<td>10</td>
<td>500 to 1200°F</td>
<td>1. 10,000 ft of high temperature chromel-constantan wire.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0 to 700°F</td>
<td>2. 24-channel multipoint recorders (one for each range).</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>3. 4 pen stripchart.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>4. 8 visual readout instruments (for ranges shown).</td>
</tr>
<tr>
<td>b) Fluid Temperatures</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Continuous (strip chart)</td>
<td>4</td>
<td>200 to 1000°F</td>
<td></td>
</tr>
<tr>
<td>T₁, T₄, T₅, T₇</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Visual Readout and Intermittent Recording on Multipoint</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T₈</td>
<td>1</td>
<td>0 to 300°F</td>
<td></td>
</tr>
<tr>
<td>T₉</td>
<td>1</td>
<td>0 to 500°F</td>
<td></td>
</tr>
<tr>
<td>T₁₀</td>
<td>1</td>
<td>0 to 800°F</td>
<td></td>
</tr>
<tr>
<td>T₁₄, T₂₄, T₃₄, T₆₄, T₇₄</td>
<td>5</td>
<td>0 to 200°F</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Pressure Measurement</strong></td>
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<td></td>
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</tr>
<tr>
<td>a) Continuous (strip chart)</td>
<td>3</td>
<td>0 to 1800 psi</td>
<td>1. 16 pneumatic pressure transducer/transmitters.</td>
</tr>
<tr>
<td>P₁, P₃, P₉</td>
<td></td>
<td></td>
<td>2. 1 pneumatic differential pressure transducer/transmitter.</td>
</tr>
<tr>
<td>b) Visual Readout and Intermittent Record</td>
<td></td>
<td></td>
<td>3. 3 pneumatic control recorders (4 channel); equipped with a trend/pneumatic patch board.</td>
</tr>
<tr>
<td>P₁₃, P₁₅, P₁₇, P₁₉, P₁₀</td>
<td>8</td>
<td>0 to 1800 psi</td>
<td>2-0-1800]; 1-0-2100].</td>
</tr>
<tr>
<td>ΔP</td>
<td>1</td>
<td>0 to 100 psi</td>
<td></td>
</tr>
<tr>
<td>P₁₁, P₁₂</td>
<td>2</td>
<td>0 to 200 psi</td>
<td></td>
</tr>
<tr>
<td>c) Intermittent Record</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P₂, P₃, P₇</td>
<td>3</td>
<td>0 to 1800 psi</td>
<td></td>
</tr>
<tr>
<td>d) Total Pressure Measurements</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>17</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Flowrate</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>a) Continuous (strip chart)</td>
<td>6</td>
<td>0 to 15 gpm</td>
<td>1. 8 orifice plates.</td>
</tr>
<tr>
<td>F₃, F₅, F₇, F₉</td>
<td></td>
<td></td>
<td>2. 8 pneumatic d/p cell transducer/transmitters.</td>
</tr>
<tr>
<td>F₅, F₇</td>
<td>2</td>
<td>0 to 300 gpm</td>
<td>3. 1, 4 channel pneumatic control recorder (0 to 15 gpm).</td>
</tr>
<tr>
<td>b) Visual Readout</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>F₁</td>
<td>1</td>
<td>0 to 3500 lb/hr</td>
<td>4. 1, 4 channel pneumatic control recorder (0 to 300 gpm).</td>
</tr>
<tr>
<td>F₇</td>
<td>1</td>
<td>0 to 15 gpm</td>
<td></td>
</tr>
<tr>
<td>c) Total Measurements</td>
<td></td>
<td></td>
<td>5. 2 visual readouts for the ranges shown.</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Liquid Level</strong></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Visual Readout</td>
<td>4</td>
<td>0 to 15 in.</td>
<td>1. 4 pneumatic d/p cell transducer/transmitter.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>2. 2 visual readouts.</td>
</tr>
</tbody>
</table>

**Alarms**
1. All level sensors (both high and low)
2. Temperature (T₂, high)
3. Pressure (P₁₂, high)
4. Pressure (P₁₃, high)

*Strain gauge requirements are discussed with the stress analysis later in the report.

See Fig. VI-9 for measurement location.
2) The insulation is 6 in. thick;*
3) No secondary collector is used because it is not required;
4) A reflector is used to obtain the desired flux pattern;
5) A section of the superheater is placed on the top surface because of the relatively high incident flux in that area;
6) The entire bottom surface and the top surface not covered with superheater tubes are covered with preheater tubes to assure adequate cooling of these surfaces.

A flow diagram of the assembly is shown in Fig. VI-11. The preheater consists of two heat exchanger surfaces in series. Each heat exchanger consists of 1.0-in. tubes side by side. In every case, there are four tubes per pass so the tubes can be bent with an acceptable bend radius. The number of passes for each surface is shown. The output of the preheaters is piped to the steam drum. Location of the boiler section is shown in Fig. VI-11. All boiler sections use 1-in. side-by-side tubes in parallel. Each boiler section is double-headed, upper and lower, as shown. The boiler is designed to operate with natural circulation. However, the design includes provisions to add a circulation pump at the bottom of the single downcomer if test data show it is required. The steam drum receives two-phase flow from the upper manifolds (Fig. VI-11).

The upper manifolds are "vented" at the top just downstream of the last riser to the steam (upper) portion of the drum to minimize the probability of "slug" flow in the header. Baffles are provided in the drum to prevent steam bubbles from entering the downcomer that could damage the circulating pump or significantly reduce the natural circulation flow rate. The baffles also minimize water carryover into the superheater.

*In thermal scaling the insulation thickness should theoretically not be reduced if the same thermal conductivity is used on the model and prototype. Because we would expect to use the same insulation, we should theoretically maintain the 12 in. of insulation. However, this is impractical because of corner effects and space limitations. As a result we have selected to reduce the insulation thickness to 6 in., which will result in about a 0.5% loss in the bench mode. This is considered acceptable for the purposes of this design.
[Text content from the image]
The steam is taken off the top of the drum through a single outlet. There are two superheater surfaces in series consisting of 3/4-in.-side-by-side tubes. The location of the surfaces is as shown in Fig. VI-11. Each has four tubes in parallel per pass to fabricate the bends as shown in the detail. The number of passes is given. Six-inch fiberglass insulation is used on all cavity walls to limit the heat loss. A set of detail fabrication drawings are included in Appendix B.

3. Analysis of Bench Model

The analytical model for the solar cavity is based on the MITAS thermal analyzer computer program developed by Martin Marietta for design studies of a large variety of heat transfer problems. This program uses the conventional approach of describing the physical problem by a set of simultaneous differential equations by finite differences. The physical system is described by a number of isothermal, lumped-mass nodes that are connected by appropriate radiation, conduction, convection, and fluid flow paths. The computer program solves the finite-difference network for nodal temperatures by iterative techniques. Iterative techniques are available for both transient and steady-state solutions.

The analytical model includes the three basic heat transfer processes of convection (liquid preheat, vaporization, and vapor superheat), conduction, and radiation in a single cavity. All cavity surfaces are covered by tube surfaces (except the surfaces identified in Fig. VI-12) having a high solar absorptivity ($\alpha = 0.9$) and a high infrared emissivity ($\varepsilon = 0.8$). No cavity surface requires a high solar reflectivity for temperature control. Natural convection heat transfer has been modeled and incorporated in the cavity analysis. All of the heat transfer processes are interrelated through the radiation, conduction, and convection network. The calculated convection loss of 2.57 (based on total incident energy) is slightly greater than the measured natural convective loss of 1.5%. The model has a total of 440 nodes. The cavity nodal breakdown is shown in Fig. VI-12.

Each of the fluid nodes is connected to the associated cavity surface node by a temperature-dependent conductance that includes both the convection film coefficient and the tube wall resistance. These conductances assume that the outer tube surface temperature is uniform and the transfer coefficients are based on the total tube wall area. Conductances based on only half the tube wall area had little effect on the results. The fluid film coefficient for both single-phase liquid and single-phase vapor is based on the correlation

$$[VI-12] \ hA = \alpha \left( \frac{k}{D} \right) \left( \frac{4\omega}{\pi D\mu} \right)^{0.8} \left( \frac{C\mu}{k} \right)^{0.4} (\tau DL),$$
Fig. VI-12 Cavity Surface Node Numbers
which may be written in the form

\[ hA = \alpha \pi (4/\pi D)^{0.8} (C^{0.4}k^{0.6}/\mu^{0.4}) w^{0.8}L, \]

where

- \( h \) = fluid film coefficient,
- \( A \) = surface area,
- \( \alpha = 0.023 \) (dimensionless constant),
- \( \pi = 3.1416 \),
- \( D \) = tube diameter,
- \( w \) = flow rate, mass/time,
- \( L \) = tube length,
- \( C \) = specific heat of fluid,
- \( k \) = thermal conductivity of fluid,
- \( \mu \) = viscosity of fluid.

In the computer program, the group \((C^{0.4}k^{0.6}/\mu^{0.4})\) is input as a temperature-dependent property and \( w \) is internally calculated. The temperature-dependent properties are defined in Table VI-6 and Fig. VI-13 thru VI-17. The numerical value of \( \alpha \pi (4/\pi D)^{0.8} L \) is internally calculated for each tube section.

Table VI-6 Temperature-Dependent Properties of Water

<table>
<thead>
<tr>
<th>Temperature, °F</th>
<th>Specific Heat of Fluid, ( C ), Btu/lb-°F</th>
<th>Thermal Conductivity of Fluid, ( k ), Btu/hr-ft-°F</th>
<th>Viscosity of Fluid, ( \mu ), lb/ft-hr</th>
<th>( C^{0.4}k^{0.6}/\mu^{0.4} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>1.078</td>
<td>0.382</td>
<td>0.336</td>
<td>0.895</td>
</tr>
<tr>
<td>500</td>
<td>1.195</td>
<td>0.338</td>
<td>0.273</td>
<td>0.942</td>
</tr>
<tr>
<td>600</td>
<td>1.51</td>
<td>0.293</td>
<td>0.209</td>
<td>1.056</td>
</tr>
</tbody>
</table>

VI-37
Fig. VI-13  Specific Heat of Liquid Water at 1500 psia
Fig. VI-14
Specific Heat of Steam as a Function of Temperature through Superheater
Fig. VI-15

Viscosity of Steam as a Function of Temperature at 1500 psia
Fig. VI-16  Thermal Conductivity of Steam at 1500 psia
Fig. VI-17 Temperature-Dependent Properties of Steam
The tube wall conductance is internally calculated from \( (k \pi DL/x) \), where \( k \) is the tube metal conductivity, \( D \) is the tube diameter, \( L \) the tube length, and \( x \) the wall thickness. The tube wall conductance is combined with the fluid film conductance according to the relation

\[
U_A \approx \frac{(hA)(k \pi DL/x)}{(hA + k \pi DL/x)}
\]

where \( U_A \) is the overall conductance between the fluid and the tube surface.

The two-phase boiling transfer coefficient is input equal to 1000 Btu/hr-ft\(^2\)-°F. The calculated results are insensitive to this parameter for a value greater than 750 Btu/hr-ft\(^2\)-°F. After computing all nodal temperatures, a computer subroutine is used to determine the natural circulation rate of liquid entering the vaporizer tube and volume fraction of liquid water in the two-phase fluid leaving the vaporizer tube. This subroutine essentially determines the fluid pressure drop due to velocity that will balance the liquid head available in the downcomer.

The pressure drops include expansion and contraction losses and mass acceleration terms, in addition to the straight pipe losses. Current results indicate that the natural circulation liquid velocity entering the vaporizer tube is between 1.5 and 2 fps and that the volume fraction of liquid in the two-phase mixture leaving the vaporization tube is between 0.65 and 0.75. This subroutine can also be used to determine these conditions for forced circulation.

The gas convection mechanism assumed for the analytical model is shown in Fig. VI-18. Mass interchange between the cavity and the surroundings is assumed by a unidirectional flow path that starts in the lower half of the aperture, flows along the floor, rises up the rear wall and then returns along the roof toward the upper half of the aperture. The fluid velocity is assumed to be 1 fps up the rear wall and mass continuity is assumed along the flow path. In addition, internal circulation is accounted for by assuming eddy convection between adjacent gas nodes.

The gas nodes are three-dimensional gas volumes associated with cavity wall nodes as shown in Fig. VI-19. A natural convection transfer coefficient of 1.0 Btu/hr-ft\(^2\)-°F is assumed for the path between the cavity wall and the gas. This mechanism predicts a natural convection loss from the cavity of about 2.3% of the incident solar energy, which compares favorably with the convection test data.
Legend:
1. One-Way Flow
2. Cavity Wall-Air Convection
3. Internal Eddy Convection

Fig. VI-18  Schematic of Cavity Convection Mechanisms
Because the cavity shape for the bench model is determined by the anticipated solar energy distribution in the full-scale unit, the solar distribution in the French CNRS facility must be modified by a reflector that directs the high-elevation-angle energy from the upper part of the CNRS parabolic reflector into the cavity aperture in a more horizontal direction. The reflector design is a cone frustum mounted in front of the cavity aperture as shown in Fig. VI-20.

Fig. VI-20 Reflector Concept for Bench Model

The solar flux distribution produced by both direct and reflected solar energy from the reflector in the bench model cavity is shown in Fig. VI-21 thru VI-25. In the bench model incident solar energy impinges on both the cavity roof and the floor.
Fig. VI-19  Cavity Convection Node Model
Fig. VI-21 Flux Pattern, Boiler Section 1, W/cm$^2$
Fig. VI-21  Flux Pattern, Boiler Section 1, W/cm²
Fig. VI-23  Flux Pattern, Boiler Section 2, W/cm²
Fig. VI-25 Flux Pattern, Cavity Lower Surface, W/cm²
The bench model energy distribution produces a peak flux of 4.38 W/cm² (13,900 Btu/hr-ft²) in the preheating section on the cavity floor, a peak flux of 20.0 W/cm² (63,416 Btu/hr-ft²) in the boiler section, and a peak flux of 25.6 W/cm² (81,331 Btu/hr-ft²) in the superheater section at the center of the rear wall and 17.5 W/cm² (55,500 Btu/hr-ft²) in the superheater at the outer edges. Where the solar flux level is too high in a relatively small area (in the range of 1 ft²), it can be reduced by mounting a radiation shield in front of the tubes. A small shield would not seriously modify the solar reflection and IR radiation distribution within the cavity. An approximately 1 ft² ceramic shield (uncooled) will be used in the center back of the superheater area to reduce the incident high flux (Fig. VI-21). Another possibility is to cover sections of heliostats to reduce high flux areas.

Based on the flux pattern the calculated superheater outlet temperature is 1040°F or 85°F above our desired temperature of 955°F. Because the superheater outlet temperature will be controlled by limiting the number of mirrors whose energy is incident on the superheater surface, it is very important that the predicted temperature be higher than the desired control temperature.

To reduce the superheater exit temperature, the solar energy impinging on the rear superheater surface shown in Fig. VI-21 was reduced and an analytical operational case rerun. The resulting incident and absorbed energies on each of the cavity surface nodes for this case are listed in Table VI-7 thru VI-9.

The computer model was used to predict the performance of the cavity for the above solar flux and fluid flow patterns. The resultant temperature profile along the flow path through the preheater is plotted in Fig. VI-26, the corresponding temperature profile through the superheater is plotted in Fig. VI-27. Feedwater is introduced into the top preheater at 400°F and the temperature of the water increases monotonically along the flow path to a bottom preheater exit temperature of 548°F. The maximum preheater tube wall temperature is 555°F. This temperature represents the "average" tube wall temperature and is less than the maximum local temperature at the point of peak incident solar flux.

Saturated steam enters the top superheater at a temperature of 600°F. The steam temperature increases monotonically along the flow path to an exit temperature of 982°F from the rear section.

The flow rate of liquid water through the preheater and the flow rate of steam through the superheater is equal to the steam production rate of 2546 lb/hr. The overall system energy balance for the case is given in Table VI-10.
### Table VI-7
Total Incident and Absorbed Solar Energy per Node for Superheater, kW

<table>
<thead>
<tr>
<th>Nodes</th>
<th>Incident</th>
<th>Absorbed</th>
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<tbody>
<tr>
<td>1</td>
<td>5.4705</td>
<td>5.2818</td>
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<tr>
<td>2</td>
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<td>3.8500</td>
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<td>4</td>
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<td>2.6877</td>
<td>2.7354</td>
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<td>6</td>
<td>7.8889</td>
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<td>14</td>
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<td>19</td>
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<td>4.1056</td>
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<td>20</td>
<td>3.0956</td>
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<td>21</td>
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<td>3.1648</td>
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<td></td>
<td><strong>129.4391</strong></td>
<td><strong>125.1611</strong></td>
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<tr>
<td>Subtotal</td>
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<td>55</td>
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<td>10.4285</td>
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<td>28.7718</td>
<td>28.3547</td>
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<td>63</td>
<td>6.3592</td>
<td>6.6711</td>
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<td><strong>126.4287</strong></td>
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<td></td>
<td><strong>254.6495</strong></td>
<td><strong>251.5898</strong></td>
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<tr>
<td>Superheater</td>
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<tr>
<td>89,90</td>
<td>0.7316, 0.7316</td>
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<tr>
<td>91,92</td>
<td>0.8019, 0.8540</td>
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<td>93,94</td>
<td>0.8809, 0.9504</td>
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<td>95,98</td>
<td>0.0473, 0.0508</td>
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<td>96,99</td>
<td>0.1768, 0.1727</td>
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<td>97,100</td>
<td>0.3197, 0.2574</td>
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<td>113,114</td>
<td>0.7318, 0.7318</td>
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<td>115,116</td>
<td>0.8019, 0.8542</td>
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<td>117,118</td>
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<td>119,122</td>
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<td>120,123</td>
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<td>121,124</td>
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<td>Bare Walls</td>
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### Table VI-8
Total Incident and Absorbed Solar Energy per Node for Boiler, kW

<table>
<thead>
<tr>
<th>Nodes</th>
<th>Incident</th>
<th>Absorbed</th>
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<td>25</td>
<td>15.313</td>
<td>14.2309</td>
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<td>26</td>
<td>15.683</td>
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<td>38</td>
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<td>45</td>
<td>8.806</td>
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<td></td>
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<tr>
<td></td>
<td><strong>291.474</strong></td>
<td><strong>275.0275</strong></td>
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<td>Subtotal</td>
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<td>Slant</td>
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<td>101,125</td>
<td>9.656</td>
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<td>102,126</td>
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<td>103,127</td>
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<td>105,129</td>
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<td>112,136</td>
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<td>Subtotal</td>
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<tr>
<td>Sides</td>
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<tr>
<td></td>
<td><strong>193.090</strong></td>
<td><strong>183.1597</strong></td>
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<td>Total Boiler</td>
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<tr>
<td></td>
<td><strong>484.564</strong></td>
<td><strong>458.1872</strong></td>
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The totals for Tables VI-7 thru VI-9 can then be tabulated as follows:

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<th>Support</th>
<th>Top</th>
<th>Support</th>
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<td>74.4652</td>
<td>63.208</td>
<td>69</td>
<td>72</td>
<td>74</td>
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<tr>
<td>2.1122</td>
<td>1.470</td>
<td>83</td>
<td>82</td>
<td>77</td>
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<td>8.9412</td>
<td>7.134</td>
<td>87</td>
<td>80</td>
<td>79</td>
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<tr>
<td>3.1042</td>
<td>2.887</td>
<td>88</td>
<td>86</td>
<td>75</td>
</tr>
<tr>
<td>2.4080</td>
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</tr>
<tr>
<td>0.6397</td>
<td>0.618</td>
<td>79</td>
<td>71</td>
<td>73</td>
</tr>
<tr>
<td>2.3533</td>
<td>1.599</td>
<td>78</td>
<td>72</td>
<td>72</td>
</tr>
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<td>6.6425</td>
<td>5.210</td>
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</tr>
<tr>
<td>2.5893</td>
<td>2.299</td>
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<td>71</td>
<td>71</td>
</tr>
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<td>3.9867</td>
<td>3.888</td>
<td>72</td>
<td>70</td>
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<td>4.2275</td>
<td>4.271</td>
<td>71</td>
<td>70</td>
<td>69</td>
</tr>
</tbody>
</table>

Absorbed Incident

Energy per Mode for Preheater, kW
Total Incident and Absorbed Solar

Table VI-9
Fig. VI-27  Superheater Temperature Profile
Fig. VI-26 Preheater Temperature Profile
Table VI-10  System Energy Balance

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<tr>
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<th>Btu/hr</th>
<th>Percent</th>
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<tr>
<td>IR Aperture Loss</td>
<td>37,729</td>
<td>1.31</td>
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<tr>
<td>Solar Aperture Loss</td>
<td>23,568</td>
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<tr>
<td>Convection Aperture Loss</td>
<td>73,223</td>
<td>2.54</td>
</tr>
<tr>
<td>Total Loss</td>
<td>134,520</td>
<td>4.67</td>
</tr>
<tr>
<td>Solar to Preheat</td>
<td>437,850</td>
<td>15.19</td>
</tr>
<tr>
<td>Solar to Vaporization</td>
<td>1,560,715</td>
<td>54.14</td>
</tr>
<tr>
<td>Solar to Superheat</td>
<td>749,835</td>
<td>26.00</td>
</tr>
<tr>
<td>Total Incident Solar</td>
<td>2,882,920</td>
<td>100.00</td>
</tr>
</tbody>
</table>

Cavity Convective Heat Loss Test Results

A preliminary cavity convective scale-model test program was completed during the program. This test program is described in detail in Appendix C. The results are summarized in Fig. VI-28, which shows the cavity convective loss compared with what would be expected without a cavity. The natural convection (wind speed = 0) loss is about the same for a cavity as for an open heat exchanger because the cavity area is much larger. However, as the wind speed increases, the convective loss for the cavity is significantly less than for an open heat exchanger. It should be understood that the results presented are for our conditions of heat flux, steam temperatures, and cavity configuration. Extrapolation to other sets of variables is difficult.
Fig. VI-28 Convection Loss vs Wind Speed
VII. Georgia Tech Subcontract
VII. SOLAR POWER SYSTEM AND COMPONENT RESEARCH

BOILER DESIGN AND BENCH MODEL DESIGN AND TEST PLAN

Final Report

February 10, 1974 through December 15, 1974

by

S. H. Bomar, Jr.
C. T. Brown
J. H. Murphy
J. D. Walton, Jr.

NSF Grant No. GI-41305
Martin Marietta Corporation Contract No. RC4-370450
Georgia Tech Project A-1604

Engineering Experiment Station
Georgia Institute of Technology
Atlanta, Georgia 30332
TABLE OF CONTENTS

A. INTRODUCTION

B. BOILER AND SUPERHEATER DESIGN AND ANALYSIS
   1. Stresses Due to Fluid Pressure and Heat Transfer Conditions
   2. Stresses and State of Stress at a Point
   3. Stress Limits
   4. Material Selection & Properties
   5. Stress Analysis
   6. Further Considerations of Thermal Stresses
   7. Reduction of Stresses
   8. Radiant Heat Transfer Conditions
   9. Stress Reactions Among System Components
  10. Safety Appliance Requirements for Bench Model Boiler

C. COLLABORATION WITH THE CNRS SOLAR ENERGY LABORATORY

D. MODEL OF CNRS FACILITY AND PROPOSED BENCH MODEL BOILER/SUPERHEATER

E. TEST PLAN FOR BENCH MODEL SYSTEM
   1. Flux Redirector
   2. Bench Model Boiler/Superheater Assembly

F. WATER TECHNOLOGY FOR BENCH MODEL BOILER

G. PROPOSED INSTRUMENTATION FOR BENCH MODEL BOILER

REFERENCES
TABLE OF CONTENTS (Continued)

APPENDIX A. REQUIRED GAUGES, INSTRUMENTS AND SAFETY EQUIPMENT FOR SOLAR BOILER

APPENDIX B. PROGRAM MEETINGS & CONTACTS
A. INTRODUCTION

Georgia Tech's areas of primary responsibility during this phase of the current program were the thermal stress analysis of the central receiver boiler and superheater tubes, the ASME Boiler Code and practice adherence analysis, the test installation engineering, and the test plan generation for the bench model boiler to be operated in the CNRS Solar Furnace in southern France. Boiler feedwater technology was included in the test installation engineering task.

The Martin Marietta/Georgia Tech cavity type bench model boiler is designed to operate at a power level of 1000 kWth. Conceptually, this central receiver will consist of banks of fluid filled tubes lining the walls, floor and ceiling of the cavity. Arrangements of these tubes will serve as the preheater, boiler and superheater sections of a solar powered boiler. In this design as in most solar thermal conversion designs the radiant energy impinging on these tubes will not be symmetric about the axes of these tubes. The result will be asymmetric heating of the tubes, and in some cases the creation of quite large thermal stresses in these tubes.

Considering the area of thermal stresses due to asymmetric heating to be of extreme importance to the Solar Thermal Conversion program in general and to the Martin Marietta/Georgia Tech effort in particular, Georgia Tech undertook a program to analyze the thermal stresses that would develop in asymmetrically heated tubes. These theoretical studies were conducted as a function of the incident heat flux, the distribution of that heat flux, the properties of the fluid within the tube, the heat transfer conditions, and
the physical properties of the tubes under irradiation. The results of these studies have yielded candidate boiler and superheater tubes for the Martin Marietta/Georgia Tech bench model and equally important have indicated the magnitude of the stress problem associated with the asymmetric heating of boiler/superheater tubes. A summary of this work is presented in Section B of this report.

A second major area of Georgia Tech responsibility in the current program is the ASME Boiler and Pressure Vessel Code and practice adherence analysis. The American Society of Mechanical Engineers set up a committee in 1911 for the purpose of formulating standard rules for the construction of steam boilers and other pressure vessels. This committee is now called the Boiler and Pressure Vessel Committee and its set of established rules of safety, consisting of eleven sections in fourteen volumes, is called the ASME Boiler and Pressure Vessel Code. Addenda which includes additions and revisions to the Code are published twice a year and the Code is completely revised and reissued every three years. The purpose of the Code is to set forth established rules of safety governing the design, fabrication, and inspection during construction of boilers and unfired pressure vessels. The objective of these rules is to afford reasonably certain protection of life and property and to provide a margin for deterioration in service so as to give a reasonably long safe period of usefulness.

Every effort has been made to design the Martin Marietta/Georgia Tech one megawatt bench model boiler in accordance with the ASME Boiler Code and with good design practice. Toward that goal there has been an ongoing safety analysis program paralleling the design phase of the boiler. In addition
to the thermal stress analysis program mentioned earlier, work has been done by Georgia Tech in the areas of safety appliances, steam drum sizing, steam drum fittings and in other areas of overall boiler design. Babcock and Wilcox Company, consultant to Georgia Tech in the area of good design practice and adherence to the Code, participated in this effort and added substantial technical expertise in this area. Much of this work has been of a routine nature and will not be reported here. The noteworthy results of this analysis also appear in Section B of this report.

It is anticipated that the Martin Marietta/Georgia Tech bench model boiler, which has been designed and engineered during the current phase of this program, will be constructed in this country and tested in the 1000 kW French CNRS Solar Furnace during the next work phase. Toward that goal a significant effort has been expended correlating the design of the boiler with the capabilities and facilities of the CNRS Solar Laboratory.

Due to the shear size of the cavity boiler it has been necessary to prepare "as built" drawings of the CNRS focal building and the adjacent laboratory facilities to insure that the boiler can be moved into the focal area and mate with existing equipment and facilities. Reduced-scale copies of these drawings and a photographic description of the path the boiler must take through the facility on its way to the focal room are presented in Section C of this report. The usefulness of these data has been greatly enhanced by the availability of scaled models of the CNRS focal building and the cavity boiler and its accessory equipment. These models were constructed at Georgia Tech. Photographs of these models appear in Section D.
A test plan for operation of the bench model boiler at the CNRS facility has been prepared and appears as Section E. The purpose of this test plan is to provide general guidelines for the operation and testing of the boiler. Included in the plan are procedures for the initial startup of the boiler and its system, the steady state and transient operation of the boiler, and an outline of experimentation and testing to be conducted at the CNRS facility.

This test plan is intended only as a guide to the operation and testing of the boiler, and as such, is intended to be flexible. In the testing of any new complex system surprises are to be expected. Therefore, many decisions concerning the day-to-day operation of the boiler must of necessity be made in the field. Likewise, the exact experimental plan to be followed on a given day may be formulated only after a detailed analysis of the previous day's work has been completed. This test plan does, however, provide a framework of required tasks to insure that the objectives of the test program will be satisfied.

In addition to the actual testing of the boiler, the test plan calls for the design, construction, checkout and characterization of a flux redirector for the CNRS Solar Furnace. Because the cavity shape for the bench model is determined by the anticipated solar energy distribution in the full scale unit, the solar distribution in the French CNRS facility must be modified by a reflector which directs the high elevation-angle energy from the upper part of the CNRS parabolic reflector into the cavity aperture in a more horizontal direction. Design, construction, and characterization of the redirector will be the responsibility of Georgia
Tech. The major portion of this activity will occur during the next program phase. The current reflector design is a cone frustum mounted in front of the cavity aperture.

Section F of this report concerns water technology for the bench model boiler. In the field of high pressure boilers such as the bench model, there cannot be any compromise in water quality. The material in Section F summarizes a procedure for the initial cleaning of the new boiler, outlines the requirements for a boiler feedwater system, and summarizes the procedures for maintaining high quality water in the boiler.

An electronic control and data collection system for the bench model boiler is proposed in Section G of this report. This system, based on the use of a small digital computer, would allow for the automatic monitoring and control of the boiler and would simultaneously allow for the tape recording of 96 channels of raw and processed data. All components of the data collection system are standard off-the-shelf items. No electronic design or buildup would be required. Since the heart of the proposed system would be a small general purpose computer, rapid, on-site data analysis would be possible.
Asymmetrical heating of tubes and the cyclic nature of anticipated operations impose severe stresses on components of a solar heated boiler. The designs used in fuel-fired boilers can overcome these stress problems to a large extent by arranging for critical tubes to be heated from all sides and by long term operation between shutdowns. Professor Francia, at the University of Genoa, has selected a solar heated boiler design which minimizes asymmetrical heating by employing a boiler facing downwards over a field of collector mirrors. Energy is received into the boiler through a wide viewing angle and the tubes, particularly in the boiling section, are rather uniformly illuminated throughout their circumference. In the very large systems envisioned on this program, prohibitive tower heights would be required to support the boiler assembly above the mirror field. Martin Marietta Corporation has chosen a boiler geometry in which the receiving aperture faces slightly below the horizontal, and the incoming concentrated solar radiation is received predominantly on one side of the tubes. This concept appears to give the most acceptable balance among the many factors that must be considered in a cavity shaped boiler. Thus, asymmetrical heating of the tubes must be accepted and accommodated in the boiler design. Because of the high temperatures involved and the cyclic operation of a solar boiler, the structural aspects of the design are very critical.

As part of its contract commitment to the Martin Marietta/Georgia Tech Solar Thermal Conversion effort, Georgia Tech undertook a major program to determine and analyze the thermal stresses associated with asymmetrically heated boiler and superheater tubes. The objective of this program was to provide the Martin Marietta Corporation with thermal stress design data for
the boiler and superheater tubes to be used in the one megawatt bench model boiler. This study, along with the engineering design of the bench model has been completed. The accomplishments of this study included the following:

1. The development of several heat transfer/thermal stress analysis computer codes for the calculation of thermal stresses in asymmetrically heated tubes;
2. The use of these codes to evaluate numerous tube types and sizes, and finally, to specify in detail the tubes to be used in the bench model boiler;
3. The development of a greater understanding of the problems created by the asymmetric heating of boiler tubes; and finally, (4) The exploration and development of techniques for the reduction of such stresses.

A summary of the results of this study appears in the following paragraphs. Thermal stresses, properties of stresses and stress limits are discussed in introductory remarks. The heat transfer and thermal stress analysis computer codes developed during the course of this study are described. The procedures and allowable values of stress dictated by the ASME Boiler and Pressure Vessel Code for the selection of boiler tubing are discussed in some detail. Using the computer codes and complying with the requirements of the ASME Boiler Code, candidate tubing for the bench model boiler is specified. Considering the magnitude that thermal stresses can achieve in asymmetrically heated tubes, some preliminary stress reduction studies were undertaken during this program phase. The results of these studies also appear in this section. Finally, the minimum safety equipment requirements for the bench model are presented.

1. Stresses Due to Fluid Flow and Heat Transfer Conditions

   Procedures have been established during this program phase for the
calculation of stresses occurring in round tubes containing fluid and heated by arbitrary distributions of heat flux on the outside tube surface. A brief introduction to the possible stresses and their relative importance is appropriate here to aid in understanding the accompanying data. Figure 1 illustrates the stresses which can act on a differential volume of the metal within a tube wall.

There are two types of stresses, normal and shear denoted respectively by the symbols \( \sigma \) and \( \tau \). To denote the direction of the plane on which the stress is acting, subscripts to these letters are used. In cylindrical coordinates, \( \sigma_r \) denotes a radial stress, acting on a plane normal to the radius of the tube; similarly, \( \sigma_\phi \) and \( \sigma_z \) denote tangential and axial stresses, acting on planes normal to the tube tangent and axis. Normal stresses are considered positive when they act in a direction to cause tension within the differential volume element, and negative when they cause compression within the element. Shear stresses require two subscripts for identification, the first denoting the normal to the plane under consideration and the second indicating the direction of the component of the stress. For example, \( \tau_{\phi,r} \) acts in the plane normal to the \( \phi \) direction, and the shear stress itself acts in the \( r \) direction. A convention exists for identifying positive and negative shear stresses. In the limit where the dimensions of the volume element become vanishingly small, its shape will approach the shape of a cube and the stresses are assumed to act on mutually perpendicular sides.

A comprehensive discussion of stress theory is beyond the scope of this report, but several properties of stresses are of interest in the present
Fig. 1
Stresses Acting on a Differential Volume of a Tube
discussion 3/: (1) It is always possible to define mutually perpendicular reference axes in such a manner that the shear stresses at a point vanish. Then only normal stresses will be acting at that point; these normal stresses are known as the principal stresses and their directions the principal axes. (2) It can be shown that the maximum stress at any point is the largest of the three principal stresses at this point. (3) For a uniformly heated tube the shear stresses are zero because of symmetry considerations; this means that the normal stresses $\sigma_r$, $\sigma_\phi$, and $\sigma_z$ are the principal stresses. In a tube which is heated asymmetrically but whose temperature distribution does not vary with length, shear stresses acting in planes perpendicular to the tube axis are zero; this means that $\sigma_z$ is a principal stress. The implications of these properties of stresses will be indicated as the results of stress analyses are described.

The method for calculating stresses within tube walls involves computation of the temperature distribution within the wall followed by computation of the resulting radial, tangential, axial and shear stresses. The algorithms necessary to calculate these temperatures and stresses have been programmed in FORTRAN IV and are operational as two separate programs on the Georgia Tech UNIVAC 1108 computer. The temperature distribution is calculated by a numerical relaxation method 4/, assuming two-dimensional steady state conduction in a diametral plane of the tube. The tube is partitioned as shown in Figure 2, and because of symmetry about the axis of incoming radiation the calculation need be performed on only one side of the tube. An arbitrary heat flux can be assigned to the outer surface of each circumferential zone; the tube is assumed to contain a fluid receiving
Fig. 2
Partitioning of Sample for Thermal Analysis Program
heat from the inside surface of the tube. The input data are: (1) the heat flux to the outside surface in each circumferential segment, (2) the thermal conductivity of the metal tube, (3) the convection coefficient of the fluid, (4) the temperature of the fluid, and (5) the inside and outside diameters of the tube. The computer program determines the temperature at each node by averaging the temperatures of the surrounding nodes. The heat conduction equations in the program are designed to accommodate the cylindrical geometry of the system and special heat balances are required at the tube surfaces. This program generates an array of 336 temperatures, representing the distribution in the wall.

The temperatures at the node points are then used as input data to the second program which calculates the three normal stresses and the shear stress $\tau_r, \phi$ at each point. The stress program is based primarily on a paper published by investigators at The Babcock and Wilcox Company. The temperature distribution is divided into a constant component, a radial component, and an asymmetric component. The constant temperature component makes no contribution to the stresses. The radial and asymmetric temperature components each contribute to the stresses, and individual contributions are calculated and summed to obtain the total $\sigma_r$, $\sigma_\phi$, $\sigma_z$, and $\tau_r, \phi$ stresses at each of the 336 node points.

The ASME Boiler and Pressure Vessel Code establishes maximum allowable working stresses which are dependent on material properties and operating temperature. These stresses have been chosen on the basis of experience and test data for the approved varieties of steel used in boilers; for example, in a typical case the maximum allowable working stress is two-thirds of the
yield stress of the metal at the operating temperature. The Code then
prescribes formulas for calculating wall thickness, based on pressure,
dimensions and stress. These formulas incorporate safety factors for tubes
fired on all sides, but do not account for asymmetrical heating. It was
therefore decided that all calculations related to stress should be based on
the computer program rather than Code formulas, and terms were added to the
stress program to account for fluid pressure.

Both the thermal and stress programs were carefully checked for
convergence (approach to the true solution) by calculating cases for which
analytical solutions had been published. It was determined that eight angular
and six radial partitions were sufficient to achieve convergence in the
thermal program. However, 16 angular and 21 radial partitions were required
to achieve convergence in the stress program, so this partitioning scheme
was adopted. It was of course necessary to use the same number of partitions
in both programs. After the validity of the programs had been assured,
temperature and stress distributions were calculated for the preheater,
boiler and superheater sections of the system for a number of tube sizes,
tube construction materials, and boiler operating conditions.

For purposes of illustration only, the results of one such set of
calculations are shown in Table I. These calculations were made assuming
the tubes to be of 304H stainless steel. All runs listed correspond to a
maximum heat flux of 50,000 Btu/hr-ft² with a cosine flux distribution
varying from maximum at the front of the tube to zero at the side. All
tubes are assumed to be mounted in contact with adjacent tubes, so that the
heat flux in the rear quadrant is zero. Fluid conditions in the preheater
<table>
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<th>Run No. and Conditions</th>
<th>Maximum Stresses (psi)</th>
<th>Maximum Temperature of Tube(°F)</th>
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<tr>
<td></td>
<td>Longitudinal</td>
<td>Radial</td>
</tr>
<tr>
<td>Run 39A-Preheater inlet, ( \frac{1}{4} )&quot; sch. 40 pipe (0.540&quot; O.D. 0.364&quot; I.D.) TF= 447° F; PF = 1550 psig; ( h = 1000 \text{ Btu/hr-ft-}°\text{F} )</td>
<td>16,750</td>
<td>1550</td>
</tr>
<tr>
<td></td>
<td>(ASYM-14,700)</td>
<td>(Fluid-1550)</td>
</tr>
<tr>
<td>Run 39B-Preheater outlet, ( \frac{1}{4} )&quot; sch. 40 pipe (0.540&quot; O.D. 0.364&quot; I.D.), TF= 550° F; PF = 1540 psig; ( h = 1000 \text{ Btu/hr-ft-}°\text{F} )</td>
<td>16,750</td>
<td>1540</td>
</tr>
<tr>
<td></td>
<td>(ASYM-14,700)</td>
<td>(Fluid-1540)</td>
</tr>
<tr>
<td>Run 40A-Boiler, ( \frac{3}{4} )&quot; sch. 40 pipe (1.050&quot; O.D. 0.824&quot; I.D.) TF = 600° F; PF = 1540 psig; ( h = 10,000 \text{ Btu/hr-ft-}°\text{F} )</td>
<td>12,225</td>
<td>1540</td>
</tr>
<tr>
<td></td>
<td>(ASYM-9330)</td>
<td>(Fluid-1540)</td>
</tr>
<tr>
<td>Run 40B-Superheater inlet (0.75&quot; O.D., 0.50&quot; I.D.) TF = 600° F; PF = 1540 psig; ( h = 360 \text{ Btu/hr-ft-}°\text{F} )</td>
<td>31,200</td>
<td>1540</td>
</tr>
<tr>
<td></td>
<td>(ASYM-28,300)</td>
<td>(Fluid-1540)</td>
</tr>
<tr>
<td>Run 40C-Superheater outlet (0.75&quot; O.D., 0.50&quot; I.D.) TF = 955° F; PF = 1300 psig; ( h = 360 \text{ Btu/hr-ft-}°\text{F} )</td>
<td>31,200</td>
<td>1300</td>
</tr>
<tr>
<td></td>
<td>(ASYM-28,300)</td>
<td>(Fluid-1300)</td>
</tr>
</tbody>
</table>
and superheater vary substantially from inlet to outlet, so that results for both inlet and outlet conditions are given. Stress values shown in parentheses are the major contributors to the accompanying maximum total stresses.

In reviewing Table I, the properties of stresses mentioned earlier should be recalled. In all the stress calculations performed so far, shear stress has been quite small, always less than 200 psi. Thus, the calculated radial, tangential and longitudinal values correspond very closely to the principal stresses; it follows that the largest of these three stresses is the maximum stress that can exist at the point in question.

In each case shown in Table I, the highest stress is longitudinal (axial) and the predominant contributor is the asymmetrical temperature distribution. Maximum allowable stresses in the preheater and boiler are given at 700 °F, in accordance with the ASME Code specification that any heated pressure vessel shall be assumed to be operating at not less than that temperature.

Figures 3-5 show temperature and stress contour plots for the superheater inlet example given in Table I. The temperature distribution is given first, and illustrates the fact that little heat is received by the fluid in the rear quadrant of the tube because the temperature gradient there is low. Next, the shear stress is shown and is seen to be quite low in comparison to other stresses. Shear stresses in a tube tend to cause the tube cross section to assume a shape other than circular; for example, elliptical, triangular, square, etc., depending on the shear stress pattern. The low values found for shear stress support the assumption that the normal stresses
Temperature Contours - °F
Q = 50,000 Btu/hr-ft$^2$
Outside Diameter - 0.75 inch
Wall Thickness - 0.125 inch

Shear Stress Contours ($\tau_{r,z}$) - psi

Fig. 3
Temperature and Shear Stress Contours for Typical Superheater Tube
Fig. 4
Radial Stress and Tangential Stress Contours for Typical Superheater Tube
Fig. 5
Longitudinal Stress Contours from Asymmetric Temperature Distribution - psi

Longitudinal Stress Contours - psi

Longitudinal Stress Contours for Typical Superheater Tube
σ_r, σ_φ, and σ_z closely approximate the principal stresses, and that the largest normal stress represents the maximum stress at any point.

Radial and tangential stress contour lines for the same tube are shown in Figure 4. These contours are roughly concentric with the tube wall, indicating that the asymmetrical temperature distribution is of minor importance in these cases; fluid pressure is a major contributor to σ_r and σ_φ.

Contour lines representing total longitudinal stress and the asymmetric temperature contribution to longitudinal stress are shown in Figure 5. These lines show steep gradients of stress around the tube; the major contribution of the asymmetric temperature gradient to the longitudinal stress is also evident.

2. Stresses and State of Stress at a Point

Normal, shear and principal stresses were defined in the previous discussion. The convention used in identifying stresses was also explained. The properties of stresses which are of interest in the present discussion are the following:

(1) The magnitude and character of the stresses at a point depend on the orientation of three mutually perpendicular stress planes. It is always possible to find mutually perpendicular reference axes such that the corresponding planes have no shear stress acting on them. Then only normal stresses will be acting; these normal stresses are known as the principal stresses and their directions the principal axes;
(2) One of the principal stresses will be the algebraic maximum normal stress at the point and one will be the algebraic minimum normal stress at the point;

(3) The maximum shear stress at the point is one-half the difference between the maximum and minimum principal stresses; and finally,

(4) Any plane which is oriented symmetrically with respect to the geometry and loading of the material will contain no shear stress and therefore the normal stress acting on this plane will be one of the principal stresses.

3. Stress Limits

To assess the significance of the stresses and their relationship to structural integrity, the procedures and allowable values of stress as given by the ASME Boiler Code have been used 6,7/. The allowable value of stresses at a point depends on the value of the stress intensity at that point. Stress intensity is the difference between the maximum and minimum principal stresses, which from the previous discussion, is seen to be equal to twice the maximum shear stress at the point. Stress intensity is limited to some fraction of the yield strength (usually $\frac{2}{3}$) of the material at the operating temperature. This value, denoted by $S_m$, is given in the Boiler Code for various materials and temperatures. For cyclic variation of the stresses consideration must be given to fatigue. The limiting value of stress intensity variation for a given number of cycles is governed by the fatigue strength of the material, $S_a$. 
In applying the procedures of the ASME Boiler Code, stresses are given various classifications. Two general categories of stresses are defined: primary stresses and secondary stresses. Primary stresses are those which are developed by the imposed loading on the structure. Imposed loadings consist of the internal pressure in tubes, pipes and shells and dead weight loads. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses which considerably exceed the yield strength of the material will result in gross distortions and possibly material failure. Secondary stresses are those which are developed by the constraint of adjacent parts or by self-constraint of the structure. Secondary stresses are self-limiting in that local yielding and minor distortions will elevate the condition causing the stresses. Thermal stresses resulting from constrained thermal expansion are classified as secondary stresses.

In addition to these two general classifications of stresses, further identification of stress types are membrane stresses, gross structural discontinuity stresses and local structural discontinuity stresses. Membrane stress is the value of normal stress averaged across the thickness of the section under consideration. Gross structural discontinuity stresses are associated with a stress intensification affecting a relatively large portion of the structure; for example, head-to-shell and similar junctions. Local structural discontinuity stresses are associated with a stress intensification affecting a small volume of the structure; for example, notches, fillets and partial penetration welds. The following summarizes the limits on stress intensities imposed by the code 7/:

VII-24
a. General primary membrane stress intensity \( (P_m) \) is limited to \( S_m \).

b. Primary membrane stress intensity including the effect of gross structural discontinuity \( (P_L) \) is limited to \( 1.5 S_m \).

c. Primary stress intensity at a point including effects of gross structural discontinuity and bending \( (P_L + P_b) \) is limited to \( 1.5 S_m \).

d. Primary plus secondary \( (Q) \) stress intensity including all effects of gross structural discontinuity and bending \( (P_L + P_b + Q) \) is limited to \( 3 S_m \).

e. Peak stress intensity, primary plus secondary stresses including all effects of gross and local structural discontinuities is limited under cyclic conditions by fatigue considerations. The amplitude of variation of the peak stress intensity is limited to the value of \( S_a \) corresponding to the expected number of cycles.

f. Under cyclic conditions the maximum thermal stress permitted in a shell loaded by steady internal pressure is limited by thermal ratchet considerations. Thermal ratchet is a distortion which progressively increases with each thermal cycle.
4. Material Selection and Properties

Materials selected for the various boiler components have been chosen from the list of approved materials given in the ASME Boiler Code. Preheater and boiler sections operate within the temperature capabilities of carbon steels and so this material will be used for these components. The superheater operates at higher temperatures and it is necessary to use a high alloy austenitic steel in this section. Grade 321 stabilized stainless steel has been selected for the superheater.

In the heat transfer, temperature and stress predictions, the variation of material properties with temperature has been considered. All analyses have been performed on a constant property basis, but the properties used are those existing at the average temperature of the component in question.

A thorough search was conducted in order to find the best data available for these thermal and mechanical properties. Property variation with temperature for ferritic and austenitic stainless steels are shown in Figures 6-11.

5. Stress Analysis

Stresses in the various components of the boiler arise from several causes. It is convenient in evaluating stresses and their significance to follow the classification of stresses as given in the ASME Boiler Code.

It is recalled that the two main categories of stress are primary and secondary. Primary stresses are those resulting from the internal pressure within tubes and shells and deadweight loads acting on the structure. Secondary stresses are the thermal stresses. Thermal stresses
Fig. 6 Modulus of Elasticity of Stainless Steel
Fig. 7 Modulus of Elasticity of Low Chrome Steel
Fig. 8 Modulus of Elasticity of Carbon Steels
Stainless Steel
Carbon & Low Chrome Steel

Fig. 9 Coefficient of Thermal Expansion for Steels
Fig. 10 Thermal Conductivity of Carbon and Low Chrome Steels
**Thermophysical Properties of Materials**

**ASME Boiler Code**

---

**Fig. 11 Thermal Conductivity of 304 Stainless Steel**

![Thermal Conductivity Graph](image)

- **Thermophysical Properties of Materials**
- **Industrial Source**
- **ASME Boiler Code**

---

*Fig. 11 Thermal Conductivity of 304 Stainless Steel*
exist within a component because of the temperature gradient in the component. Additional thermal stresses are due to the constraint against thermal expansion offered by the adjacent structure to which the component is connected.

Stress calculations have been carried out for many tube sizes, tube materials, and operating conditions in an attempt to establish candidates for the various heat receiving tubes of the bench model boiler. The majority of these calculations are now obsolete as a result of the steady evolution of the boiler to its final design. In the material that follows, only the most recent stress calculations are presented and discussed.

a. Primary Stress Due to Internal Fluid Pressure. Because of complete symmetry, there will be no shear stress acting in either the radial, tangential or axial directions and, therefore, these will be the principal directions and the corresponding normal stresses the principal stresses.

The method of stress calculation is based on the standard solution for stresses in a thick wall tube subject to internal pressure 8/. It can be shown that the maximum stress intensity occurs at the inside surface of the tube and is the difference between the tangential and radial normal stresses at that point. The value of this stress intensity must be limited to \( S_m \) as specified in condition (a) of Section B.3. Results obtained for preheater, boiler and superheater tubes both at inlet and outlet conditions are shown in Tables II through V. It is seen that for the cases considered all tubes are within the allowed limits for primary stress.
Table II Results of Stress Calculations - Preheater Tubes

HEAT FLUX: 27,580 Btu/ft\(^2\)-hr.  
MATERIAL: Carbon Steel (O.D. = 0.75", I.D. = 0.55")

<table>
<thead>
<tr>
<th>Operating Conditions</th>
<th>Tube Temperature (°F)</th>
<th>Primary Stresses (psi)</th>
<th>Total Stresses (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Mean</td>
<td>Maximum</td>
</tr>
</tbody>
</table>
| Inlet
TF = 4290 °F  
PF = 1500 psig | 475     | 443  | 6490     | 11500 (@ 700° F) | 8200     | 34500 (@ 700° F) |
| Outlet
TF = 5920 °F  
PF = 1500 psig | 639     | 606  | 6490     | 11500 (@ 700° F) | 8620     | 34500 (@ 700° F) |

Table III Results of Stress Calculations - Boiler Tubes

HEAT FLUX: 63,400 Btu/ft\(^2\)-hr.  
MATERIAL: Carbon Steel (O.D. = 1.00", I.D. = 0.80")

<table>
<thead>
<tr>
<th>Operating Conditions</th>
<th>Tube Temperature (°F)</th>
<th>Primary Stresses (psi)</th>
<th>Total Stresses (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Mean</td>
<td>Maximum</td>
</tr>
</tbody>
</table>
| TF = 5920 °F  
PF = 1450 psig | 677     | 617  | 8055     | 11500 (@ 700° F) | 14650     | 34500 (@ 700° F) |
Table IV  Results of Stress Calculations—Superheater Inlet Conditions

**HEAT FLUX:** 47,500 Btu/ft$^2$-hr  
**FLUID TEMPERATURE:** 592° F  
**FLUID PRESSURE:** 1450 psig

<table>
<thead>
<tr>
<th>Tube Specification</th>
<th>Tube Temperature (°F)</th>
<th>Primary Stresses (psi)</th>
<th>Total Stresses (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Mean</td>
<td>Maximum</td>
</tr>
<tr>
<td>Croloy 2¼</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td>720</td>
<td>647</td>
<td>5220</td>
</tr>
<tr>
<td>I.D. = 0.50&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TP 321*</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td>754</td>
<td>646</td>
<td>7608</td>
</tr>
<tr>
<td>I.D. = 0.59&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TP 304*</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td>763</td>
<td>656</td>
<td>5220</td>
</tr>
<tr>
<td>I.D. = 0.50&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*The carbon content of these stainless steels must be greater than 0.04 percent for these allowable stress values to be applicable.
Table V Results of Stress Calculations—Superheater Outlet Conditions

HEAT FLUX: 47,500 Btu/ft$^2$-hr
FLUID TEMPERATURE: 955°F
FLUID PRESSURE: 1300 psig

<table>
<thead>
<tr>
<th>Test Specification</th>
<th>Tube Temperature (°F)</th>
<th>Primary Stresses (psi)</th>
<th>Total Stresses (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Mean</td>
<td>Maximum</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Croloy 2½</td>
<td>1088</td>
<td>1011</td>
<td>4680</td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I.D. = 0.50&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TP 321*</td>
<td>1112</td>
<td>1007</td>
<td>6821</td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I.D. = 0.59&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TP 304*</td>
<td>1117</td>
<td>1017</td>
<td>4680</td>
</tr>
<tr>
<td>O.D. = 0.75&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I.D. = 0.50&quot;</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*The carbon content of these stainless steels must be greater than 0.04 percent for these allowable stress values to be applicable.
b. Primary Fluid Pressure Stresses at Points of Gross Structural Discontinuity. No basic calculations have been made to predict the stresses existing at the tube and pipe joints associated with the internal pressure loading. Rather, the rules of Section VIII, Division 2, of the Boiler Code have been used to assess the suitability of these joints. These rules prescribe accepted methods and the amount of metal required for reinforcement around such connections. The detail design of all such joints is not complete and, therefore, a complete analysis has not been conducted. It appears from the calculations done, that it will be possible to satisfy the Code for these connections with respect to the primary stress requirements.

c. Primary Stresses Due to Dead Weight Loading. Dead weight loads consist of metal and water weights which must be carried by component cross-sections or by joints. In general these loads would be expected to be small. However, there are some critical components within the boiler where it is believed that this loading should be evaluated.

The condition which has already been considered is the bending of preheater tubes lying in the upper and lower panels due to metal and water weight. These stresses have been found to be significant, but well within allowable limits.

d. Secondary Thermal Stresses. Stresses existing in the tubes because of temperature gradients and constrained distortions of the tubes have been evaluated. The method for calculating stresses within tube walls involves computation of the temperature distribution within the wall followed by computation of the resulting radial, tangential, axial and shear stresses. The algorithms necessary to calculate these temperatures and
stresses have been programmed in FORTRAN IV and are operational as two separate programs on the Georgia Tech UNIVAC 1108 computer.

The procedures used for temperature and stress predictions were described in Section B.1. It is recalled that in the analysis of thermal stresses in the tube, the tube has been assumed free to expand axially but is constrained against any bending.

Temperature and stress distributions were calculated for the preheater, boiler and superheater sections of the system. Because of the asymmetrical temperature distribution, shear stresses exist at points within the tube wall. However, these shear stresses were found to be quite small in comparison to the normal stresses and therefore the radial, tangential and axial normal stresses correspond very closely to the principal stresses. The largest numerical value of principal stress was always found to be a large compressive axial stress existing at the outer surface on the heated side of the tube. This stress is due primarily to the restraint of the tube against bending.

In assessing the significance of thermal stresses, the requirements of the Boiler Code with respect to the allowable value of total primary plus secondary stress intensity was considered. The maximum stress intensity was always found to occur at the outside surface on the heated side of the tube. At this point the maximum stress intensity is equal to the difference between the axial thermal compressive stress and the tangential tensile stress due to internal fluid pressure. As shown in condition (d) of Section B.3, this value must be limited to $3 S_m$. 

VII-38
Comparisons of allowable and calculated total stress for the final design operating conditions are shown in Tables II through V for preheater, boiler and superheater tubes of the bench model. Carbon steel was chosen for the preheater and boiler sections. Superheater stress calculations were made assuming TP304 and TP321 stainless steels and Croloy 2½ low chrome steel.

With the exception of the superheater outlet operating conditions all calculated total stresses were significantly less than the associated allowable stress. Such was not the case for the superheater outlet assuming Croloy 2½ or TP321 tubing. For Croloy 2½ the calculated maximum stress was approximately seven percent above the allowable total stress. For TP321 stainless steel the calculated maximum stress was approximately one percent below the allowable total stress. In each of the above cases there is some safety factor hidden in the fact that the allowable stress was taken at a mean tube wall temperature of 40 to 45 degrees above the calculated mean wall temperature.

The TP304 stainless steel tube appears to be a safe design, but this is an unstabilized alloy which is subject to rapid corrosion in high pressure, high temperature steam service. The TP321 tubing is stabilized with a small amount of titanium, is therefore preferred for superheater service and has been selected for the superheating tubing.

The above superheater data were calculated for the maximum heat flux expected in the superheater section of the bench model boiler; i.e., 47,500 Btu/ft²-hr. Additional thermal stress data have been calculated for incident heat fluxes ranging from 40,000 to 75,000 Btu/ft²-hr assuming TP321 and TP321H steels and superheater outlet operating conditions. The TP321H
is a high carbon version of the TP321 and the calculated maximum total stress at a given heat flux is essentially the same for the two steels.

A plot of maximum calculated total stress versus incident heat flux for the two steels appears as Figure 12. Mean tube wall temperatures were also calculated for each heat flux. The allowable total stress as defined by the ASME Boiler and Pressure Vessel Code is a function of the tube construction material and the mean tube temperature. Allowable total stresses for both steels were determined as a function of temperature, and thus heat flux, from Section I tables of the Code. These data are also plotted as part of Figure 12. From these data it appears that the TP321 steel can tolerate a maximum asymmetric heat flux of approximately 61,000 Btu/ft²-hr, and the TP321H steel can tolerate a maximum asymmetric heat flux of approximately 67,500 Btu/ft²-hr.

6. Further Considerations of Thermal Stresses

As previously indicated, the critical thermal stress in the tube is the axial compressive stress arising from the constraint of the tube against bending. The solution used in predicting these stresses assumes that the constraint is accomplished by a uniform moment applied at each end of the tube. In actual fact, the tubes will be supported by buckstays which apply a concentrated load at their points of attachment. The buckstays will allow axial motion but will restrict transverse motion, and therefore bending of the tubes. Because of the different moment distribution introduced in the tubes from that assumed in the computer solution, it was decided to give further consideration to this method of constraint.
Fig. 12
Maximum Allowable Total Stress and Maximum Calculated Total Stress for TP321 and TP321H Stainless Steels as a Function of Incident Heat Flux. Superheater Outlet Operating Conditions
Two methods of applying buckstays have been considered. In each case the tube is assumed to be rigidly fixed at one end. In the first case one buckstay is attached at the free end; in the second case two buckstays are used, one at the free end and one at the midpoint. The configurations and the corresponding moment distribution introduced in the tubes are illustrated in Figures 13 and 14. It has been found that the value of the moment introduced into the tube by the use of buckstays might actually exceed the fixed end moment unless the buckstays allow some transverse movement. Since a higher moment will cause higher axial stress than those predicted, it will be necessary to allow some movement of the tube, particularly in the superheater section.

Figure 15 shows the results of an analysis made to establish the relationship between the maximum moment in the tube and any allowed deflection at the buckstay points. Both the one buckstay and two buckstay cases are shown. The "free" deflection is the deflection the end of the tube would have if no buckstays were used and the tube was supported only at its upper end as a cantilever. As an example, the superheater tube at inlet conditions is shown in Figure 16. For the predicted temperature distribution in this tube, the end moments necessary to completely prevent bending (from the computer program) was found to be 564 in-lbs. The axial compressive stress for this condition is 26,643 psi. If no restraint to bending were applied, the end of the tube would have a transverse deflection of 5.62 inches and the axial stresses would be reduced to 4566 psi compression. Suppose two buckstays are used, one at the end and one at midspan. If the end buckstay will allow the tube to deflect 1.00 inch then the ratio of
Fig. 13 Moment Distribution in Stayed Tube--One Buck Stay
Fig. 14 Moment Distribution in Stayed Tube—Two Buck Stays
Fig. 15
Stress Reduction by Controlled Deflection Through the Use of Buck Stays
Fig. 16
Example of Deflections and Reactions in Superheater Inlet Tube

Heat Flux = 47,510 Btu/ft²-hr
Tube: 321 S.S.
O.D. = 0.75 in.
I.D. = 0.59 in.
E = 25.1x10⁶ psi
α = 10.6x10⁻⁶/°F
ℓ = 69.2 in.
I = 0.00958 in.²
M_T = 564 in.-lb
δ_f = 5.62 in.
δ_e = 1.00 in.
Z = 0.897
δ_m = 0.145 in.
M = 506 in.-lb
P = 14.6 lb
allowed deflection/"free" deflection is 1.00/5.62 or 0.178. From Figure 15 the moment reduction factor is 0.897 giving an actual moment in the tube of 506 in-lbs with a resulting net axial compression of 24,372 psi. Thus, some reduction in stress has been obtained. The midpoint buckstay must allow a deflection of 0.145 inch in order to satisfy this condition. Note from Figure 15 that in order to obtain a moment reduction factor less than 1.00 some deflection must be allowed. The use of one buckstay requires a much higher deflection allowance than does the use of two buckstays.

Another consideration is the force and moment reactions set up at tube-to-header joints due to thermal expansion of the tubes. This has been evaluated at all sections throughout the boiler-superheater and because of the flexibility of the tube bundles has been found to be small. For example, Figure 17 shows the conditions for a superheater tube. The temperature rise of the superheater tube has been assumed to be the full rise of temperature from room conditions to the mean superheater tube temperature with the connection points remaining fixed. Even under this extreme condition the reaction force and moment are seen to be negligible. However, it should be mentioned that this considers only thermal expansion and does not account for any tendency for the tube to rotate due to bending. It is believed that the effects of rotation should be considered, but a satisfactory method of analysis has not yet been completed.

7. Reduction of Stresses

Of the possible methods for reducing longitudinal stresses, reduction of the asymmetric temperature gradient is the most obvious. This might be accomplished by allowing spaces between the tubes, and placing them in front
Superheater Tube

O.D. = 0.75 in.
I.D. = 0.59 in.
\( I = 0.00958 \text{ in.}^4 \)
\( E = 23.8 \times 10^6 \text{ psi} \)
\( \alpha = 10.95 \times 10^{-6} / \text{°F} \)
\( \Delta T = 836 \text{ °F} \)

Fig. 17 Example of Thermal Reaction Loads
of a diffusely reflecting wall. Then a portion of the incident radiation
would arrive at the wall and be diffusely reflected onto the rear surfaces
of the tubes. Utilization of the rear sections of the tubes as heat
receiving surfaces would make more receiving area available than in the
side-by-side design, thereby helping to offset the additional wall area
required to allow for spacing.

The current design for the 1000 kW bench model boiler calls for close
spacing of all preheater, boiler and superheater tubes. Thus, the heat
flux incident on any given tube would be a maximum on the front of the tube
and would fall to zero on the side of the tube. More precisely, the heat
flux distribution on any given tube could be expressed as:

\[ Q'(\beta) = Q \cos \beta \quad \text{for} \quad 0^\circ \leq \beta \leq 90^\circ, \text{ and} \]

\[ Q'(\beta) = 0 \quad \text{for} \quad 90^\circ \leq \beta \leq 180^\circ, \]

where \( Q \) is the heat flux incident on the front of the tube, \( \beta \) is the angle
between the incident flux and the radius vector defining a point on the tube
and \( Q'(\beta) \) is the heat flux at the point defined by the angle \( \beta \).

The thermal stress associated with this asymmetric heat flux can be
quite large. For example, in the superheater section of the boiler, with an
incident heat flux of 47,500 Btu/hr-ft\(^2\), the longitudinal stresses have been
calculated to be as large as 28,600 psi at a mean operating temperature of
1007\(^\circ\) F. Such a stress is comparable to the maximum allowable working
stress for the TP321 stainless steel tubes* to be used in that section of the boiler. It should be noted that of the 28,600 psi longitudinal stress, approximately 26,000 psi can be associated with the asymmetrical heat flux to the tubes.

The temperature distributions discussed earlier are further evidence of the problem. Note the almost complete lack of temperature rise on the back of the tube shown in Figure 3. There is essentially no heat flow around the relatively thin-walled tubes. Thus, the application of additional heat to the back of the tubes appears to offer certain advantages.

A preliminary study has been conducted to determine the effects of placing an insulating surface behind the tubes, and allowing a fraction of the incident radiation to pass between tubes and be diffusely reflected to their back surfaces. Such an indirect source of heat applied to the backs of the tubes would tend to lower the asymmetry associated with the heat flux, and thus would lower the thermal stresses due to this asymmetry. A primary objective of this study is to determine the optimum spacing of the tubes with respect to each other and with respect to the wall so that effective heating of the backs of the tubes can be accomplished.

A mathematical equation has been derived which expresses the heat flux incident at any point on the back of a tube in terms of the direct heat flux, the reflectivity of the wall, the spacing of the tubes with respect to each other, and the spacing of the tubes with respect to the wall. The

---

*The ASME Boiler and Pressure Vessel Code gives a maximum allowable working stress of 28,800 psi for grade TP321 steel operated at 1050°F.
"reflectivity" as used here is the ratio of the reflected and re-emitted energy to the total incident energy.

The mathematical model used in this derivation is illustrated in Figures 18 and 19. An incident flux of $Q$ Btu/hr-ft$^2$ was assumed to impinge on the tubes and the wall. The heat flux passing between adjacent tubes was allowed to strike the wall and be diffusely reflected. The radiation incident on the tube at a point defined by the angle $\beta$, and originating from the infinitesimal surface element $\Delta y \Delta z$ at the point $(y,z)$ on the wall was determined to be:

$$\Delta Q'(\beta) = \frac{QR}{\pi} \left[ \gamma^2 \cos (\pi-\beta) + \gamma \beta \sin (\pi-\beta) \frac{\Delta y \Delta z}{S^4} - \gamma \sin (\pi-\beta) \frac{\Delta y \Delta z}{S^4} \right],$$

where:

- $Q$ = incident heat flux,
- $R$ = wall reflectivity,
- $\beta$ = angle defining point on tube where heat flux is to be calculated,
- $S^2 = \gamma^2 + (\delta-y)^2 + z^2$,
- $\gamma = H - A \sin (\beta-\pi/2)$,
- $\delta = D - A \cos (\beta-\pi/2)$,
- $A$ = radius of tube,
- $H$ = distance from the center of the tube to the wall,
- $D$ = center-to-center spacing between adjacent tubes, and
- $(x,y,z)$ = coordinates of a point in the infinitesimal element $\Delta y \Delta z$. 

VII-51
Fig. 18
Pipe and Diffusely Reflective Surface as Seen From Behind Surface
Fig. 19 Plan View Pipes and Diffusely Reflective Wall
When this expression is integrated over the wall surface between two adjacent tubes, the heat flux at any point on the back of one of these tubes is given by:

\[
Q'(\beta) = \frac{QR}{2} \left\{ \cos (\pi - \beta) + \frac{\delta}{\gamma} \sin (\pi - \beta) \left[ \frac{(\delta - A)}{(\delta - A)^2 + \gamma^2} - \frac{(\delta - D + A)}{(\delta - D + A)^2 + \gamma^2} \right] \right. \\
- \sin (\pi - \beta) \left[ \frac{\delta(\delta - A) + \gamma^2}{\gamma \sqrt{(\delta - A)^2 + \gamma^2}} - \frac{\delta(\delta - D + A) + \gamma^2}{\gamma \sqrt{(\delta - D + A)^2 + \gamma^2}} \right] \right\},
\]

where all symbols have been previously defined.

The above expression has been evaluated for four different tube-to-wall spacings. For each of these cases, the heat flux at eight different angles around the tube between 90 and 180 degrees was determined as a function of the tube-to-tube separation. In all cases an incident flux of 50,000 Btu/hr-ft\(^2\), and a reflectivity of 1.0 was assumed. The results of these calculations appear as Figures 20-23, and Table VI. Note that all dimensions are given in terms of an arbitrary tube diameter of \(T\) units.

From Figure 20 it is seen that a wall-to-tube-center separation of 0.75 tube diameters yields fluxes that are quite small on the back of the tube (\(\beta \approx 180^\circ\)) for all tube separations. From Figure 23 it is seen that a wall-to-tube-center separation of three tube diameters yields fluxes that are small in the neighborhood of 90 degrees for all practical tube separations. Optimum values for tube spacings appear to lie in the neighborhood of one tube diameter for the wall-to-tube-center spacing, and two and one-half to three tube diameters for the tube-center to tube-center separation.
Q(β) x 10³ Btu/ft²-hr

Incident Heat Flux = 50,000 Btu/ft²-hr
Wall Reflectivity = 1.0

β = 123.75°, 112.5°, 101.25°, 135.0°, 90.0°, 146.25°, 157.5°, 168.75°

Tube Diameters (T)
D = Pipe Separation (center-to-center)

Fig. 20
Heat Flux to Back of Tube as A Function of Tube-to-Tube Separation. Distance of Tube-Center from Wall Equals 3/4 of a Tube Diameter.

VII-55
Incident Heat Flux = 50,000 Btu/ft\(^2\)-hr
Wall Reflectivity = 1.0

Fig. 21
Heat Flux to Back of Tube as a Function of Tube-to-Tube Separation.
Distance to Tube-Center from Wall Equals One Tube Diameter

Q(θ) = 10^3 Btu/ft\(^2\)-hr

VII-56
Incident Flux = 50,000 Btu/ft²-hr
Wall Reflectivity = 1.0

Heat Flux to Back of Tube as a Function of Tube-to-Tube Separation.
Distance of Tube-Center from Wall Equals 1.5 Tube Diameters

Fig. 22
Incident Flux = 50,000 Btu/ft\(^2\)-hr
Wall Reflectivity = 1.0

Heat Flux to Back of Tube as a Function of Tube-to-Tube Separation.
Distance of Tube-Center from Wall Equals 3.0 Tube Diameters
Table VI
Variation of Maximum and Minimum Heat Flux to Backs of Tubes as a Function of Tube Position

<table>
<thead>
<tr>
<th>Tube-Center-to-Tube-Center Separation</th>
<th>Distance of Tube Center from Wall</th>
<th>Minimum Flux (and its angle)</th>
<th>Maximum Flux (and its angle)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5 T *</td>
<td>0.75 T</td>
<td>4100 (168°)</td>
<td>16250 (122.5°)</td>
</tr>
<tr>
<td></td>
<td>1.0 T</td>
<td>2750 (90°)</td>
<td>13300 (135°)</td>
</tr>
<tr>
<td></td>
<td>1.5 T</td>
<td>1750 (90°)</td>
<td>9750 (157.5°)</td>
</tr>
<tr>
<td></td>
<td>3.0 T</td>
<td>500 (90°)</td>
<td>4500 (168.75°)</td>
</tr>
<tr>
<td>2.0 T</td>
<td>0.75 T</td>
<td>5250 (168°)</td>
<td>22750 (123.75°)</td>
</tr>
<tr>
<td></td>
<td>1.0 T</td>
<td>7250 (90°)</td>
<td>20000 (135°)</td>
</tr>
<tr>
<td></td>
<td>1.5 T</td>
<td>4200 (90°)</td>
<td>16000 (146.75°)</td>
</tr>
<tr>
<td></td>
<td>3.0 T</td>
<td>1300 (90°)</td>
<td>8600 (157.5°)</td>
</tr>
<tr>
<td>2.5 T</td>
<td>0.75 T</td>
<td>5650 (168°)</td>
<td>26250 (123.75°)</td>
</tr>
<tr>
<td></td>
<td>1.0 T</td>
<td>10500 (90°)</td>
<td>23250 (135°)</td>
</tr>
<tr>
<td></td>
<td>1.5 T</td>
<td>7200 (90°)</td>
<td>19750 (146.75°)</td>
</tr>
<tr>
<td></td>
<td>3.0 T</td>
<td>2250 (90°)</td>
<td>13250 (157.5°)</td>
</tr>
<tr>
<td>3.0 T</td>
<td>0.75 T</td>
<td>5800 (168°)</td>
<td>27500 (123.75°)</td>
</tr>
<tr>
<td></td>
<td>1.0 T</td>
<td>11500 (168°)</td>
<td>25500 (123.75°)</td>
</tr>
<tr>
<td></td>
<td>1.5 T</td>
<td>10000 (90°)</td>
<td>22500 (135°)</td>
</tr>
<tr>
<td></td>
<td>3.0 T</td>
<td>3500 (90°)</td>
<td>15875 (157.5°)</td>
</tr>
</tbody>
</table>

*T = arbitrary tube diameter.
The above temperature profiles were used to evaluate the reduction in stresses due to indirect heating for a trial superheater case. The case chosen was 0.75 inches O.D., by 0.50 inches I.D. stainless steel tubing operating at 50,000 Btu/ft$^2$-hr and at superheater inlet conditions. Temperature profile and thermal stress calculations were first made assuming heat from one direction only. A maximum longitudinal stress of 31,200 psi was calculated from this cos$\beta$-type heating. Temperature profile and thermal stress calculations were then made for two superheater inlet cases involving additional heating by radiation scattered from a rear wall. Plots of temperature and longitudinal stress contours for these three cases are shown in Figures 24-26. Figure 24 shows side-by-side mounting of the tubes for reference.

In Figure 25, the tube centers were separated by two tube diameters (1.5 inches for superheater tubes) and the wall-to-tube-center distance was one tube diameter. Adding rear heating to this tube reduced the maximum stress from 31,000 to 24,000 psi and increased the total heat input from 3130 to 4570 Btu/hr per foot of tube.

In Figure 26, the tube centers were separated by two and one-half diameters and all other conditions were as described above. The maximum longitudinal stress was reduced from 31,000 to 23,000 psi in comparison with the side-by-side case, and the total heat input increased from 3130 to 4835 Btu/hr-ft of tube.

In the above calculations the reflected heat was assumed to come only from the illuminated region on each side of a given tube. Expansion of this work to allow contributions from all illuminated regions has been completed.
Temperature Contours - °F
Q = 50,000 Btu/hr-ft²
Outside Diameter = 0.75 inch
Wall Thickness = 0.125 inch

Longitudinal Stress Contours - psi
Tube Spacing = 1 diameter center to center

Fig. 24
Temperature and Longitudinal Stress Contours for Superheater Tubes Without Spacing Between Tubes
Temperature Contours - °F  
Q = 50,000 Btu/hr-ft²  
Outside Diameter = 0.75 inch  
Wall Thickness = 0.125 inch

Longitudinal Stress Contours - psi  
Tube Spacing = 2 diameters center to center  
Tube Center to Wall Distance = 1 diameter

Fig. 25  
Temperature and Longitudinal Stress Contours for  
Superheater Tubes Spaced in Front of Diffusely Reflecting Wall  

VII-62
Temperature Contours - °F
Q = 50,000 Btu/hr-ft²
Outside Diameter = 0.75 inch
Wall Thickness = 0.125 inch

Longitudinal Stress Contours - psi
Tube Spacing = 2-1/2 diameter center to center
Tube Center to Wall Distance = 1 diameter

Fig. 26
Temperature and Longitudinal Stress Contours for Superheater Tubes Spaced in Front of Diffusely Reflecting Wall
and the results are similar to those presented above. These calculations indicate that the optimum tube-center-to-tube-center separation would be \(2\frac{1}{2}\) tube diameters—unchanged from the previous work. The optimum tube-center-to-wall separation would be increased to \(1\frac{1}{2}\) tube diameters.

8. Radiant Heat Transfer Considerations

In the CNRS Solar Furnace radiant energy is received at the focal point from the concentrating parabola through a very wide viewing angle, up to about 150 degrees. The current bench model design thus anticipates the use of a redirecting cone to collect the wide angle energy and direct it into the boiler aperture. One consequence of the redirecting cone is the creation of an area at the rear of the cavity having heat fluxes on the order of 80,000 Btu/hr-ft\(^2\), which are considered too high for safe operation. The problem is peculiar to the bench model optical arrangement, and would not occur in a full scale plant because no flux redirecting cone would be required.

One method of moderating the high heat flux is to place a relatively thin, diffusely reflecting ceramic plate in front of the "hot spot" and allow it to receive the energy then reflect and reradiate it within the furnace cavity. If this approach is adopted it is necessary to know the equilibrium temperature of the plate and the heat flux received on the tubes behind the plate.

Consider the system shown in Figure 27, with a thin plate receiving radiant energy and mounted in front of a line of closely spaced tubes. If the temperature drop caused by conduction through the plate is neglected, it can be assumed that half the absorbed energy leaves from each side of
Fig. 27  Diffusely Reflecting Plate and Tube Assembly
the plate, and at steady state the rate of energy absorption is equal to
the rate of emission:

\[ a (80,000) A = 2 A \varepsilon \sigma T^4, \]

where \( a \) = absorptance of the plate,
\( A \) = area of the plate,
\( \varepsilon \) = emittance of the plate,
\( \sigma \) = Stefan-Boltzmann constant, and
\( T \) = absolute temperature of the plate.

Solution of this equation leads to the following values for the plate
equilibrium temperature as a function of the \( a/\varepsilon \) ratio:

<table>
<thead>
<tr>
<th>Plate Temperature (deg F)</th>
<th>( a/\varepsilon )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1620</td>
<td>0.8</td>
</tr>
<tr>
<td>1740</td>
<td>1.0</td>
</tr>
</tbody>
</table>

To estimate the heat flux to the tubes from the diffusely emitting plate,
it is necessary to know the respective view factors. Designating the plate
by the subscript 1 and the tube assembly by the subscript 2:

\[ F_{12} = \text{view factor for the plate to the tubes} = 1 \]

\[ F_{11} = \text{view factor for the plate to the plate} = 0 \]

\[ \frac{A_2}{A_1} = \text{ratio of tube to plate area per unit length} = \frac{\pi}{2} \]
Note that

\[ A_1 F_{12} = A_2 F_{21} \]

\[ F_{21} = F_{12} \frac{A_1}{A_2} = \frac{2}{\pi} = 0.637 \]

\[ F_{22} = 1 - F_{21} = 0.363 \]

where \( F_{21} \) = view factor for the tubes to the plate

\( F_{22} \) = view factor for the tubes to the tubes

The gray body shape factor is given by:

\[ \gamma_{21} = \frac{1}{A_2 \left( \frac{1}{\varepsilon_1} - 1 \right) + \left( \frac{1}{\varepsilon_2} - 1 \right) + \frac{1}{F_{21}}} \]

Assuming \( \varepsilon_2 = \varepsilon_1 = 0.8 \) and evaluating the gray body shape factor gives

\[ \gamma_{21} = 0.452 \]

Taking a plate temperature of 1600\(^\circ\) F and a tube temperature of 1100\(^\circ\) F, the heat flux to the tubes becomes

\[ \frac{-Q}{A_2} = \gamma_{21} \sigma (T_1^4 - T_2^4) = 9360 \text{ Btu/hr-ft}^2 \]
based on tube area, or

\[
\frac{Q}{A_1} = 14,700 \text{ Btu/hr-ft}^2
\]

based on plate area.

Thus, a substantial drop in heat flux to the superheater tubes will occur if a plate is installed to intercept the incident radiation. A reduction to the range of 50,000 Btu/hr-ft\(^2\) is preferred; a lower value wastes heat transfer surface. It appears that a ceramic plate could be designed to operate at the equilibrium temperature.

Other thermal calculations have shown that a diffusely reflective coating on the tubes, with an absorptance of about 0.6, would reduce the heat flux in the "hot spot" to about the range desired. The reflecting coating would scatter the radiation not absorbed, to other areas inside the cavity. This possibility will be inspected more closely for the purpose of controlling longitudinal stresses as well as eliminating the hot spot.

9. Stress Reactions Among System Components

Temperature changes in restrained piping can cause bending stresses in two-dimensional systems, and bending and torsional stresses in three-dimensional systems. For safe operation the maximum stress due to thermal changes must be within the allowable stress range \(A_s\). The allowable stress range established for the thermal expansion stress is

\[
A_s = 1.25 C_s + 0.25 H_s \quad \text{(References 9 and 10),}
\]
where \( A_s \) = allowable expansion stress in psi,

\( C_s \) = allowable stress in the cold condition in psi, and

\( H_s \) = allowable stress in the hot condition in psi.

Preliminary calculations were made to understand a possible piping system in the steam generating section of the 1000 kW bench model boiler. The piping system was analyzed by using the two-dimensional method described by Spielvogel 9/1. The heat absorbing pipe was assumed to be fixed at its two ends and subjected to thermal expansion. Under these conditions, each end of the tube will react with a force and a moment. The piping configuration used for the analysis of thermal expansion stresses is shown in Figure 28. The calculations were performed assuming a header-to-header separation of seven feet. Results were obtained for tube-to-header standoff distances of one, two, three and four feet. Thermal expansion stresses at a, b, c and d were calculated in each case. The results are shown in Table VII. The thermal stresses of tube bend at points b and c must be corrected by multiplying by a stress intensification factor \( i \) 9/1. For one-inch schedule 80 tube, \( i \) is equal to \( 0.9 / (0.5548R)^{2/3} \), where \( R \) is the bend radius of the tube. The data in Table VII for points b and c were obtained by assuming a value of 1.0 for the intensification factor \( i \). This corresponds to a bend radius of 1.54 inches. A larger bend radius would give a safer design at points b and c, although the rules of the stress calculation procedure do not allow assumption of stress intensification factors less than 1.0. The results of these calculations indicated that when ab and cd are less than two feet, the thermal stresses at points a and d are beyond the allowable expansion stress of 25,050 psi (see Table VII). When ab and cd are greater...
Fig. 28 Two-Dimensional Piping System
Table VII Thermal Stresses in Piping Systems

Assumption: (1) Use one inch nominal pipe size, schedule 80, TP347 stainless steel

(2) Steam temperature in 1-inch pipe = 1100°F

(3) Thermal expansion 12.7 inch/100 ft

(4) Modulus of elasticity = \(16.0 \times 10^6\) psi at 1100°F
   \[= 29.0 \times 10^6\] psi at 70°F

\[A_s = 1.25 C_s + 0.25 H_s\]
\[= 1.25 \times 18,750 + 0.25 \times 6450 = 25,050\] psi

<table>
<thead>
<tr>
<th>Location</th>
<th>Thermal Stresses (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(ab = cd = 1) ft</td>
</tr>
<tr>
<td>a</td>
<td>188,600</td>
</tr>
<tr>
<td>b</td>
<td>23,300</td>
</tr>
<tr>
<td>c</td>
<td>23,300</td>
</tr>
<tr>
<td>d</td>
<td>188,600</td>
</tr>
</tbody>
</table>

than three feet, the thermal stresses at points a and d are within the allowable stress range. The thermal expansion stresses at points b and c were within the allowable range for all cases considered.

10. Safety Appliance Requirements for the Bench Model Boiler

The ASME Boiler and Pressure Vessel Code outlines very specific requirements concerning the implementation and use of safety equipment on
all boilers. These requirements vary with the type of boiler, its capacity and its designed operating temperature and pressure. Included in the list of necessary safety equipment are appropriately sized, constructed and mounted pressure gauges, water level indicators, relief valves, drain systems, test instrument fittings, and the plumbing necessary to attach these systems to the boiler. The specific requirements for power boilers and high pressure high temperature water boilers are given in Part PG, Section I of the Code. A list of the minimum safety equipment requirements for the bench model boiler is given below. Referenced and supporting material from the Code can be found in Appendix A. The minimum bench model safety equipment requirements are as follows:

(a) A superheater drain (PG-59.4.1.1)
(b) A boiler drain (PG-59-4.1.2)
(c) Two water gauge glass, or one water gauge glass and two independent remote level indicators (PG-60.1.1)
(d) One pressure gauge graduated to approximately twice (at least 1½ times) the operating pressure of the boiler (PG-60.6.1)
(e) One valve connection of at least ½-inch pipe size for the exclusive purpose of attaching a test pressure gauge to the boiler (PG-60.6.3)
(f) One temperature gauge to indicate the temperature of water in the boiler near the outlet connection (PG-60.6.4)
(g) One safety relief valve for the boiler section (PG-67, PG-68.2)
(h) One safety valve for the superheater outlet (PG-68.1)
(i) One stop valve for the boiler discharge outlet

(ANSI* B31.1 122.1.7, A. 1 and B. 6)

(j) The ability to add makeup water to the system while under pressure (PG-61.4).

*American National Standards Institute.
C. COLLABORATION WITH THE CNRS SOLAR ENERGY LABORATORY

The CNRS Solar Energy Laboratory operates the world's largest solar furnace in the Pyrenees Mountains of southern France. This facility supplies approximately one megawatt of thermal energy and is larger than the next three comparable solar furnaces by a factor of about 30. Because of its size, the CNRS 1000 kW Solar Furnace is the only realistic test facility now existing for large scale central receiver development programs. In addition to the facility itself, the CNRS Solar Energy Laboratory possesses the world's most experienced professional staff in the field of large solar installations.

The major components of the CNRS Solar Laboratory are: (1) an array of 63 heliostats each 6 meters by 7.5 meters, (2) a 1923 square meter parabolic concentrator built into the side of the 9 story laboratory building, and (3) a focal building with a working space of approximately 10 feet cubed into which the collected solar energy is concentrated. Figures 29 through 31 illustrate the general scale and layout of the facility.

Figure 32 shows the CNRS focal building as seen from the plane of the parabola. Heliostats are visible on the hillside in the background and the focal building doors are brightly illuminated. These doors are positioned several feet forward of the focal plane when they are closed as shown in Figure 32, so that the incident radiation is spread over a relatively large area and the door surfaces do not require artificial cooling. The lower center area of the doors displays a shadow of the focal building; since a parabola focusses only that energy arriving parallel to its axis, this shadow is always present and mirrors are omitted in the heliostat field and
Fig. 29 Aerial View of CNRS Solar Energy Laboratory--Odeillo, France
PARABOLIC REFLECTOR CONCENTRATES SUN'S RAYS ONTO TARGET AREA

Fig. 30 Schematic Diagram of CNRS Solar Furnace
Fig. 31 CNRS Solar Furnace, Odeillo-Font Romeu, France
on the parabola where the focal building interrupts the beam. The bench model boiler design must be adjusted to compensate for this feature of the flux pattern.

The usual mode of operation at the CNRS Solar Furnace is to adjust heliostats while the focal building doors are closed, then to open the doors at the beginning of an experiment. Figure 33 shows the doors opened and an experiment in progress in the focal area. Details of experiments cannot be distinguished except by the use of optical filters to attenuate the extremely bright illumination at the focal point.

The Solar Furnace is provided with an elevator for raising experimental equipment to the focal area which is about 60 feet above the ground. Figure 34 shows the ground level hallway which extends from the focal building to the main laboratory building on which the parabola is mounted. The folding gate at the rear of the hall in Figure 34 is the elevator entrance, and similar folding gates visible in the upper and lower sections of Figure 33 are elevator entrances at those levels of the focal building. The areas shown in these two figures are key locations in planning for this program, because the experimental boiler-superheater equipment must follow this route to the focal zone where tests are conducted.

Close inspection of Figure 33 will show that test equipment is supported on a rolling carriage, which in turn is supported on a two-level platform in the focal building. The carriage is self propelled by an electric drive system so that experiments can be setup in the ground level hallway, then conveniently moved into the elevator, raised to the level of the focal zone and moved onto the platform. The upper platform can be moved east or
Fig. 33 Focal Area of CNRS Solar Furnace with Experiment in Progress
Fig. 34 CNRS Solar Furnace, Ground Level Hall Between Focal Building and Laboratory Building
west with respect to the lower platform, and the lower platform can be moved up or down. This system makes position adjustments possible in order to place the test apparatus at the desired location with respect to the focal point of the furnace.

The hallway immediately beneath the focal building, shown in Figure 34, has rather low headroom for assembly of equipment (about 11.5 feet beneath the roof beams). An adjacent assembly room inside the main laboratory building is shown in Figure 35; this area has clear headroom of about 18 feet excluding the lighting fixtures and is located near the main machine shop of the Solar Energy Laboratory.

As part of the planning activity on the present research program, two investigators from Georgia Tech visited the CNRS Solar Furnace for ten days during August and September 1974. The purposes of this trip were to obtain data on the CNRS facility and to confer with CNRS scientists on the proposed boiler-superheater design and test plan. Careful measurements of the facility were made, detailed photographs were taken, and a set of scale drawings of the focal area were prepared. The drawings, shown in Figures 36 through 38, are intended to show details affecting the placement and operation of experimental equipment; some features have been omitted in the interest of clarity for the intended purposes. The maximum size of an item that can be moved into the focal area is approximately a ten-foot cube. Details of the available electrical and cooling water service are given in Table VIII.

Photographs, dimensioned sketches, operational details and other data concerning the CNRS Solar Furnace are available at Georgia Tech and will be provided to NSF and its contractors as needed in support of solar energy
Fig. 35 CNRS Solar Furnace, Assembly Room in Laboratory Building
Fig. 36
Top Plan View of Focal Building at CNRS Solar Furnace. Approximate Scale: 1/4 in. = 1 ft
Fig. 37
West Side Elevation of Focal Building at CNRS Solar Furnace. Approximate Scale: 1/4 in. = 1 ft
Fig. 38
Front Elevation of Focal Building at CNRS Solar Furnace.
Approximate Scale: 1/4 in. = 1 ft
research activities. The CNRS staff and facility represent the highest state-of-the-art competence now in existence for large scale use of solar energy.

Table VIII Utilities Available at CNRS Solar Furnace

<table>
<thead>
<tr>
<th>Utility</th>
<th>Availability</th>
</tr>
</thead>
<tbody>
<tr>
<td>Useable cooling water capacity in one day</td>
<td>250 m³ (66,000 gal)</td>
</tr>
<tr>
<td>Maximum cooling water flow rate</td>
<td>40 m³/hr (10,500 gal/hr)</td>
</tr>
<tr>
<td>Maximum cooling water flow rate during electrical power failure</td>
<td>10 m³/hr (2,600 gal/hr)</td>
</tr>
<tr>
<td>Cooling water delivery pressure</td>
<td>17 atm (250 psi)</td>
</tr>
<tr>
<td>Electrical power for lighting and instruments</td>
<td>230 volts, 1 phase, 50 Hz</td>
</tr>
<tr>
<td>Electrical power for motors and equipment</td>
<td>380 volts, 3 phase, 50 Hz</td>
</tr>
<tr>
<td>Weight capacity of elevator and focal building platform</td>
<td>10 metric tons (22,000 lbs)</td>
</tr>
</tbody>
</table>
D. MODEL OF CNRS FACILITY AND PROPOSED BENCH MODEL BOILER/SUPERHEATER

In order to insure compatibility between the CNRS Solar Furnace and the bench model boiler-superheater, Georgia Tech has constructed scale models of the focal building and the major items of boiler equipment. The focal building model is shown in Figure 39 and the equipment models are shown in Figure 40. The ground level hall connecting the laboratory and focal buildings is quite prominent in the focal building model, but is easily overlooked in pictures of the complete solar furnace. The curved roof edge on this hall corresponds to the position of the parabolic concentrator. Models of three separate equipment assemblies are shown; these are the boiler and carriage on the left, the feedwater pump and preheater in the center and the flux redirector on the right. The steam condenser is mounted on the boiler carriage (this is the vertical vessel at the rear), pipes carrying high pressure superheated steam are shown in white, pipes carrying high pressure water and saturated steam are shown in black, and hoses carrying cooling water are shown in gray with a slightly smaller diameter than the pipes.

The flux redirector shown on the right in Figure 40 is peculiar to the bench model system and would not be required on full scale boilers. Its purpose is to match the radiant energy pattern available at the Solar Furnace to that required by the boiler-superheater cavity. The cavity shape was chosen to be a miniature of the full scale design in order to facilitate eventual scale up. The full scale boiler will receive radiation from a maximum viewing angle of about 90 degrees, but energy arrives at the focal point of the Solar Furnace from a maximum viewing angle of about 150 degrees.
This incompatibility could have been resolved by eliminating all heliostats supplying radiant energy outside of a 90 degree angle at a substantial loss in thermal power, or by designing a device to reflect the energy arriving outside this angle and direct it into the cavity; the latter approach was selected. The flat panel adjoining the flux redirector is a refractory door to permit quick shutdown of the boiler-superheater. The large furnace doors require about one minute to open or close and the 63 heliostats require several minutes to be moved into a parking position. In the event that a serious problem is detected in the experimental boiler, the ability to shut down quickly is absolutely necessary to protect personnel and equipment.

Figure 41 shows the boiler and carriage resting on the focal building platform and the feedwater preheater and pump located below the boiler in the configuration currently envisioned for the installation at the Solar Furnace. The model of the platform accurately represents the installation at the Solar Furnace; the platform is supported by four vertical posts to permit movement of the equipment in a vertical direction.

Figure 42 shows the focal building model with the boiler assembly and feedwater preheater and pump positioned in the ground level hallway which was previously shown in Figure 34. The relationship of the hall, elevator and focal area can be seen in this figure, along with the boiler components for size comparison.

Figure 43 is a side view of the focal building model showing the feedwater preheater and pump on the fourth floor and the boiler and carriage on the fifth floor. The importance of careful planning can be seen in this
Fig. 41 Bench Model Boiler, Feed-water Preheater and Pump
Fig. 42 Model of Focal Building and Bench Model Boiler.
Fig. 43 Side View of Focal Building Model with Boiler in Operating Position
figure, because working space is severely limited in the focal area. There is a wall between the boiler and fifth floor hallway in the foreground of the photograph which has been omitted in the model to permit the boiler to be seen, and there is machinery on the fourth floor to adjust the platform level. The large elevator shaft is immediately to the left of the focal area, followed by laboratory space, a stairway and passenger elevator shaft, and on the fifth floor the heliostat control room is at the extreme left.

Figure 44 is a front view of the focal area as it would be seen from the plane of the parabolic mirror. (This view is analogous to Figure 33.) The flux redirector is positioned in front of the boiler, and other equipment items are located as they will be during actual tests. It should be noted that all the experimental equipment can be located in the focal area of the CNRS facility, on the fourth and fifth floors of the focal building. It is planned that recording and control instruments will be placed in the laboratory area on the fifth floor near the center of the structure. Earlier in the program it had appeared that some equipment would have to be located on the ground and interconnected by piping along the side of the focal building. The presently proposed arrangement greatly simplifies assembly and testing, both at the Solar Furnace and before shipment to France.
Fig. 44
Model of Focal Area with Flux Redirector and Bench Model Boiler in Operating Position
E. TEST PLAN FOR BENCH MODEL SYSTEM

Georgia Tech's experience in conducting experiments at the CNRS Solar Furnace indicates that a carefully prepared test plan will be required in order to maintain control of the bench model tests. During a test, events occur very rapidly, the noise level is high because of pumps and high pressure flowing water, and it is sometimes difficult to make visual observations because light levels range from very high at the focal point to normal in other parts of the apparatus. Usually several people are required to carry out various tasks, and some are located in the focal room, some in the pyrometry room (in the laboratory building), and some in the heliostat control room. Closed circuit television and an intercom system are employed, but their effectiveness is hampered by limited resolution capability in the television system and high noise level interference with the intercom.

The first version of the bench model test plan has been prepared. It is possible to devise a plan at this time based on our current estimates of how the bench model apparatus will behave. However, surprises must be expected when the tests are underway and continuous revision of plans must therefore be anticipated. One major factor which must be considered in the development of any test plan involving the use of a solar furnace facility is the availability of solar energy. This is especially important at the CNRS facility where cloudy days occur far more frequently than in the southwestern United States. Careful planning and scheduling is required to most effectively utilize the available solar furnace time.

Figure 45 shows the number of hours of clear sky (hours available for solar furnace tests at Odeillo, France) as a function of months of the year.
Fig. 45
Number of Hours of Clear Sky for Continuous Periods of at Least Three Hours at Odeillo, France, 1969-1973
This figure was compiled from data published by the CNRS Solar Energy Laboratory for Odeillo for the years 1969-1973. Only those hours were considered which were continuous for at least one-half day. Depending upon the time of the year these half-day continuous periods ranged from 3 hours and 10 minutes in December to 4 hours and 5 minutes in July and August. Figure 45 suggests that the five months of July through November are the best for conducting long time solar furnace tests. Also, the second half of January and the month of February usually are good months for solar furnace tests. There is very little solar furnace testing conducted during the period March through June.

In considering possible schedules for bench model receivers, first attention should be given to selecting a tentative date for the initiation of the solar furnace tests. Since the longest period of poor weather is March through June, it is important to avoid intentionally scheduling tests to begin too close to March since if the tests are not completed on schedule they might be delayed for several months. However, April, May and June can be considered good months for setting up equipment, hardware and instrumentation and checking out systems in preparation for tests which might be scheduled for the period July through November.

1. Flux Redirector

It is presently planned and has been proposed to the National Science Foundation that the flux redirector be constructed at Georgia Tech and tested at the CNRS Solar Furnace during the same period that the boiler-superheater is being constructed and tested at Martin Marietta Corporation in Denver. This plan anticipates that the flux redirector tests in France
would occur during July or August 1975; tests of the boiler-superheater in Denver would occur at about the same time. Thus, the flux redirector would be ready for use immediately upon arrival of the boiler-superheater components at the Solar Energy Laboratory.

A method must be devised to characterize the radiant flux output of the flux redirector in a manner that permits selection of heliostats to illuminate particular areas of the boiler cavity inner surface. For example, it will be necessary during start-up of the boiler to minimize heat input to the preheater and superheater tubes until steam has been generated in the system. After steam is available, the preheater and superheater will be started. The characterization should be verified with contributions from all 63 heliostats.

In order to accomplish the flux redirector characterization and permit operation of the boiler-superheater with less than full power, a masking system for the heliostats is planned. Each heliostat can be masked with a fabric which covers 75 percent of its area and thereby permits transmission of 25 percent of the normal sunlight. While the masks are in place a contribution from each heliostat will be received in the boiler, but the power input will be only 250 kW. Similarly, a final characterization of the flux redirector can be obtained with 25 percent of full power while all heliostats are operating.

The test program for the flux redirector is planned along the following lines:

(1) The flux redirector and emergency door will be installed on a frame connecting directly to the CNRS Focal Building so that
they are independent from the boiler-superheater. Pressure and flow tests will be conducted.

(2) The flux redirector will be tested at full power (one megawatt) using a water cooled cavity to simulate the boiler assembly and prevent damage to the Focal Building.

(3) A temporary boiler cavity consisting of plywood walls lined with a refractory fiber felt will be installed in the focal building. Its walls will be arranged to open so that the surfaces can be viewed with an optical pyrometer.

(4) Each heliostat in turn will be operated and its flux pattern on the cavity wall mapped using the pyrometer. After this measurement the masks will be installed and the pattern of each heliostat determined with the mask in place. The optical pyrometry technique has been used successfully to map flux patterns from single heliostats. The heliostat masks will be left in place for the following step.

(5) The flux pattern from groups of masked heliostats will be determined to verify the predicted patterns calculated from the results of step (4). After this sequence of tests is completed, it will be possible to identify which heliostats illuminate specific areas on the cavity wall and thus to exercise a degree of control over the flux applied to the preheating, boiling and superheating areas.
2. Bench Model Boiler-Superheater Assembly

The objectives of the bench model tests include the following:

1. demonstrate that the system can meet its design requirements,
2. collect experimental data which will be useful in the design and operation of larger solar boilers such as the proposed 5 megawatt system, and finally,
3. gain operational experience with a high temperature solar thermal conversion system. The test plan discussed below is an attempt to satisfy these objectives. This plan will be updated as necessary throughout the bench model construction program and after the tests are underway. In all probability each day's test plan will be finalized shortly before the tests are run so that results from previous tests can be taken into account. However, the plan for each day must fit into the overall test sequence in a logical manner, so that the goals of the program can be achieved within the allotted time.

The startup of any boiler is, in general, a critical operation. Most modern power boilers consist of at least a preheating section, a boiling section, and a superheating section. Additionally, many commercial plants also contain reheating systems. During startup the heat applied to each of these sections and the fluid flow through these sections must be carefully controlled to avoid unstable operating conditions and possibly boiler damage.

Startup of the bench model cavity boiler will involve many of the problems associated with the startup of a conventional boiler, plus those peculiar to a solar heated system. For example, during the early stages of startup, prior to the generation of significant quantities of steam, there will be little or no flow of heat removing fluid in either the preheater or
superheater sections of the boiler. Thus, these sections must not receive large quantities of heat at startup. For the superheater section the problem is even more critical. The superheater must receive some heat to keep steam from condensing in that section of the system. A small amount of water in a superheater tube will plug that tube to the flow of steam. The result will be a loss of coolant flow in the tube and eventually tube burnout. Thus, the heat flux applied to the superheater must keep all parts of the superheater above the saturated steam temperature and yet well below the maximum allowable tube wall temperature.

The control of heat to the various sections of the bench model will be affected primarily by the judicious selection of heliostats. The flux redirector will have been checked out and characterized before the boiler-superheater equipment arrives at the CNRS Solar Energy Laboratory. Thus, it will be possible to select heliostats to illuminate a particular section of the surface in the cavity receiver. If, during startup, adequate control of the flux cannot be accomplished by this method, then ceramic and/or water cooled baffles will be incorporated for startup. These baffles may be placed on either side of the focal point, but preferably on the side nearest the CNRS concentrator mirror, outside of the cavity volume.

Upon arrival of the boiler at CNRS and prior to its installation in the focal room, the bench model will undergo extensive testing. The purpose of these tests will be to insure that the boiler or any of its accessory equipment has not been damaged in shipment. These tests will include a complete operational checkout of all instrumentation and hydrostatic pressure tests of all pressure containing components.
Following the installation of the boiler in the focal zone, the boiler will be used to generate steam at approximately 100 psi. This operation will serve several purposes besides being the first true check on the operation of the system. First, low pressure operation will allow much needed hands-on experience to be gained with the boiler. Coordination between the boiler operator and the heliostat operator will be established during this period. The effects of various transient phenomena such as cloud movement, loss of cooling water, an emergency door closing will be studied.

The 100 psi operation will also serve as part of the final cleanup procedure for the water side walls of the system. The boiler will be operated during this period with water containing high parts-per-million concentrations of several strong alkaline compounds. Blowdown will be performed periodically to remove sludge and debris. Following this procedure the boiler will be drained, flushed, inspected and refilled with high purity water. The details of this procedure are found in Section F of this report.

Operation of the boiler at various power levels, including the initial 100 psi operation will be affected by the masking of heliostats with black cloth, and possibly by not using all heliostats. The masking technique will be identical to that to be used in characterizing the flux redirector.

It is presently envisioned that provision will be made by masking to operate each heliostat at 10, 25, 50 and 100 percent of its area, and thus power. This will yield power levels of 100, 250, 500 and 1000 kilowatts at the furnace. These masks will allow a heat flux reduced in power but
otherwise characteristic of the furnace to impinge on the boiler. Operating experience and detailed experimental data will be obtained at each of these power levels on the way to 1000 kW. This will require that significant amounts of time be spent at and below each of these power levels.

At each of the above mentioned power levels the following goals will be met: (1) demonstrate that the system is functioning as designed with respect to fluid flow, temperatures, and pressures; (2) gain operating experience with the system; (3) determine the need for new and/or different tests and experiments at this or other power levels; (4) study the effect of using various combinations of heliostats to control power level and heat distribution; (5) determine the effects of transient phenomena such as those associated with cloud movement, loss of electrical power, and the closing of the emergency door; (6) determine temperatures and strains at numerous positions in all heat receiving sections of the system; (7) determine the power level and efficiency of the system; and finally, (8) determine the major sources of heat loss from the system. The achievement of these goals will require the joint cooperation of Martin Marietta, Georgia Tech and CNRS researchers and personnel.
F. WATER TECHNOLOGY FOR THE BENCH MODEL BOILER

In the field of high pressure boilers, there cannot be any compromise in water quality. The most efficient combination of techniques must be used to assure a delivery of pure water to the feedwater system of the boiler. Likewise, great care must be exercised in maintaining the quality of the water after it has entered the boiler. At the designed pressures of the proposed bench model boiler, the presence of feedwater impurities in even the parts-per-billion range can rapidly lead to serious problems related to corrosion, scale formation, and carryover of solids with the steam to the superheater. It should also be noted that a significant relaxation of these water quality criteria is not justified based on the anticipated short service life of the bench model boiler.

The greatest incidence of problems in steam generating equipment is related to water quality, or more correctly, the lack of it. To minimize these problems in the bench model boiler great care must be exercised in the design, fabrication, initial cleaning and startup, and in the steady state operation of the system.

The design of the boiler is important in that it affects the fabrication and service procedures necessary to insure a clean system. Care must be taken to design a system that can be fabricated with minimal contamination, cleaned efficiently, inspected thoroughly, and operated with positive control over its water quality.

Fabrication procedures for the system should be such as to minimize the amount of foreign material left in the system. Necessary precautions include the use of water soluble lubricating oils when at all possible, and the

VII-106
thorough cleaning of construction materials prior to their assembly. Every effort should be made to minimize the introduction of oil and grease into the boiler since the quantity of these materials determines the length of cleaning and the degree of difficulty in obtaining clean surfaces. Regardless of the care taken during fabrication, the new boiler will require extensive cleaning to remove all foreign materials from its water side surfaces. Materials likely to be found in the new boiler include lubricants, oils, sand, metal fragments, loose mill scale, corrosion products, and assorted debris. It is anticipated that much of the cleanup procedure can be performed in Denver prior to shipment of the boiler to France.

The following is a procedure used by the Georgia Power Company to prepare their new boilers for firing. This procedure has been recommended 12/ as suitable for cleanup of the bench model boiler:

(1) Flush system several times with water by filling and draining. This procedure should be carried out by filling from the superheater end of the system to avoid forcing material into the superheater.

(2) Perform necessary hydrostatic testing.

(3) Perform an alkaline boilout of the system by filling the boiler to its normal operating level with a trisodium phosphate-caustic soda-water solution and generate steam at approximately 100 psig for 10 to 12 hours. Short blowdowns should be performed once an hour during this period to remove sludge and debris. The trisodium
phosphate and caustic soda is present in high-parts-per-million concentrations.

(4) Completely drain system.

(5) Open and inspect steam drum for the presence of sludge and debris.

(6) Flush steam drum with water to remove sludge and debris. Reseal steam drum.

(7) Using water, fill and drain system several times to remove residual material.

(8) Fill boiler with water to normal operating level and generate steam at 100 psig. Cool and drain system.

(9) Fill boiler with chemically treated pure water and attempt to go to power. More blowdown may be required at this time to achieve desired water purity.

During high temperature, high pressure operation of the bench model boiler it is absolutely essential that the quality of the boiler water be carefully controlled. The pH of the water must remain between 9.2 and 9.3 if corrosion of the steel is to be minimized. The presence of dissolved oxygen greatly accelerates this corrosion. Therefore, dissolved oxygen levels must be kept in the low parts-per-billion range. The presence of ionizable solids increases the conductivity of the water, and thus its corrosive nature. In all modern high pressure boilers the conductivity
of the water is maintained below 0.1 micromho/cm to minimize this effect. The presence of parts-per-million concentrations of oils or undissolved solids in the boiler water tends to cause foaming in the steam drum and subsequent carryover of solids into the superheater section. The bench model system must be maintained free of these impurities.

The proposed feedwater system for the bench model boiler would use commercially available distilled water as input. The feedwater system itself would include a demineralizer section, a deaerator section and a chemical treatment section. The demineralizer section would consist of a pair of series-connected mixed bed ion exchange columns. Their purpose would be to guarantee the purity of the distilled water by removing any ionized mineral salt impurities.

The purpose of the deaerator section would be to remove dissolved gases such as oxygen and carbon dioxide from the feedwater. Commercial deaerators for large scale systems are available that can supply water containing less than 30 parts-per-billion dissolved oxygen. The search for a supplier of laboratory size units is presently underway.

The chemical section of the feedwater system would be used to control the pH of the boiler water and to treat the feedwater with an oxygen scavenger. It has been recommended by Georgia Power Company’s water treatment specialists that we use morpholine to maintain the pH between 9.2 and 9.3 and that we use hydrazine as an oxygen scavenger. Hydrazine concentrations of the order of 10 to 20 parts-per-billion in the preheater section of the boiler have been recommended.

Regardless of the purity of the makeup water, the concentration of impurities in the boiler water will increase to unacceptable levels if
unchecked. The mechanism for controlling and reducing this concentration of impurities in the boiler is called blowdown. Blowdown is the removal of a portion of the boiler water for the purpose of reducing its impurity concentration or to remove sludge. In the interest of precise control most modern high pressure boilers perform continuous blowdown; it may be advisable to incorporate capability to perform continuous blowdown into the bench model boiler.
Instrumentation for the bench model boiler must provide for the safe operation of the boiler, and simultaneously supply that data necessary to perform a complete and comprehensive analysis of the system's performance. Included in the data that must be recorded and monitored are fluid temperatures, fluid pressures, fluid flow rates, liquid levels, thermocouple temperatures, strain gauge readings, heat flux measurements, and instantaneous insolation measurements. The number of such variables requiring simultaneous logging is likely to exceed seventy-five in number. The system to monitor and record these data must be reliable, flexible, cost effective and within the economic limitations of the program. Such a system, centered around the use of a small digital computer, is described in the following paragraphs.

A schematic of the proposed instrumentation is shown in Figure 46. The heart of the system is a PDP-8/E* computer, a machine that has found numerous industrial applications in areas such as data acquisition, process control, and general computing. All components of the data collection system, including the interface and the transducers, are standard off-the-shelf items. No electronic design or buildup would be anticipated. Raw data supplied to the system would be in the form of electrical signals from the various transducers located on or in the vicinity of the boiler.

The computer interface subsystem would consist of a 12-bit analog-to-digital converter, a multiplexing system and an input port for each transducer. The purpose of the interface would be to sample each data line in turn and transmit the resulting digitized information to the small computer.

* Digital Equipment Corporation, Maynard, Mass.
Fig. 46 Proposed Instrumentation for Bench Model Boiler
for further processing. This operation would be done under computer control and could be accomplished at sampling rates up to 200 data channels per second. Thus, 100 thermocouples and transducers could be interrogated and the resulting data stored in the computer in an elapsed time of approximately 500 milliseconds. These data could be updated as often as necessary — once a second, once every three seconds, etc.

Once stored in the memory of the small computer, the data are available for a wide variety of tasks. What can be done with the data is then limited only by the program resident in the machine and the peripheral devices connected to it. Peripheral devices for the proposed system include a dual DECtape unit to allow magnetic tape recording of all raw data, a video display unit to allow visual display of pertinent data, and a teletype terminal to facilitate operator control of the system.

In the preceding paragraphs, the data acquisition capability of the proposed system has been described. Under probable operating conditions (i.e., sampling 100 transducers every three seconds) the data acquisition task would occupy the small computer for approximately 20 to 30 percent of the time. To demonstrate the versatility of the proposed system, the following are given as examples of tasks that could be performed during the remaining time:

1. Display pertinent data in alphanumeric form on the video display unit;

2. Plot pertinent data on the video display unit;
(3) Monitor every parameter and alert operator if any variable gets outside of predetermined limits;

(4) Determine rates of change of pertinent parameters and alert operator if any such rate of change gets outside of predetermined limits;

(5) Calculate the instantaneous input power to boiler;

(6) Calculate the instantaneous power loss in various parts of the system;

(7) Calculate the instantaneous power delivered to the load;

(8) Calculate the instantaneous efficiency of system; and finally,

(9) Upon detection of certain predetermined abnormal conditions, generate a signal for the automatic closing of the emergency door.

A last point should be made concerning the versatility of the proposed system. The PDP-8/E is a general purpose computer having extensive computational ability. Thus, the reduction of data from a given experiment could begin as soon as that experiment is concluded and the analysis of a given day's experiments would be available in time to help plan the following day's activities.
REFERENCES


3. Reference 2, Chapters 7 and 13.


12. Private communication with C. W. Jones, Chief Chemist, Georgia Power Company, Atlanta, Georgia.
APPENDIX A

REQUIRED GAUGES, INSTRUMENTS AND SAFETY EQUIPMENT FOR SOLAR BOILER

Excerpts From
PG-59. Application Requirements for the Boiler Program

59.4 Drains

59.4.1 Ample drains shall be provided where required to permit complete drainage of all piping, superheaters, waterwalls, water screens, integral economizers, high-temperature water boilers, and all other boiler components in which water may collect. Piping shall conform to the requirements of PG-58.3.6 or PG-58.3.7.

59.4.1.1 Each superheater shall be equipped with at least one drain so located as to most effectively provide for the proper operation of the apparatus.

59.4.1.2 Each high-temperature water boiler shall have a 1-inch minimum pipe size bottom drain connection in direct connection with the lowest water space practical for external piping conforming to PG-58.3.7.

60.1 Water Level Indicators

60.1.1 Each boiler shall have at least one water gage glass. Boilers operated at pressures over 400 psi, except electric boilers of the electrode type shall be provided with two water gage glasses which may be connected to a single water column or connected directly to the drum.

*The following material is not a literal reproduction of that found in the Code. Only that material pertinent to the solar boiler has been reproduced.
Two independent remote level indicators may be used instead of one of the two required gage glasses for boiler drum water level indication in the case of power boilers with all drum safety valves set at or above 900 psi. When both remote level indicators are in reliable operation, the gage glass may be shut off but shall be maintained in serviceable condition.

When the direct reading of gage glass water level is not readily visible to the operator in his working area, two dependable indirect indications shall be provided, either by transmission of the gage glass image or by remote level indicators.

The lowest visible part of the water gage glass shall be at least 2 in. above the lowest permissible water level, at which level there will be no danger of overheating any part of the boiler when in operation at that level. When remote level indication is provided for the operator in lieu of the gage glass, the same minimum level reference shall be clearly marked.

Connections on the drum and the provisions for the piping between the drum and the remote level indicator shall comply with PG-60.2.2 for water columns and shall be completely independent of connections for any function other than water-level indication.

60.1.6 All connections on the gage glass shall be not less than 1/2-inch pipe size. Each water-gage glass shall be fitted with a drain cock or valve having an unrestricted drain opening of not less than 1/4 inch diameter to facilitate cleaning. When the boiler operating pressure exceeds 100 psi the glass shall be furnished with a connection to install a valved drain to the ash pit or other safe discharge point.

Each water gage glass shall be equipped with a top and a bottom shutoff valve of such through flow construction as to prevent stoppage by deposits.
of sediments. If the lowest valve is more than 7 ft above the floor or platform from which it is operated, the operating mechanism shall indicate by its position whether the valve is open or closed. The pressure-temperature rating shall be at least equal to that of the lowest set pressure of any safety valve on the boiler drum and the corresponding saturated-steam temperature.

Straight-run globe valves shall not be used on such connections.

Automatic shutoff valves, if permitted to be used, shall conform to the requirements given in A-18.

60.2 Water Columns

60.2.1 The water column shall be so mounted that it will maintain its correct position relative to the normal waterline under operating conditions.

60.2.2 The minimum size of pipes connecting the water column to a boiler shall be 1 in. For pressures of 400 psi or over, lower water column connections to drums shall be provided with shields, sleeves, or other suitable means to reduce the effect of temperature differentials in the shells or heads. Water glass fittings or gage cocks may be connected directly to the boiler.

60.2.3 The steam and water connections to a water column or a water gage glass shall be such that they are readily accessible for internal inspection and cleaning. Some acceptable methods of meeting this requirement are by providing a cross or fitting with a back outlet at each right-angle turn to permit inspection and cleaning in both directions, or by using pipe bends or fittings of a type which does not leave an internal shoulder or

VII-119
pocket in the pipe connection and with a radius of curvature which will permit the passage of a rotary cleaner. The water column shall be fitted with a connection for a drain cock or drain valve to install a blowoff pipe of at least 3/4 in. pipe size to the ash pit or other safe point of discharge. If the water connection to the water columns has a rising bend or pocket which cannot be drained by means of the water column drain, an additional drain shall be placed on this connection in order that it may be blown off to clear any sediment from the pipe.

60.2.4 The design and material of a water column shall comply with the requirements of PG-42. Water column made of cast iron in accordance with Specification SA-278 may be used for maximum boiler pressures not exceeding 250 psi. Water columns made of ductile iron in accordance with Specification SA-395 may be used for maximum boiler pressures not exceeding 350 psi. For higher pressures, steel construction shall be used.

60.2.5 Shutoff valves shall not be used in the pipe connections between a boiler and a water column or between a boiler and the shutoff valves required for the gage glass (PG-60.1.6), unless they are either outside-screw-and-yoke or level-lifting type gate valves or stopcocks with lever permanently fastened thereto and marked in line with their passage, or of such other through-flow construction as to prevent stoppage by deposits of sediment, and to indicate by the position of the operating mechanism whether they are in open or closed position; and such valves or cocks shall be locked or sealed open. Where stopcocks are used they shall be of a type with the plug held in place by a guard or gland.
60.2.6 No outlet connections, except for damper regulator, feed-
water regulator, drains, stem gages, or apparatus of such form as does
not permit the escape of an appreciable amount of steam or water therefrom
shall be placed on the pipes connecting a water column or gage glass to a
boiler.

60.3 Gage Glass Connections

60.3.1 Gage glasses and gage cocks that are required by PG-60.1
and PG-60.4 and are not connected directly to a shell or drum of the boiler,
shall be connected by one of the following methods:

60.3.1.1 The water gage glass or glasses and gage cocks shall be
connected to an intervening water column.

60.3.1.2 When only water gage glasses are used, they may be
mounted away from the shell or drum and the water column omitted, provided
the following requirements are met:

60.3.1.2.1 The top and bottom gage glass fittings are aligned,
supported, and secured so as to maintain the alignment of the gage glass; and

60.3.1.2.2 The steam and water connections are not less than 1 in
pipe size and each water glass is provided with a valved drain; and

60.3.1.2.3 The steam and water connections comply with the
requirements of the following PG-60.3.2 and PG-60.3.3.

60.3.2 The lower edge of the steam connection to a water column
or gage glass in the boiler shall not be below the highest visible water
level in the water gage glass. There shall be no sag or offset in the piping
which will permit the accumulation of water.
60.3.3 The upper edge of the water connection to a water column or gage glass and the boiler shall not be above the lowest visible water level in the gage glass. No part of this pipe connection shall be above the point of connection at the water column.

60.4 Gage Cocks. Each boiler (except those not requiring water level indicators per PG-60.1.2) shall have three or more gage cocks located within the visible length of the water glass, except when the boiler has two water glasses located on the same horizontal lines.

Boilers not over 36 in. in diameter in which the heating surface does not exceed 100 sq ft need have but two gage cocks.

The gage cock connections shall be not less than \( \frac{1}{2} \) inch pipe sizes.

60.6 Pressure Gages

60.6.1 Each boiler shall have a pressure gage so located that it is easily readable. The pressure gage shall be installed so that it shall at all times indicate the pressure in the boiler. Each steam boiler shall have the pressure gage connected to the steam space or to the water column or its steam connection. A valve or cock shall be placed in the gage connection adjacent to the gage. An additional valve or cock may be located near the boiler providing it is locked or sealed in the open position. No other shutoff valves shall be located between the gage and the boiler. The pipe connection shall be of ample size and arranged so that it may be cleared by blowing out. For a steam boiler the gage or connection shall contain a syphon or equivalent device which will develop and maintain a water seal that will prevent steam from entering the gage tube. Pressure gage
connections shall be suitable for the maximum allowable working pressure and
temperature, but if the temperature exceeds $406^\circ$ F, brass or copper pipe or
tubing shall not be used. The connections to the boiler, except the syphon, if used, they shall not be less than $\frac{1}{4}$ in. standard pipe size but where steel or wrought iron pipe or tubing is used, they shall not be less than $\frac{1}{2}$ in. inside diameter. The minimum size of a syphon, if used, shall be $\frac{1}{4}$ in. inside diameter. The dial of the pressure gage shall be graduated to approximately double the pressure at which the safety valve is set, but in no case to less than $1\frac{1}{2}$ times this pressure.

60.6.3 Each boiler shall be provided with a valve connection at least $\frac{1}{4}$ inch pipe size for the exclusive purpose of attaching a test gage when the boiler is in service, so that the accuracy of the boiler pressure gage can be ascertained.

60.6.4 Each high-temperature water boiler shall have a temperature gage so located and connected that it shall be easily readable. The temperature gage shall be installed so that it at all times indicates the temperature in degree Fahrenheit of the water in the boiler, at or near the outlet connection.

PG-61 Feedwater Supply

61.4 High Temperature water boilers shall be provided with means of adding water to the boiler or system while under pressure.

PG-67 Boiler Safety Valve Requirements

67.1 Each boiler shall have at least one safety valve or safety relief valve and if it has more than 500 sq ft of water heating surface, or if an
electric boiler has a power input more than 500 kW it shall have two or more safety valves or safety relief valves. The method of computing the steam-generating capacity of the boiler shall be as given in A-12.

67.2 The safety valve or safety relief valve capacity for each boiler (except as noted in PG-67.4) shall be such that the safety valve, or valves will discharge all the steam that can be generated by the boiler without allowing the pressure to rise more than 6 percent above the highest pressure at which any valve is set and in no case to more than 6 percent above the maximum allowable working pressure. The safety valve or safety relief valve capacity shall be in compliance with PG-70 but shall not be less than the maximum designed steaming capacity as determined by the manufacturer.

67.5 All safety valves or safety relief valves shall be so constructed that the failure of any part cannot obstruct the free and full discharge of steam and water from the valve. Safety valves shall be of the direct spring loaded pop type, with seat inclined at any angle between 45 and 90 degrees inclusive, to the centerline of the spindle. The coefficient of discharge of safety valves shall be determined by actual steam flow measurements at a pressure not more than 3 percent above the pressure at which the valve is set to blow and when adjusted for blowdown in accordance with PG-72. The valves shall be credited with capacities as determined by the provisions of PG-69.2.

Safety valves or safety relief valves may be used which give an opening up to the full discharge capacity of the area of the opening of the inlet of the valve (see PG-69.5), provided the movement of the steam safety valve is such as not to induce lifting of water in the boiler.

Deadweight or weighted lever safety valves or safety relief valves shall not be used.

VII-124
For high temperature water boilers safety relief valves shall be used. Such valves shall have a closed bonnet. For purposes of selection the capacity rating of such safety relief valves shall be expressed in terms of actual steam flow determined on the same basis as for safety valves. In addition the safety relief valves shall be capable of satisfactory operation when relieving water at the saturation temperature corresponding to the pressure at which the valve is set to blow.

67.7 Safety valves or safety relief valves may have bronze parts complying with either Specifications SB-61 or SB-62, provided the maximum allowable working stresses and temperatures do not exceed the values given in Table PG-23.2 and shall be marked to indicate the class of material used. Such valves shall not be used on superheaters delivering steam at a temperature over 450°F and 306°F respectively, and shall not be used for high temperature water boilers.

PG-68 Superheater Safety Valve Requirements

68.1 Every attached superheater shall have one or more safety valves near the outlet. If the superheater outlet header has a full, free, steam passage from end to end and is so constructed that steam is supplied to it at practically equal intervals throughout its length so that there is a uniform flow of steam through the superheater tubes and the header, the safety valve, or valves, may be located anywhere in the length of the header.

68.2 The discharge capacity of the safety valve, or valves, on an attached superheater may be included in determining the number and size of the safety valves for the boiler, provided there are no intervening valves between the superheater safety valve and the boiler, and provided the
discharge capacity of the safety valve, or valves, on the boiler, as distinct from the superheater is at least 75 percent of the aggregate valve capacity required.

68.6 Every safety valve used on a superheater or reheater discharging superheated steam at a temperature over 450°F shall have a casing, including the base, body and bonnet and spindle, of steel, steel alloy, or equivalent heat resisting material.

The valve shall have a flanged inlet connection, or a weld-end inlet connection. It shall have the seat and disk of suitable heat erosive and corrosive-resisting material, and the spring fully exposed outside of the valve casing so that it shall be protected from contact with the escaping steam.
APPENDIX B

PROGRAM MEETINGS AND CONTACTS
APPENDIX E

PROGRAM MEETINGS AND CONTACTS

1. First Quarter (February 10 through April 18, 1974)

J. D. Walton, Jr. and S. H. Bomar, Jr. of Georgia Tech visited the Martin Marietta plant in Denver on February 10 and 11, for the purpose of planning study activities. F. A. Blake, T. R. Tracey and M. T. Howerton of Martin Marietta participated in the meeting.

F. A. Blake, T. R. Tracey, J. D. Walton, Jr. and S. H. Bomar, Jr. attended the NSF Semi-annual Solar Thermal Conversion Program Review in Minneapolis on March 10 through 14. A presentation was given describing the Martin Marietta-Georgia Tech program and further discussions took place concerning the current designs.

J. D. Walton, Jr. of Georgia Tech was in Europe during the period 28 March to 20 April 1974, for this and another research program. He met with representatives of the American Embassy and CNRS in Paris in efforts to arrange CNRS participation in this and subsequent solar thermal conversion programs. It was tentatively agreed that a new contract would be established between CNRS and Georgia Tech for this program only, and that CNRS participation in subsequent work would be under agreements between CNRS and the National Science Foundation.

Mr. Walton then went to the Solar Energy Laboratory at Odeillo, and while there prepared a draft of the technical portion of the proposed agreement between CNRS and Georgia Tech. Discussions of the technical aspects of this program were held among Professor Trombe and Messrs. Royere, Le Phat Vinh and Walton.

VII-128
2. Second Quarter (April 19 through July 18, 1974)

During this quarter there were numerous telephone contacts which will not be recorded here. These concerned particularly the arrangements for the subcontract between Georgia Tech and CNRS, coordination of a research agreement between NSF and CNRS, technical matters between Martin Marietta Corporation and Georgia Tech, and technical discussions between Georgia Tech and Babcock and Wilcox.

J. D. Walton, Jr. and S. H. Bomar, Jr. met with Dr. Herwig and Messrs. Kaplan, Speuler and Horowitz at NSF on May 7, 1974. The purpose of this meeting was to discuss the Martin Marietta-Georgia Tech program and photographs of the solar steam boiler at the University of Genoa.

A program meeting was held at Georgia Tech on May 13 and 14, 1974. F. A. Blake and T. R. Tracey of Martin Marietta and D. Van Fossen and R. Lee of Babcock and Wilcox participated in technical discussions on the bench model system design with Georgia Tech personnel.

Mr. Claude Royere of CNRS visited the United States during the month of June 1974. Royere, Walton and Bomar visited Martin Marietta in Denver on June 17 and 18, and NSF on June 21. The purpose of these visits was to acquaint Mr. Royere with program progress and to discuss CNRS participation in this and future cooperative programs.

T. R. Tracey and S. H. Bomar, Jr. visited the Martin Marietta-Baltimore plant on June 27, for the purpose of discussing construction of a bench model boiler system and cost estimates to be prepared by MMC-Baltimore.

3. Third Quarter (July 19 through October 18, 1974)

On August 4, 1974, a program briefing was conducted at the National
Science Foundation in Washington, D. C. Attendees were George Kaplan and Harold Spuhler of NSF, Reid Clausen, Floyd Blake and Tom Tracey of Martin Marietta, J. D. Walton and Steve Bomar of Georgia Tech, and Marvin Squires of White Sands Missile Range.

On August 6, F. A. Blake and T. R. Tracey of Martin Marietta, S. H. Bomar, Jr. and J. H. Murphy of Georgia Tech and D. Van Foseen, R. A. Lee, Dick Grams and H. P. Markant of Babcock and Wilcox met at the Research and Development Division of Babcock and Wilcox in Alliance, Ohio. At this meeting, the current bench model boiler design was reviewed and specific advice on technical points was received from B&W.

During the period August 24 through September 5, 1974, J. D. Walton, Jr. and S. H. Bomar, Jr. of Georgia Tech visited the CNRS Solar Energy Laboratory at Odeillo for several tasks in connection with this program. Contacts there were Professor F. Trombe, Mr. A. Le Phat Vinh and Mr. C. Royere of the Solar Energy Laboratory and Mr. M. Lagasse of CNRS in Paris. Detailed drawings of the focal building and adjacent work areas of interest to this program were prepared and interface requirements for utilities were established. The bench model boiler design existing at that time was discussed with Professor Trombe, Mr. Le Phat Vinh and Mr. Royere; they made technical comments on several design features. The proposed NSF-CNRS agreement for collaboration on Solar Thermal Conversion was discussed and understanding among the technical personnel of each organization was greatly improved.


VII-130
On September 9, 1974, Floyd Blake of Martin Marietta and Steve Bomar of Georgia Tech attended a National Science Foundation meeting at the offices of Aerospace Corporation in Los Angeles.

On September 11, 1974, Steve Bomar visited the Martin Marietta plant in Denver for project coordination.

On September 22 through 25, 1974, J. D. Walton and Steve Bomar of Georgia Tech participated in the NSF Semi-annual Program Review for Solar Thermal Conversion held at the Martin Marietta Corporation in Denver.

4. Fourth Quarter (October 19 through December 15, 1974)

On November 18 and 19, 1974, J. D. Walton, Steve Bomar, John Murphy and Tom Brown of Georgia Tech attended the Large Scale Solar Energy Test Facilities Seminar at New Mexico State University, Las Cruces, New Mexico. George Kaplan of NSF and Floyd Blake of Martin Marietta were also in attendance. Floyd Blake and Steve Bomar presented a paper at the meeting describing the bench model boiler. Informal discussions concerning the bench model took place between Martin Marietta/Georgia Tech team members on several occasions.
VIII. Meteorological Analysis
VIII. METEOROLOGICAL ANALYSIS

The major effort during this study is the analysis and modeling of direct solar insolation data taken with 2°-Collimation Eppley Normal Incidence pyrheliometers at Phoenix, Arizona, during 1962 (see Fig. VIII-1). The initial task is to determine the number of hours for which the direct insolation is above the threshold level for operating the solar plant at its rated output. The second task is to derive diffuse sunlight patterns for various times of day and times of year for the Phoenix site. This is accomplished by relating the normal-incidence insolation projected onto a horizontal surface to the total insolation on a horizontal surface observed by the U.S. Weather Bureau.

The two tasks are interrelated, in that, completing the threshold analysis requires filling in the weekend gaps, which means retrieving the normal-incidence component from the Weather Bureau data, based on the diffuse radiation pattern established for the corresponding time of year. Phoenix threshold-level insolation data are being generated from weekly sets of direct insolation curves similar to those in Fig. VIII-2.

Comparable threshold-level insolation hours are being extracted from the Aerospace Corporation direct insolation model for Inyokern, California, for 1962. Weekly plots have been prepared from the Aerospace Corporation-supplied magnetic tape. Figure VIII-3 compares the direct insolation plots for the first week of the year with the direct insolation in Phoenix, Arizona, for the same dates.

In support of the user analyses, total insolation data for Phoenix, Arizona, during 1973 was transformed into direct insolation data for the specific days examined for the user application analysis.

One modeling technique used to derive the direct insolation from the total insolation on a horizontal surface rests on the following relationship:

\[
\text{Direct Insolation} = \frac{\text{Total Horizontal Insolation} \ - \ \text{Diffuse Horizontal Insolation}}{\cos(\text{Solar Zenith Angle})}
\]
Fig. VIII-1
2°-Collimation Eppley Pyrheliometers at Phoenix, Arizona, Solar Test Facility, 1962
Fig. VIII-2
Direct Insolation Curves, Phoenix, Arizona,
January 1 thru 6, 1962
The critical parameter is the diffuse component of the insolation measured on the horizontal surface. Using the known direct and total insolation data to solve for the diffuse component of the Phoenix 1962 data, we will generate patterns for the diffuse insolation as functions of the time of day and time of year. Some results are already in hand, as illustrated for the early January days shown in Fig. VII-4. The absolute value of the diffuse insolation rises to a midday peak, and then falls in a somewhat symmetrical manner in the afternoon.

To add data from at least one other site for the diffuse insolation study, additional observations are being made at the Martin Marietta plant site southwest of Denver. The Atmospheric Sciences Mobile Laboratory (Fig. VII-5) was set up with total-insolation-monitoring Eppley pyranometers on the roof and tripod-mounted, normal-incidence tracking pyrheliometers south of the trailer. Figure VIII-6 shows the diffuse insolation and the diffuse/total ratio for two clear days in April 1974. The morning-to-midday rise was substantially less than that for Phoenix, as was the absolute value, probably attributable to the higher altitude of the Martin Marietta site. The diffuse/total pattern declined in the morning, but rose again in the afternoon. This was probably due to the near proximity of mountains well laden with melting snow.

While these studies are not complete, it is likely that each site has a unique diffuse/total pattern on a daily and annual basis, that must be established in order to model the Weather Bureau data in terms of direct insolation. This is also borne out by the pattern for Tucson, Arizona, on June 7, 1956 (Fig. VII-7).*

Completion of the data reduction, insolation modeling, and correlation with the Aerospace Corporation model results beyond that required to support the User Application analysis reported in Chapter III has been deferred due to program constraints.

REFERENCE

Fig. VIII-4 Diffuse Sunlight Model, Phoenix, Arizona, January 1962
Fig. VIII-5
Martin Marietta Atmospheric Sciences Mobile Laboratory, Monitoring Direct and Total Horizontal Solar Insolation
Fig. VIII-6.
Diffuse Solar Insolation, Martin Marietta Corporation, Denver, Colorado, April 1974

VIII-8
Fig. VIII-7
Diffuse Solar Insolation, Tucson, Arizona, June 7, 1956 (Ref IV-1)
IX. Long Range Plan
Study - Subsystems
Technology Research
An added task was established to define requirements, goals, and supporting rationale for the long range development program of solar power subsystem technologies. The results of this study were reported to the review committee on solar power assembled at the invitation of Mr. George Kaplan, at the Aerospace Headquarters in Los Angeles September 9, 1974. The material presented at this meeting is included in the following pages of this chapter, and includes (1) a preliminary configuration of the "Proof-of-Concept-Experiment," (2) a preliminary configuration of a 5 MWth solar facility for subsystem technology research, (3) requirements for a small heliostat array program, (4) requirements for a central receiver technology program, and (5) requirements for an energy storage study leading to the subsystem technology program.
Rationale for Choice of P. O. C. E. Configuration

1. Most Fully Demonstrates Full-Scale Equipment of Full-Size Solar Power Plant

   a. Solar Collection - Concentration Field of 1840 Heliostats = Module of 100 MWe Plant
   b. Tower-Mounted Boiler/Superheater = Module of 100 MWe Plant
   c. Turbine-Generator Power Plant = 1/10 Scale of 100 MWe Plant
Field Contours and Laboratory/Tower, 1-5 MW\textsubscript{TH} Solar Facility

5 MW\textsubscript{TH} SOLAR TEST FACILITY

Objectives:
- Provide Modular Elements of Facility to Enable Testing at the 1 MW\textsubscript{TH} Level by Sept. 1976.

Key Features Required:
- Flexibility of Solar Collection Heliostats to Provide Both "Concentrated" and "1 Sun" Energy.
- Flexibility of Solar Collection Field to Provide "North Side Only" and "Surrounding of Central Tower" Type Concentration at Both 1 MW\textsubscript{TH} and 5 MW\textsubscript{TH} Levels.
- "Open Focal Zone" Adaptable to all Candidate Receivers.
5 MW$_{th}$ SOLAR FACILITY - KEY SPECIFICATIONS 1 of 2

| Site Description: | 1000 ft East-West by 910 ft North-South (910,000 ft$^2$, 20.9 acres) Northern Third of Site on Slope to South to Accommodate Terracing. |
| Laboratory Building: | 3-Story (45 ft) Height 140 x 120-ft Plan Area |
| Tower Structure: | Gantry Structure from Center of Building, Rising 200 ft from Base Plane |
| Tower Features: | Personnel Elevator, 10-ton Overhead Crane |
| No. of Heliostats: | 310 Units of 400 ft$^2$ |
| No. of Heliostat Foundations: | 546 |
| Heliostats for 1 MW$_{th}$ Module: | 68 Units of 400 ft$^2$ (1=1.3 MW$_{th}$ in All-North Configuration) (1=1.1 MW$_{th}$ in Surrounding Configuration) |
| Thermal Energy at Focal Zone: | 5 MW$_{th}$ with 861 W/m$^2$ Solar Insolation |
| Focal Length Adjustment: | Flat to 248 ft Focal Length |
| Focal Zone Clear Area (Except Corner Posts): | 45 x 45-ft Base x 30-ft Height |
| Portability: | Heliostats Shall Be Designed for Repositioning by Use of Traveling Crane as Straightforward Maintenance Process |
Nine Mirror Heliostat Configuration - 7x7 ft Mirrors

5MW<sub>th</sub> SOLAR FACILITY - RATIONALE

Why Combined Flat and Sloping Site?

a) To Provide Flexibility in Collector Configurations Enabling Operation As "North-Only" or "Surrounding Field" Collector

b) Slope of North Field plus Tower Height Enables 5 MW<sub>th</sub> to be Concentrated On or Through a 10x10-ft Focal Zone, (Average Flux up to 0.54 MW/ft<sup>2</sup>; 170,850 Btu/ft<sup>2</sup>-hr)

Why Open Tower Structure with Overhead Crane?

a) Precedent Set on Launch Pad Vehicle Assembly Areas. More Economical Lifting Capability for Heavy Receiver Units Than Elevator.

b) Results in Minimum Obstruction to Focal Zone for Surrounding Field Testing.

Why 310 Heliostats of 400 ft<sup>2</sup> Each?

a) Although Fully Flexible Facility has 546 Heliostat Positions, only 304 and 310 Are Used for the Two Primary Configurations.

Equinoval Capacity of 2 Modes.

North Only: 6.03 MW<sub>th</sub> (310 Mirrors)
1.32 MW<sub>th</sub> (68 Mirrors)

Surrounding Field: 5.01 MW<sub>th</sub> (304 Mirrors)
1.13 MW<sub>th</sub> (68 Mirrors)

Why Focal Length Adjustibility?

a) Enables Attainment of High Fluxes of Full-Scale Receiver Designs without Reducing Scale of Heliostats or Increasing Number Required.

b) Flat Mode Will Permit Testing of Large Exterior Receivers to Define Thermal Performance Although Flux Levels Are Reduced.
5 MW<sub>th</sub> SOLAR TEST FACILITY

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<th>FY 78</th>
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<td>I. Design 5 MW&lt;sub&gt;th&lt;/sub&gt; Solar Facility</td>
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<td>1.1 Preliminary Design</td>
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<td>1.2 Detail Design Full Facility</td>
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<td>1.3 Detail Design 1 MW&lt;sub&gt;th&lt;/sub&gt; Module</td>
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<td>II. Construction and Checkout</td>
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<td>2.1 Site Selection &amp; Negotiations</td>
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<td>2.2 Material Procurement</td>
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<td>2.3 Construction of Lab/Tower</td>
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<td>2.4 Construction of Terrace/Foundations</td>
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<tr>
<td>2.5 Procurement of Heliostats</td>
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<tr>
<td>III. Solar Component Testing</td>
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</tr>
<tr>
<td>3.1 1 MW&lt;sub&gt;th&lt;/sub&gt; Receivers</td>
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<tr>
<td>3.2 5 MW&lt;sub&gt;th&lt;/sub&gt; Receivers</td>
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<td>3.3 Energy Storage Experiments</td>
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<td>3.4 Power Generation Experiments</td>
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(During 1978)
Full-Size Heliostats Should Be Used in All Four Applications. Since the Heliostat Structure and Tracking Control Represent Approximately 40% of the Capital Cost of a Solar Power System, We Need to Start Down the Learning Curve As Early As Possible. Competing Programs from Several Organizations Can be Carried Up Through the Proof-of-Concept Experiment.

The Commercial Plant Will Probably Use the Demonstrated Optimum Design of the POCE.

High Unit Cost of High-Pressure High-Temperature Energy Conversion Equipment Dictates that Maximum Experience Be Obtained at Small Scales prior to Full-Scale Commitment of POCE Unit.

Relatively Even Progression from 1 to 5 to 30 MWth Should Yield Efficient Minimum-Risk Development.

Storage Development Status Ranges from On-Hand to Conceptual.

Highest Priority Should Be Given Storage Needed for Short-Term Stability Since Most Utilities Have Unused Capacity Peaking Units in Excess of Capacity of Early Solar Plants.

KEY SPECIFICATIONS - SMALL HELIOSTAT ARRAY DEVELOPMENT -

No. Heliostats = 12

Size of Heliostats = 6 x 6 m (19.7 x 19.7 ft) Min
= 7 x 7 m (22.9 x 22.9 ft) Max

Reflector Material = Identical with Each Contractor's Selection
(Second Surface Glass Mirror for Martin Marietta)

Special Feature = Provide Method of Adjusting Curvature of
Heliostat for Focal Lengths of 150 to 205 ft

Tracker Configuration = Identical with Each Contractor's Selection
Azimuth - Elevation for Martin Marietta

Tracking Accuracy = +0.33 milliradian, 0 to 5 mph Wind
= ±1 milliradian, 5 to 30 mph Wind

Safety Provisions = Override Based on Wind Speeds Above 30 mph,
to Position Mirrors' Edge into Wind.

Secondary Override Initiated by Operator for Vertical
Positioning, if Hail Danger Exists.
RATIONALE FOR SMALL HELIOSTAT ARRAY SPECIFICATIONS

Why 12 Heliostats?

a) Urgent Need to Establish Technology for Reliable, Accurately Tracked Field of Heliostats,
b) Urgent Need to Obtain Economic Data and “Learning Curve” Pattern,
c) Total Energy Available Would Enable “Cavity Performance” to be Evaluated on Bench Model Boiler/Superheater if Backup Position is Needed.

Why 6 x 6 m to 7 x 7 m Mirror Size?

a) Provides Direct Development of Mirrors for Full-Scale Heliostats Without Scaling,
b) Provides Realistic Structure for Economic Evaluation of Full-Scale System.

Why Reflector Material Identical with Contractor’s Selection?

a) The Significant Performance Differences Between Candidate Reflector Materials Need to be Defined on Full-Scale Equipment of Parallel Programs.

Why Second Surface Glass Mirrors (Martin Marietta Selection)?

a) Currently Mass Produced at Price Level of $1.00 per Square Foot, with Prospects of Cost Reduction,
b) Demonstrated Long Life Without Significant Degradation, (Montlouis Mirrors > 24 Years’ Exposure).

Why Curvature?

a) Provides Experience with Adjustable Curvature Mechanism,
b) Enables Small-Size Calorimeter,
c) Enables Up to 1800:1 Concentration for Short Test Periods.

Why Tracker Configuration Identical with Contractors’ Selection?

a) To Establish a Cost Comparison Base Indicative of Likely Cost Relationship in Large Production.

Why Azimuth-Elevation (Martin Marietta Selection)?

a) Direct Experience with Both Equatorial and Azimuth-Elevation Trackers During Solar Thermionics Programs of 1961-1965 Highlighted Basic Advantages of Azimuth-Elevation Mount:
1) Ease of Equipment Mounting and Servicing;
2) Flexibility of Positioning for Safety Reasons.

Why Tracking Accuracy +0.33 Milliradians for 0 to 5 mph Wind?

a) Ample Precedent for Attaining Accuracies Well Inside 1 Minute of arc with Simple Sensors has been Established. This is One of Direct Inheritances of Space Program.

Why Tracking Accuracy + 1 Milliradian for 5 to 30 mph Wind?

a) Increase Tolerance for Windy Conditions to Permit Structural Deflections and Backlash Oscillations without Control Hunting.

Why Shut Down Override for Winds Above 30 mph?

a) In Vast Majority of Time when Operable Sunlight Exists, Wind Conditions are Minimal. Therefore this Override will have Insignificant Effect on Attainable Operating Time,
b) Permits Safe Use of Lighter Weight Structure with Resulting Cost Advantage and Minimal Impact of Operation.

Why Manually Actuated Override for Hail Danger?

a) Hail is Relatively Rare Phenomenon in Southwest U. S. and Occurs During “Downtime”;
b) “Hail Safe” Position is Mirror in Vertical Plane Rather than Horizontal “Wind Safe” Position.
I. Heliostat Design
   1.1 Structural Design
   1.2 Control System Design
   1.3 Solar Reflector Design
   1.4 Assembly Tooling Design

II. Fabrication - Engineering Prototype
   2.1 Material Procurement
   2.2 Structure Fabrication
   2.3 Control System Fabrication
   2.4 Reflector Fabrication

III. Test Engineering Prototype
   3.1 Checkout/Optical Test
   3.2 Thermal Efficiency Test
   3.3 Flux Mapping Test

IV. Small Array Heliostats
   4.1 Fabrication
   4.2 Checkout & Test
BENCH MODEL - CAVITY STEAM GENERATOR - SIDE ELEVATION

1 - 5 MW\textsubscript{th} RECEIVER TECHNOLOGY - KEY SPECIFICATIONS

Scope: Component Development of Central Receivers for Solar Thermal Power Plants based on Experimental Testing of Scaled-Down Units Capable of Operating with 1 and 5 MW\textsubscript{th} Concentrated Solar Energy.

1 MW\textsubscript{th} "Bench Model" Boiler/Superheater

- Input Capacity: 1 MW\textsubscript{th} (3,413,000 Btu/hr)
- Thermal Efficiency Goal: 0.88
- Output Capacity: 2225 lb of steam/hr
- Steam Conditions: 950°F\textsubscript{min}, 1250 psig ±50
- Feedwater (Condenser Discharge): < 185°F, 0.3 - 17 psia
- Enthalpy Gain: 1350 Btu/lb
- Configuration: Cavity, Exterior Truncated Cone, Exterior Hemisphere, or Cruciform Depending on Individual Contractor
1 - 5 MW<sub>th</sub> RECEIVER TECHNOLOGY - KEY SPECIFICATIONS

Scaling Criteria: Basic Thermal Parameters To Be Simulated to Maximum Extent Feasible

- Peak Solar Flux Levels of Full-Scale Plant.

Test Program Stages:

1) Cold Hydrostatic Pressure Tests and Leak Check
2) Progressive Functional Performance Testing Bringing the Unit Up to Rated Output in Series of Thermal Energy Steps.
3) Performance Refinement Testing with Range of Design Features Aimed at Attaining Maximum Efficiency
4) Endurance Cycling for Period Equivalent to One Year's Operation

5 MW<sub>th</sub> "Pilot Scale Model" Receiver

Input Capacity: 5 MW<sub>th</sub> (17,065,000 Btu/hr)
Thermal Efficiency: 0.88 Minimum
Output Capacity: 11,125 lb of steam/hr
Steam Conditions: 950°F min, 1250 psig +50
Feedwater (Condenser Discharge): < 150°F, 0.3 - 17 psia
Enthalpy Gain: 1350 Btu/lb
Configuration: Scale of Baseline Design for Proof of Concept Experiment Receiver
1 - 5 MW \textsubscript{th} RECEIVER TECHNOLOGY - RATIONALE

Why 1 MW \textsubscript{th} and 5 MW \textsubscript{th} Development Stages?

a) 1 MW \textsubscript{th} Sizing Permits Earliest and Most Economic Receiver Enabling "Design to Test" Cycle Times Approaching One Year.
b) 1 MW \textsubscript{th} Size Is Adequate to Obtain Thermal Performance of Receiver, and Enables Parametric Experiments of Key Elements.
c) 1 MW \textsubscript{th} Size Enables Adaptation to C. N. R. S. Solar Furnace for Functional and Performance Refinement Testing.
d) 5 MW \textsubscript{th} Size Enables Definition of Problems Attributable to 5X Scaling, Which Will Provide Guidance for Proof-of-Concept Design Finalization

Why 88 Percent Receiver Thermal Efficiency?


Why 950°F, 1250 psi Outlet Steam Conditions?

a) Proof of Performance Regarding Steam Quality for 35% Efficiency Turbine Generator Will be Demonstrated.

Why < 185°F, 0.3 - 17 psia Condenser Conditions?


Why Variety of Configurations?

a) It Is Important to Establish Correlation between Hardware and Design Analysis for Promising Configurations and Empirically Establish Relative Performance Advantages Attributable to Geometry.

Why Scale to Preserve Thermal Parameters?

a) Radiation Scales Directly and Therefore Scaling Based on Geometric Area and Thermal Properties will Yield Reliable Combined Effects Data.

Why Four Test Program Stages?

a) Static Testing Reveals Fabrication Defects at Early Stage Enabling Minimum-Impact Corrective Action.
c) Performance Refinement Is Essential Stage. (Many Programs under Pressure of Schedule Have Omitted This Stage Only to Have Endurance Testing Prematurely Cut Off by Correctable Defect).
d) Endurance to Establish Base of Operating Data and Uncover More Subtle Problems That Are Cycle or Time Dependent.
1 - 5 MW

IEC RECEIVER TECHNOLOGY

<table>
<thead>
<tr>
<th>Quarter</th>
<th>FY 75</th>
<th>FY 76</th>
<th>FY 77</th>
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<td>JFM AMJ JAS OND</td>
<td>JFM AMJ JAS OND</td>
<td>JFM AMJ JAS OND</td>
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<tr>
<td>I. Receiver Design 1 MW</td>
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<tr>
<td>1.1 Solar Energy Heat Exchanger</td>
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<td>1.2 Steam Generation Control System</td>
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<td>II. Fabrication of 1 MW Receiver</td>
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<td>2.1 Material Procurement</td>
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<td>2.2 Fabrication of Receiver</td>
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<td>2.3 Fabrication of Support Inst</td>
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<td>III. Test of 1 MW Receiver</td>
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<td>3.1 Check Out/Functional</td>
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<td>3.2 Perf. Refinement</td>
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<td>3.3 Endurance Cycling</td>
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<tr>
<td>IV. 5 MW_{th} Receiver Design</td>
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<tr>
<td>V. 5 MW_{th} Receiver Fabrication</td>
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<tr>
<td>VI. 5 MW_{th} Receiver Test</td>
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</tbody>
</table>
General: The "Thermal Storage" Item in the 5-Year Plan Should be Opened up to Cover the Complete Topic of "Energy Storage" for Solar Thermal Power Plants.


Storage for Current Technology Electrical Power Is Hydroelectric.

Other Storage Candidates Are in Very Early Conceptual Stages and Are in Need of Analysis and Preliminary Design. These include Flywheels and a Broad Range of Thermal Energy Storage Methods and Gaseous Fuel Generation.

First Solar Plants May Well Utilize Standby Fuels until Suitable Storage Is Developed.
ENERGY STORAGE - TECHNOLOGY TRANSFERABILITY

Storage Technique: Pumped Water Storage

Technology Presently Exists for a Pumped Water Storage System Capable of Generating 300 MW for 6 Continuous Hours. Examples of Pumped Water Storage Facilities of this Size Presently in Operation are:

- Cabin Creek Facility Near Georgetown, Colorado, Operated by Public Service Company of Colorado which Produces 300,000 kw for 6 hours.
- Taum Sauk Facility near Lesterville, Missouri, operated by the Union Electric Company which Produces 350,000 kw for over 6 hours.

ENERGY STORAGE - RECOMMENDED SPECIFICATIONS

Storage Technique: Pumped Water Storage

- Capacity of Storage System: 100 MW for up to 6 hours of Continuous Operation
- Storage Temperature and Pressure: Ambient Temperature and Pressure
- Storage Medium: Approximately 550 acre-feet of Water Stored at an Altitude such that the Water Pressure at the Power Station is in the Range of 400-500 psi.
- Rates of Introducing and Extracting Energy: Energy Introduced at the Rate of about 600,000 kwh in 9 hours and Extracted at the rate of 600,000 kwh in 6 hours.
- Energy Loss Rates: Losses due to Evaporation of Stored Water are Dependent on Local Climate Conditions.
- Energy Efficiency: The Efficiency of this Storage Technique is about 67%.
**ENERGY STORAGE - TECHNOLOGY TRANSFERABILITY**

Storage Technique: Flywheel Storage  
Fibers Developed for Aerospace Needs Have the Properties Necessary for Development of Flywheels Capable of Storing Large Amounts of Energy. These Properties are Densities Which are 4 to 6 Times Lower than Steel and Strengths in Tension which are Stronger than the Strongest Steel. A Development Program would be Required Prior to Fabricating Flywheels Using Fibers in Order to Select the Optimum Fibers and Determine the Optimum Flywheel Physical Configuration. The Fibers are Also Much Safer for Flywheel Use than Steel since if they are Overstressed, They Fail by Shredding or Turning into Powder Rather than by Breaking into Large Chunks as is the Case with Flywheels Made of Steel.

**ENERGY STORAGE - RECOMMENDED SPECIFICATIONS**

<table>
<thead>
<tr>
<th>Storage Technique:</th>
<th>Flywheel Storage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity of Storage System:</td>
<td>100 MW, for up to 6 Hours of Continuous Operation</td>
</tr>
<tr>
<td>Storage Temperature and Pressure:</td>
<td>The Flywheel can be Operated at Ambient Temperature and Pressure. However, Operating the Flywheel and Motor/Generator in an Inert Gas (Hydrogen or Helium) Atmosphere Below Atmospheric Pressure Minimizes Air Friction Losses.</td>
</tr>
<tr>
<td>Storage Medium:</td>
<td>Thirty (30) Flywheels Each Capable of Storing 20 mWh and Generating 3.5 MW, could be Used. Each of the Flywheels would be Made of a Fiber such as Fused Silica and would be about 15 feet in Diameter, Weigh about 200 Tons and Rotate at About 3,500 Revolutions per minute.</td>
</tr>
<tr>
<td>Rates of Introducing and Extracting Energy:</td>
<td>Energy Could be Introduced at a Rate of 600 mWh in about 6.5 hours and Extracted at a rate of 600 mWh in about 6 hours.</td>
</tr>
<tr>
<td>Energy Loss Rates:</td>
<td>Friction Losses can be Minimized by Installing the Flywheel and Motor/Generator in an Inert Gas Atmosphere at Below Atmospheric Pressure and Other Design Considerations.</td>
</tr>
<tr>
<td>Energy Efficiency:</td>
<td>The Efficiency (Output Over Input) of the Flywheel System is About 93%.</td>
</tr>
</tbody>
</table>
### ENERGY STORAGE - TECHNOLOGY TRANSFERABILITY

**Storage Technique:** Hydrogen  
Technology and Facilities Presently Exist to Produce Very Large Quantities of Hydrogen (Billions of Cubic Feet per Year) and to Store and Transport the Hydrogen. There Should Be No Significant Difficulty in Transferring This Technology to the Solar Power System.

### ENERGY STORAGE - RECOMMENDED SPECIFICATIONS

<table>
<thead>
<tr>
<th>Storage Technique:</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Capacity of Storage System:</strong></td>
<td>100 MWe for up to 1/2 Hour of Operation</td>
</tr>
<tr>
<td><strong>Storage Temperature and Pressure:</strong></td>
<td>Stored at Ambient Temperature. Storage Pressure Would Be Ambient If Stored above the Ground and would Probably be Stored in a Structure That Expands Upward As It Is Fitted from the Bottom and Collapses Downward as the Weight of the Structure Forces Hydrogen out of the Bottom As Hydrogen is Released. If Underground Storage Is Used, the Hydrogen Would Be Stored at about 4 to 5 Atmospheres of Pressure.</td>
</tr>
</tbody>
</table>

**Storage Medium:** Storage Medium Is Probably Gaseous Hydrogen If Stored Aboveground, In Underground Caves, or Depleted Gas Fields. Storage Medium Could Be Liquid Hydrogen in Aboveground Tanks.

**Energy Efficiency:** Efficiency of Hydrogen Generation on a Large Scale by Electrolysis Commercially Today is about 67%.

**Rates of Introducing and Extracting Energy:** Energy is Introduced into the System by Refilling the Storage Area with Hydrogen at a Rate Limited Primarily by the Size of the Fill Pipe. Energy is Extracted at the Rate of 1,926,500 Cubic Feet (or less) of Hydrogen per Half Hour. This Equals about 529,767,000 Btu per half hour.

**Energy Loss Rates:** It Requires about 250 MWh of Energy To Produce Enough Hydrogen To Generate 50 MWh of Electrical Energy Using the Steam Boiler Technique. The Total System Efficiency Is Therefore about 20%. This Is Based on an Efficiency of Generating Hydrogen by Electrolysis of 67%, Which Is the Approximate Efficiency of Electrolysis of Water of a Large Scale Using Today's Existing Facilities. Although This Efficiency Could Theoretically Approach 100% by Using Such Techniques as Very Low Current Densities, Very Large Plate Size, and by Reducing Space between Electrodes and Catalyzing Electrodes, Development Would Be Required and Cost Would Be Significantly Increased. Hydrogen Is Lost from Storage about 2.8 Times As Fast As Natural Gas from a Volume Standpoint and about the Same Rate As Natural Gas from an Energy Standpoint. The Actual Loss Rate Will Be Determined by the Physical Characteristics of the Storage Container.
**ENERGY STORAGE - TECHNOLOGY TRANSFERABILITY**

Hitec Has Been Used in Heaters as Large as 15,000,000 Btu per hour at Salt Temperatures up to 850°F and Efficiencies of over 65%.

Considerable Development Would Be Required in Transferring This Technology to the Solar Energy Boiler System but the Concept Appears Feasible.

**ENERGY STORAGE - SPECIFICATION REQUIREMENTS**

<table>
<thead>
<tr>
<th>Storage Technique:</th>
<th>Eutectic Salt (Hitec is Dupont Trade Name)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity of Storage System:</td>
<td>6.5 x 10^6 Cubic Feet of Hitec Eutectic Salt to Store about 6350 x 10^6 Btu.</td>
</tr>
<tr>
<td>Storage Temperature and Pressure:</td>
<td>Hitec Freezes at 288°F and Can Be Heated to about 900°F. Storage Would Be at Ambient Pressure.</td>
</tr>
<tr>
<td>Storage Medium:</td>
<td>Molten Eutectic Salt Such as Dupont Hitec.</td>
</tr>
<tr>
<td>Rate at Which Energy Is Introduced and Extracted:</td>
<td>Energy Would Be Removed at the Rate of 6350 x 10^6 Btu in 6 hours, and Introduced at the Rate of 6350 x 10^6 Btu in about 9 hours.</td>
</tr>
<tr>
<td>Energy Loss Rate:</td>
<td>0.33 Btu (Hour) (ft^2) (°F/ft)</td>
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<td>Efficiency:</td>
<td>Greater than 65%</td>
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# ENERGY STORAGE SUBSYSTEMS TECHNOLOGY

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<tr>
<td>Storage System Study</td>
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<td>Identification of Candidate Storage Techniques for Solar Power</td>
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<td>Comparative Analysis of Storage Options</td>
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<td>Design of 6 Hr - 100 MW&lt;sub&gt;e&lt;/sub&gt; Storage</td>
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<td>Preliminary Design 100 MW&lt;sub&gt;e&lt;/sub&gt; Storage Subsystem</td>
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<td>Design of 1 MW&lt;sub&gt;th&lt;/sub&gt; Storage Experiment</td>
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<td>2.3</td>
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<td>Design of 5 MW&lt;sub&gt;th&lt;/sub&gt; Storage Experiment</td>
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<td>III.</td>
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<td>Fabrication of 1 MW&lt;sub&gt;th&lt;/sub&gt; Exp.</td>
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<td>Materials Procurement</td>
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<td>3.3</td>
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<td>Check Out Testing of 1 MW&lt;sub&gt;th&lt;/sub&gt; Exp</td>
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<td>Solar Testing Storage Subsystem</td>
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Appendix A
Trip Report, Application of Steam Turbine-Generators to Solar Power Plants
APPENDIX A—TRIP REPORT, APPLICATION OF STEAM TURBINE-GENERATORS TO SOLAR POWER PLANTS

TO: F. Blake
FROM: T. Tracey
CC: M. T. Howerton, J. Kidd, R. Farrell, R. Clausen

SUMMARY

The primary objectives of the trip were to obtain performance data on candidate steam turbines for a 100,000-kW solar power plant and to assess the potential problems of cyclic operation of turbines in a solar power plant application.

There has been considerable emphasis on the design of steam turbines for cyclic operation in recent years for peaking service. Thermal transients have resulted in cracking of high pressure cases and rotors, particularly at stress concentrations, and rotor rubbing. Many corrective actions have been taken to eliminate the problems including limiting inlet temperature and pressure, use of more ductile materials, thermally insulating to minimize temperature drop during an overnight shutdown, and revision of startup procedures to minimize thermal stresses. Turbines (up to 150,000 kW) have been built for cyclic operation with requirements very similar to what would be required of a turbine for a solar plant. These requirements include 6,000 to 8,000 starts, and 6,000 to 20,000 rapid load changes (depending on magnitude of load change) over a 20-year life. Startup after an overnight shutdown can be accomplished in 25 minutes.

Performance of candidate turbines is summarized in Table A-1. The heat rates include generator exciter power but no auxiliaries. There is a 10% improvement in heat rate in going from four 25,000-kW turbines to a single 100,000-kW turbine. Reheat improves performance about 6%. The sensitivity of heat rate to initial pressure and temperature and backpressure is given in Figure A-1.
Table A-1 Performance Summary

<table>
<thead>
<tr>
<th>Power Rating, kW</th>
<th>25,000</th>
<th>50,000</th>
<th>100,000</th>
<th>100,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure, psig</td>
<td>850</td>
<td>1,250</td>
<td>1,250</td>
<td>1,450</td>
</tr>
<tr>
<td>Inlet Temperature, °F</td>
<td>900</td>
<td>950</td>
<td>950</td>
<td>1,000/1,000</td>
</tr>
<tr>
<td>Backpressure, in. Hg</td>
<td>$3\frac{1}{2}$</td>
<td>$3\frac{1}{2}$</td>
<td>$3\frac{1}{2}$</td>
<td>$3\frac{1}{2}$</td>
</tr>
<tr>
<td>No. of Final Stages/Bucket</td>
<td>1/12.8 in.</td>
<td>1/20 in.</td>
<td>2/20 in.</td>
<td>2/20 in.</td>
</tr>
<tr>
<td>Length</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No. of Feed-water Heaters</td>
<td>4</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Full-Load Throttle Flow, lb/hr</td>
<td>226,850</td>
<td>436,915</td>
<td>895,721</td>
<td>666,704</td>
</tr>
<tr>
<td>Full-Load Condenser Flow, lb/hr</td>
<td>175,150</td>
<td>316,700</td>
<td>622,727</td>
<td>497,247</td>
</tr>
<tr>
<td>Feed-water Temperature, °F</td>
<td>376</td>
<td>431</td>
<td>439</td>
<td>425</td>
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<tr>
<td>Gross Heat Rate, Btu/kW-hr</td>
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<tr>
<td>100%</td>
<td>9,980</td>
<td>9,210</td>
<td>9,077</td>
<td>8,324</td>
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<tr>
<td>80%</td>
<td>10,010</td>
<td>9,260</td>
<td>9,131</td>
<td>8,432</td>
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<tr>
<td>60%</td>
<td>10,240</td>
<td>9,470</td>
<td>9,344</td>
<td>8,682</td>
</tr>
<tr>
<td>40%</td>
<td>10,790</td>
<td>9,970</td>
<td>9,838</td>
<td>9,159</td>
</tr>
</tbody>
</table>

Figure A-1
DISCUSSION

The subject meeting was held on March 5, 1974 at the General Electric Medium Steam Turbine Department, Lynn, Mass. Those present at the meeting, in addition to the writer, were:

R. E. Speery - Manager of Engineering
G. Schofield - General Manager - Part Time
C. Hansell - Sales
A. P. Rendine - Mechanical Design
C. N. Cannon - Performance

CYCLIC OPERATION OF STEAM TURBINES

Background Experience - GE has steam turbines over 400 mW that are started every day. Some older machines not designed for cycling had problems including cracked cases and rotors. However, many older turbines have been successfully cycled for years, including one that has seen over 2700 cycles. A list of GE large steam turbines used for peaking service is included. Steam turbines have been designed to utilize the exhaust heat from gas turbines requiring daily starting to full load in about 25 minutes. These turbines will be started up within the next six months.

Potential Problems Resulting from Cyclic Operation - The thick-walled high-pressure case can crack if large radial temperature gradients are imposed during thermal transients. The heavy rotors tend to crack, particularly at stress concentrations as a result of cyclic thermal stresses. Finally, rotor rubbing can result from differential thermal expansion between the case and rotor.

Design Features Used to Minimize Potential Problems - GE has incorporated the following design features in turbines that are to be used in cyclic operation:

1) Limit the inlet temperature to 950°F, primarily to allow the use of a more ductile steel;
2) Limit the pressure to below 1800 psig and preferably below 1450 psig;
3) Thermally insulate the turbines to limit the temperature drop overnight to about 300°F;
4) Match the steam temperature to the turbine temperature during startup;
5) Use a modified control valve design that admits steam over the entire annulus during the start transient. Conventional control valving admits steam over segments of the annulus, which aggravates the thermal stresses;
6) Modify casing and rotor designs to minimize thermal stresses and stress concentrations.

Typical Requirements for Cyclic Turbines - The following definitions are used by GE in defining cyclic requirements:

- **Hot Start** - Start after a shutdown less than 12 hr.
- **Warm Start** - Start after a shutdown less than 72 hr.
- **Cold Start** - Start after a shutdown over 72 hr.
- **Major Load Change** - 80%;
- **Moderate Load Change** - 40%;
- **Minor Load Change** - Less than 20%.

GE has built a series of steam turbines for cyclic operation with gas turbines for "peaking" operation with the following requirements:

- **Size** ∼ 150,000 kW
- **Steam Conditions** 1250 psig, 950°F
- **Life** 20 Years

<table>
<thead>
<tr>
<th>Type of Transient</th>
<th>Number</th>
<th>Time to Full Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Cycles</td>
<td>6,000</td>
<td>25 minutes</td>
</tr>
<tr>
<td>Warm Cycles</td>
<td>1,000</td>
<td>40 minutes</td>
</tr>
<tr>
<td>Cold Cycles</td>
<td>60</td>
<td>90 minutes</td>
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<td>Load Changes</td>
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<td>Major</td>
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<td></td>
</tr>
<tr>
<td>Moderate</td>
<td>6,000</td>
<td></td>
</tr>
<tr>
<td>Minor</td>
<td>20,000</td>
<td></td>
</tr>
</tbody>
</table>

These requirements are very compatible with a typical solar power plant. The turbines described above will be started up within the next six months.
PERFORMANCE OF CANDIDATE STEAM TURBINES

Table A-1 summarizes the performance of four candidate turbine-generator designs. The last column shows the performance of a reheat design at 1000°F. As discussed earlier, the temperature should be limited to 950°F for cyclic operation. The heat rate for a 50°F reduction in temperature is 1½%. The heat rates given are "gross" values that do not include auxiliary power (feed pumps, circulating pumps, fans, etc) but do include generator exciter power.

There is a very significant increase in performance (~10%) in going from the 25,000 kW (850 psig, 900°F) to the 100,000 kW (1250 psig, 950°F) turbine. Reheat improves performance about 6%.

Figure A-1 shows the sensitivity of heat rate to initial pressure and temperature and backpressure. The sensitivity to backpressure is particularly important in a solar plant located in a desert area where cooling water is expensive and the ambient air temperature is high.

LIST OF DATA RECEIVED FROM GENERAL ELECTRIC

A list of information received from GE follows. This information can be obtained from the writer.

1. Candidate Solar Plant Steam Turbine - Generator Performance Table.
2. Curves Showing Effect of Initial Temperature, Pressure and Backpressure on Heat Rate.
3. Typical Detailed Heat Balance Diagram for 25-mW Unit.
4. Typical Detailed Heat Balance Diagram for 44-mW Unit.
5. Typical Detailed Heat Balance Diagram for 116-mW Unit.
9. A List of Steam "Peakers" Sold to Date by GE.
10. A Package of Descriptive Material Showing Cutaways of Steam Turbines from 20 to 185 mW, Controls and Generator Design Features.

11. A Package to Estimate Turbine-Generator Cost.

12. A Title Page for a Reference "Dry-Type" Cooling Tower for Thermal Electric Generation.

In closing, I would like to point out that the personnel at GE were extremely cooperative and helpful.

T. R. Tracey
Thermophysics
Solar Power Plant Study

TRT/kh
Appendix B
Detail Design of the Bench Model Cavity Receiver Steam Generator
Figures B-1 thru B-13 show the detail design drawings of the cavity. Section I of the ASME Boiler Code (fired pressure vessels) has been applied wherever applicable. We have minimized thermal stresses throughout the design by minimizing constraints on the heated sections. Mechanical fittings have been provided to allow disassembly if necessary for shipment or to make modifications if required.

Figure B-1 shows the rear wall superheater sections. The tubes are supported by pins through the welded-on clips. The tube is constrained longitudinally only at the top pin. There are no moment constraints at the attachment points. The upper clip (detail A) attached to the bottom side of the channel is on alternate tubes. The clip on top of the channel is on the remaining tubes (i.e., there is only one clip at the top of each tube). This method of tube attachment is used throughout the design.

Figure B-2 shows the design at the boiler sections. Two headers are used at the top and bottom. Alternate tubes are welded to common headers to allow room to make the welds into the headers. There are only two tube details in the entire boiler, which greatly simplifies manufacturing.

Figure B-3 shows the boiler upper headers and the interconnecting piping on the top. Graylock fittings are used on all the pipes (including the headers) to allow disassembly. Graylock fittings are used instead of flanged fittings because they require much less space and can withstand thermal cycling without leaking. The lower header configuration is shown in Figure B-4.

The bottom panel preheater section is shown in Figure B-5. The tubes are supported in a manner similar to the superheater section to minimize constraints, which will result in minimum thermal stresses.

The design of the upper panel containing both preheater and superheater sections is shown in Figure B-6.
The steam drum design is shown in Figures B-7, B-8, and B-9. The drum is fabricated from a 20-inch diameter 1.75-inch thick seamless pipe with matching hemispherical domes welded to each end. The drum is fastened down at only one of the mounting points (detail B) and is free to slide on the other. The baffle (detail A, Fig. B-7) will be completely closed off at the downcomer end and open at the other end. The feed-water inlet (detail G, Fig. B-8) is thermally isolated to minimize the thermal stresses in the drum caused by the temperature difference between the drum and the feed water entering. Penetrations are provided in the drum for:

1) Eight boiler riser manifolds; 5) One sight glass;
2) One downcomer; 6) One steam outlet;
3) Two relief valves; 7) One ΔP sensor.
4) One feed-water inlet;

Typical structural detail is shown in Figures B-10 and B-11. The inner panels are attached to the outside superstructure at a single point near the top. The panel is free to expand in all directions from the single mounting point. The inner and outer surfaces are attached through relatively long members (rods and channels) to provide stiffness with a minimum heat leak. An external superstructure is used to pick up the loads. A carriage will be designed to mount to the bottom of the structure to maneuver the entire assembly in the CNRS facility.

Figure B-12 shows the structure support for the rear/side panel section of the boiler. Figures B-13 and B-14 show interface information required for design of the test support equipment carriage.

Figure B-15 shows the arrangement of the instrument console.
Fig. B-16

NOTES:
1. CABINET BASE SHALL CONTAIN A PNEUMATIC SUPPLY SYSTEM CONTAINING COMPRESSOR, STORAGE TANK, REGULATORS, ETC. AS REQUIRED FOR BENCH MODEL SYSTEM OPERATION.
Appendix C
Scale-Model Cavity Convective Test Program
Although natural and forced convection losses from the cavity cannot be calculated, scale-model testing can be done to evaluate these losses. The dimensionless groups that should be preserved between the scale model and prototype to properly evaluate both natural and forced convection (including possible interactions between the two heat transfer mechanisms) are as follows:

- **Reynolds number**, \( N_{Re} = \frac{\rho VL}{\mu} \)
- **Grashof number**, \( N_{Gr} = \frac{L^3 \rho^2 g \beta \Delta t}{\mu^2} \)
- **Nusselt number**, \( N_{Nu} = \frac{hL}{K} \)
- **Prandtl number**, \( N_{Pr} = \frac{\mu}{K} \)

**Definition of Terms**
- \( \rho \) Density
- \( V \) Velocity
- \( L \) Characteristic length
- \( \mu \) Viscosity
- \( g \) Acceleration of gravity
- \( \beta \) Volumetric coefficient of expansion
- \( \Delta t \) Temperature difference
- \( h \) Heat transfer coefficient
- \( K \) Thermal conductivity
- \( C_p \) Specific heat

Assuming air is the working fluid in the model and that temperatures will be preserved between the model and prototype, all of the gas properties and \( \Delta t \) will be the same in the model and prototype. Therefore, the only variables remaining are \( \rho \) and \( L \). The density is directly proportional to the pressure. To maintain the same \( N_{Gr} \), the model would have to be tested in a pressure that is calculated as follows.
Subscripts $m$ = model and $p$ = prototype and $L_m^3 P_m^2 = L_p^3 P_p^2$ have the same Grashof number,

$$\left( \frac{L_p}{L_m} \right)^{3/2} \frac{P_m}{P_p} \quad \text{[C-1]}$$

To maintain the same $N_{Re}$,

$$
\rho_m \frac{V_m L_m}{P_m} = \rho_p \frac{V_p L_p}{P_p},
$$

$$
P_m \frac{V_m L_m}{P_m} = P_p \frac{V_p L_p}{P_p},
$$

and

$$
\frac{V_m}{V_p} = \left( \frac{P_p}{P_m} \right) \left( \frac{L_p}{L_m} \right).
$$

But from Equation [D-1],

$$
\frac{P_p}{P_m} = \frac{1}{\left( \frac{L_p}{L_m} \right)^{3/2}}
$$

therefore,

$$
\frac{V_m}{V_p} = \frac{1}{\left( \frac{L_m}{L_p} \right)^{1/2}} \quad \text{[C-2]}
$$

The measured parameters in the model test will be temperatures and total heat flow into the wall heaters. The wall temperatures will be controlled to the actual full-scale values. The sum of radiation and insulation heat loss from the cavity can be measured in the test by orienting the cavity with the aperture down (no convection). The convective heat loss can then be determined by subtracting the sum of the insulation and radiation loss from the total heat input. The convective heat loss from the full-scale prototype can be determined as follows:

$$
q_m = h_m A_m \Delta T_m,
$$

$$
q_p = h_p A_p \Delta T_p,
$$

$$
q_p = h_p A_p \Delta T_p,
$$

$$
q_m = h_m A_m \Delta T_m.
$$

C-2
The area can be written as \( L^2 \), so

\[
q_p = \frac{h_L}{h_m} \frac{L_p}{L_m} \frac{\Delta T_p}{\Delta T_m}.
\]

Because we have maintained the same \( N_{Re}, N_{Gr}, N_{Pr} \) and \( \Delta T \), we have the same Nusselt number,

\[
N_{Nu} = \frac{h_L}{K},
\]

\[
\frac{h_m}{K_m} = \frac{h_p}{K_p}.
\]

Since the same working fluid is used and temperatures are the same, \( K_m = K_p \).

\[
q_p = \frac{h_L}{h_m} \frac{L_p}{L_m} \frac{\Delta T_p}{\Delta T_m}.
\]

Using Equations [C-3] and [C-4] and the fact that \( \Delta T_p = \Delta T_m \),

\[
q_p = \left( \frac{L_p}{L_m} \right).
\]

We recommend that a scale model program based on the analysis given above be conducted. Although such a program is clearly not within the scope of our present program, to obtain a preliminary estimate of the convective losses we performed the test program described in the following paragraphs.

The test article shown in Figure C-1 is a 1/32-scale model made of welded stainless steel. Calrod electrical heaters were assembled to the vertical surfaces on the back and sides. The heaters were wired into six circuits each of which was controlled by a Variac. Twenty-four thermocouples were welded to the interior surfaces. The entire assembly was insulated using layers of \( \frac{1}{2} \)-inch-thick fiberglass and aluminum radiation shields to a thickness of about 2 inches. The completed test article is shown in Figure C-2. It was supported by wires that allowed the angle of the axis to be varied with respect to the horizontal and with respect to a fan that was used to simulate winds.
Figure C-1
Structure of 1/32-Scale Model
Boiler/Superheater Cavity
Figure C-2 1/32-Scale Model Cavity Convection Test Setup
<table>
<thead>
<tr>
<th>Run No.</th>
<th>Orientation</th>
<th>With Respect to Horizon-</th>
<th>Wind Speed, mph</th>
<th>Total Power, W</th>
<th>Convective Loss, W</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>With to Horizontal</td>
<td>to Wind</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>-90</td>
<td>--</td>
<td>0</td>
<td>550</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>-25</td>
<td>--</td>
<td>0</td>
<td>1095</td>
<td>545</td>
</tr>
<tr>
<td>3</td>
<td>-25</td>
<td>0</td>
<td>2.9</td>
<td>1165</td>
<td>615</td>
</tr>
<tr>
<td>4</td>
<td>-25</td>
<td>0</td>
<td>5.7</td>
<td>1250</td>
<td>700</td>
</tr>
<tr>
<td>5</td>
<td>-25</td>
<td>0</td>
<td>9.1</td>
<td>1400</td>
<td>856</td>
</tr>
<tr>
<td>6</td>
<td>-25</td>
<td>0</td>
<td>13.7</td>
<td>1835</td>
<td>1285</td>
</tr>
<tr>
<td>7</td>
<td>-25</td>
<td>90</td>
<td>2.9</td>
<td>1130</td>
<td>580</td>
</tr>
<tr>
<td>8</td>
<td>-25</td>
<td>90</td>
<td>5.7</td>
<td>1263</td>
<td>713</td>
</tr>
<tr>
<td>9</td>
<td>-25</td>
<td>90</td>
<td>9.1</td>
<td>1305</td>
<td>755</td>
</tr>
<tr>
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<td>-25</td>
<td>90</td>
<td>13.7</td>
<td>1450</td>
<td>900</td>
</tr>
<tr>
<td>11</td>
<td>-25</td>
<td>45</td>
<td>2.9</td>
<td>1265</td>
<td>715</td>
</tr>
<tr>
<td>12</td>
<td>-25</td>
<td>45</td>
<td>5.7</td>
<td>1390</td>
<td>840</td>
</tr>
<tr>
<td>13</td>
<td>-25</td>
<td>45</td>
<td>9.1</td>
<td>1550</td>
<td>1000</td>
</tr>
<tr>
<td>14</td>
<td>-25</td>
<td>45</td>
<td>13.7</td>
<td>1710</td>
<td>1160</td>
</tr>
</tbody>
</table>

*0° Horizontal, -90-deg aperture down.

†0° Wind normal to aperture.
A series of tests was conducted in which the attitude of the test article and wind speed were varied. Table C-1 shows the test conditions that were run. Run 1 with the aperture facing downward was used to determine the power loss due to insulation and radiation. It was assumed that the convective loss was negligible in this test. The convective loss shown in Table D-1 is the total power less 550 (loss from Run 1). The results are plotted in Figure C-3 along with the calculated loss for an open vertical surface (i.e., no cavity). The conclusions are as follows:

1) Natural convective loss from the cavity will not exceed 2%;
2) At a wind speed of 10 mph the loss will not exceed 3%;
3) At high wind speeds the convective loss from the cavity is significantly less than for an exposed heat exchanger.

It is assumed that the heat transfer coefficient in the bench and full-scale model will be the same as in our scale model. We have good reason to believe that the scale model heat transfer coefficient is probably higher than for the larger sizes. This argument will be presented in our next report.
Figure C-3 Convective Loss vs Wind Speed
Appendix D
Profiles of Salt River Project Operation and Potential 100 MWe Solar Power Plant
POTENTIAL SOLAR PLANT OUTPUT POWER
1219 MW-HRS. 2551 BBL OIL EQUIV.

Figure D-1
SUNDAY, JULY 1, 1973
PHOENIX TEMPERATURE RANGE -- 111/82 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
1245 MW-HRS. 2714 BBL OIL EQUIV.

SYSTEM LOAD

HORSE MESA HYDRO NO. 4
BALANCE OF HYDRO UNITS

COMBUSTION TURBINES

POTENTIAL SOLAR PLANT

Figure D-2
MONDAY, JULY 2, 1973
PHOENIX TEMPERATURE RANGE -- 115/87 °F
Figure D-3
TUESDAY JULY 3, 1973
PHOENIX TEMPERATURE RANGE -- 111/2° F
Figure D-4

WEDNESDAY, JULY 4, 1973
PHOENIX TEMPERATURE RANGE -- 113/91 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
1219 MW-HRS. 2651 BBAL OIL EQUIV.

- SYSTEM LOAD
- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES
- POTENTIAL SOLAR PLANT

Figure D-5
THURSDAY, JULY 5, 1973
PHOENIX TEMPERATURE RANGE -- 113/87 °F
Figure D-6

Friday, July 6, 1973
Phoenix Temperature Range -- 109/89 °F
Figure D-7
SATURDAY, JULY 7, 1973
PHOENIX TEMPERATURE RANGE -- 107/95 °F
Figure 0-8
SUNDAY, JULY 29, 1973
PHOENIX TEMPERATURE RANGE -- 105/84 °F
Figure 0-9
MONDAY, JULY 30, 1973
PHOENIX TEMPERATURE RANGE -- 99/80 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
677 MW-HRS. 1471 BBL OIL EQUIV.

- - - SYSTEM LOAD

- - - HORSE MESA HYDRO NO. 4

- - - BALANCE OF HYDRO UNITS

- - - COMBUSTION TURBINES

- - - POTENTIAL SOLAR PLANT

TIME (HOURS)
Figure D-10
TUESDAY, JULY 31, 1973
PHOENIX TEMPERATURE RANGE -- 105/80 °F
Figure D-11
WEDNESDAY, AUGUST 1, 1973
PHOENIX TEMPERATURE RANGE -- 106/83 °F
Figure D-12
THURSDAY, AUGUST 2, 1973
PHOENIX TEMPERATURE RANGE -- 107/83 °F
Figure D-13

Friday. August 3. 1973
Phoenix Temperature Range -- 103/83 °F
Figure D-14
SUNDAZ. AUGUST 4 1973
PHOENIX TEMPERATURE RANGE - 103/84 °F

POWER (MW HRS) x 10^1

PHOENIX TEMPERATURE RANGE -- 103/84 °F
Figure D-15
SUNDAY, AUGUST 26, 1973
PHOENIX TEMPERATURE RANGE -- 106/74 °F
Figure D-16
MONDAY, AUGUST 27, 1973
PHOENIX TEMPERATURE RANGE -- 103/76 °F
Figure D-17

TUESDAY, AUGUST 28, 1973

PHOENIX TEMPERATURE RANGE -- 104/72 °F
Figure D-18

WEDNESDAY, AUGUST 29, 1973

PHOENIX TEMPERATURE RANGE -- 104/81 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
295 MWHRS. 642 BBL OIL EQUIV.
Figure D-19
THURSDAY, AUGUST 30, 1973
PHOENIX TEMPERATURE RANGE -- 103/77 °F
Figure D-20
FRIDAY, AUGUST 31, 1973
PHOENIX TEMPERATURE RANGE -- 103/83 °F
Figure D-21
SATURDAY, SEPTEMBER 1, 1973
PHOENIX TEMPERATURE RANGE -- 102/74 °F
Figure D-22
SUNDAY, SEPTEMBER 23, 1973
PHOENIX TEMPERATURE RANGE -- 94/69 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1002 MW-HRS. 2180 BBL OIL EQUIV.
POTENTIAL SOLAR PLANT OUTPUT POWER
1117 MW-HRS, 2429 BBL OIL EQUIV.

SYSTEM LOAD

- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES

Figure D-23
MONDAY, SEPTEMBER 24, 1973
PHOENIX TEMPERATURE RANGE -- 96/61 °F
Figure D-24
TUESDAY, SEPTEMBER 25, 1973
PHOENIX TEMPERATURE RANGE -- 91/64 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1098 MW-HRS. 2388 BBL OIL EQUIV.

- SYSTEM LOAD
- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES

POTENTIAL SOLAR PLANT

0.00 3.00 6.00 9.00 12.00 15.00 18.00 21.00 24.00
TIME (HOURS)
Figure D-25

WEDNESDAY, SEPTEMBER 25, 1973

PHOENIX TEMPERATURE RANGE -- 92/65 °F
Figure D-26
THURSDAY, SEPTEMBER 27, 1973
PHOENIX TEMPERATURE RANGE -- 95/67 °F

---

POTENTIAL SOLAR PLANT OUTPUT POWER
1107 MW-HRS, 2408 BBL OIL EQUIV.

- SYSTEM LOAD
- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- POTENTIAL SOLAR PLANT
Figure D-27
FRIDAY, SEPTEMBER 28, 1973
PHOENIX TEMPERATURE RANGE -- 87/52 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1145 MW-HRS. 2480 BBL OIL EQUIV.

SYSTEM LOAD

- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS

TIME (HOURS)
0.00  3.00  6.00  9.00  12.00  15.00  18.00  21.00  24.00
Figure D-28
SATURDAY, SEPTEMBER 29, 1973
PHOENIX TEMPERATURE RANGE -- 98/61 °F
Figure D-29
SUNDAY, OCTOBER 21, 1973
PHOENIX TEMPERATURE RANGE -- 94/58 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1024 MW-HRS. 2228 BBL OIL EQUIV.
POTENTIAL SOLAR PLANT OUTPUT POWER
1015 MW-HRS. 2207 BBL OIL EQUIV.

SYSTEM LOAD

- HORSE MESA HYDRO NO. 4

- BALANCE OF HYDRO UNITS

Pump Back

- COMBUSTION TURBINES

- POTENTIAL SOLAR PLANT

MARCH 1973
PHOENIX TEMPERATURE RANGE -- 94/57 °F

Figure D-30
MONDAY, OCTOBER 22, 1973
POTENTIAL SOLAR PLANT OUTPUT POWER
1001 MW·HRS. 2177 BBL OIL EQUIV.

Figure D-31
TUESDAY, OCTOBER 23, 1973
PHOENIX TEMPERATURE RANGE -- 89/50 °F

- COMBUSTION TURBINES
- POTENTIAL SOLAR PLANT
- BALANCE OF HYDRO UNITS
- HORSE MESA HYDRO NO. 4
- SYSTEM LOAD
- PUMP BACK
Figure D-32
WEDNESDAY, OCTOBER 24, 1973
PHOENIX TEMPERATURE RANGE -- 87/55 °F

**SYSTEM LOAD**

- **COMBUSTION TURBINES**
- **POTENTIAL SOLAR PLANT**
- **HORSE MEGA HYDRO NO. 4**
- **BALANCE OF HYDRO UNITS**

**POTENTIAL SOLAR PLANT OUTPUT POWER**
1005 MW-HRS. 2185 BBL OIL EQUIV.
Figure D-33
THURSDAY, OCTOBER 25, 1973
PHOENIX TEMPERATURE RANGE -- 33/53 °F
Figure D-34
FRIDAY, OCTOBER 26, 1973
PHOENIX TEMPERATURE RANGE -- 87/54 °F
Figure D-35
SATURDAY, OCTOBER 27, 1973
PHOENIX TEMPERATURE RANGE -- 92/52 °F
Figure D-36
SUNDAY, NOVEMBER 18, 1973
PHOENIX TEMPERATURE RANGE -- 77/47 °F
Figure D-37
MONDAY, NOVEMBER 19, 1973
PHOENIX TEMPERATURE RANGE -- 59/49 °F
Figure D-38
TUESDAY, NOVEMBER 20, 1973
PHOENIX TEMPERATURE RANGE -- 62/39 °F
Figure D-39

Wednesday, November 21, 1973
Phoenix temperature range -- 66/41 °F
Figure D-40
THURSDAY, NOVEMBER 22, 1973
PHOENIX TEMPERATURE RANGE -- 56/48 °F
Figure D-41

FRIDAY. NOVEMBER 23, 1973

PHOENIX TEMPERATURE RANGE -- 57/46°F

POTENTIAL SOLAR PLANT OUTPUT POWER
0 MW-HRS. 0 BBL OIL EQUIV.

- SYSTEM LOAD
- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES

Pump Back
Figure D-42

SATURDAY, NOVEMBER 24, 1973
PHOENIX TEMPERATURE RANGE -- 57/45 °F
Figure D-43
SUNDAY, DECEMBER 16, 1973
PHOENIX TEMPERATURE RANGE -- 78/46 °F
Figure D-44

MONDAY, DECEMBER 17, 1973
PHOENIX TEMPERATURE RANGE -- 75/44 °F
Figure D-45
TUESDAY, DECEMBER 19, 1973
PHOENIX TEMPERATURE RANGE -- 73/41 °F
Figure D-46

WEDNESDAY, DECEMBER 19, 1973
PHOENIX TEMPERATURE RANGE -- 70/38 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
661 MW-HRS, 1873 BBL OIL EQUIV.

SYSTEM LOAD

COMBUSTION TURBINES

POTENTIAL SOLAR PLANT

Figure D-47
THURSDAY, DECEMBER 20, 1973
PHOENIX TEMPERATURE RANGE -- 67/43 °F
Figure D-48

FRIDAY, DECEMBER 21, 1973

PHOENIX TEMPERATURE RANGE -- 88/36 °F
Figure D-49
SATURDAY, DECEMBER 22, 1973
PHOENIX TEMPERATURE RANGE -- 53/39 °F
Figure D-50
SUNDAY, JANUARY 13, 1974
PHOENIX TEMPERATURE RANGE -- 74/44 °F
Figure D-51
MONDAY, JANUARY 14, 1974
PHOENIX TEMPERATURE RANGE -- 73/42 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
340 MW-HRS, 2043 BBL OIL EQUIV.

180.000,000
140.000,000
120.000,000
100.000,000
80.000,000
40.000,000
20.000,000
0.000,000

SYSTEM LOAD

HORSE MESA HYDRO NO. 4

BALANCE OF HYDRO UNITS

COMBUSTION TURBINES

POTENTIAL SOLAR PLANT

TIME (HOURS)
Figure D-52
TUESDAY, JANUARY 15, 1974
PHOENIX TEMPERATURE RANGE -- 78/43 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
389 MW-HRS. 2151 BBL OIL EQUIV.
Figure D-53

WEDNESDAY, JANUARY 16, 1974

PHOENIX TEMPERATURE RANGE = 70/45 °F
Figure D-54
THURSDAY, JANUARY 17, 1974
PHOENIX TEMPERATURE RANGE -- 70/45 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
304 MW-HRS. 1956 BBL OIL EQUIV.

Figure D-55
FRIDAY, JANUARY 19, 1974
PHOENIX TEMPERATURE RANGE -- 72/43 °F
Figure D-56
SATURDAY, JANUARY 19, 1974
PHOENIX TEMPERATURE RANGE -- 72/44 °F
Figure 0-57
SUNDAY, FEBRUARY 10, 1974
PHOENIX TEMPERATURE RANGE -- 76/35 °F
Figure D-58
MONDAY, FEBRUARY 11, 1974
PHOENIX TEMPERATURE RANGE -- 75/38 °F
Figure D-59
TUESDAY, FEBRUARY 12, 1974
PHOENIX TEMPERATURE RANGE -- 71/47 °F
Figure D-61

THURSDAY, FEBRUARY 14, 1974
PHOENIX TEMPERATURE RANGE -- 70/38 °F
Figure D-62

FRIDAY, FEBRUARY 15, 1974

PHOENIX TEMPERATURE RANGE -- 74/32 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
643 MW-HRS, 1833 BBL OIL EQUIV.

SYSTEM LOAD

- M HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES
- POTENTIAL SOLAR PLANT

Pump Back
Figure D-63
SUNDAY, FEBRUARY 16, 1974
PHOENIX TEMPERATURE RANGE - 76/40 °F
TENTIRL SOLAR PLANT OUTPUT POWER
7033 MW-HRS. 2312 BBL OIL EQUIV.

SYSTEM LOAD

M | HORSE MESA HYDRO NO. 4
M | BALANCE OF HYDRO UNITS

Δ | POTENTIAL SOLAR PLANT

Figure D-64

SUNDAY, MARCH 10, 1974
PHOENIX TEMPERATURE RANGE -- 63/38 °F
Figure D-65

MARCH 11, 1974

PHOENIX TEMPERATURE RANGE -- 74/43 °F
Figure D-66

TUESDAY, MARCH 12, 1974

PHOENIX TEMPERATURE RANGE -- 81/48 °F
Figure D-67

WEDNESDAY, MARCH 13, 1974

PHOENIX TEMPERATURE RANGE -- 84/47 °F
THURSDAY, MARCH 14, 1974
PHOENIX TEMPERATURE RANGE -- 85/49 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
159 MW-HRS. 2522 BBL OIL EQUIV.

SYSTEM LOAD

- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS

- POTENTIAL SOLAR PLANT

Figure D-69
FRIDAY, MARCH 15, 1974
PHOENIX TEMPERATURE RANGE -- 87/50 °F
Figure D-70

SATURDAY, MARCH 16, 1974
PHOENIX TEMPERATURE RANGE - 90/52°F

PHASEIPLANT OUTPUT POWER
114 MW-HR, 2483 BBL OIL EQUIV.

SYSTEM LOAD

BALANCE OF HYDRO UNITS

POTENTIAL SOLAR PLANT

0.00 10.00
0.00 20.00
40.00 60.00 80.00 100.00 120.00
140.00 160.00 180.00 200.00

TIME (HOURS)
0.00 3.00 6.00 9.00 12.00 15.00 18.00 21.00 24.00

PHASE TEMPERATURE RANGE - 90/52°F
OTENTIRL SOLAR PLANT OUTPUT POWER
1157 MW-HRS. 2516 BBL OIL EQUIV.

SYSTEM LOAD

E - E HORSE MESA HYDRO NO. 4

- - BALANCE OF HYDRO UNITS

Pump Back

- - + POTENTIAL SOLAR PLANT

Figure D-71
SUNDAY, APRIL 7, 1974
PHOENIX TEMPERATURE RANGE -- 90/50 °F

PHOENIX TEMPERATURE RANGE -- 90/50 °F
Figure D-72
MONDAY, APRIL 3, 1974
PHOENIX TEMPERATURE RANGE -- 93/54 °F

- Potential Solar Plant Output Power
  1923 MWhr, 2007 BBL Oil Equiv.

- System Load

- Horse Mesa Hydro No. 4

- Balance of Hydro Units

- Pump Back

- Combustion Turbines

- Potential Solar Plant

- Phoenix Temperature Range -- 93/54 °F
Figure D-73
TUESDAY, APRIL 9, 1974
PHOENIX TEMPERATURE RANGE -- 87/59 °F
Figure D-74

WEDNESDAY, APRIL 10, 1974

PHOENIX TEMPERATURE RANGE -- 74/53 °F
Figure D-75
THURSDAY, APRIL 11, 1974
PHOENIX TEMPERATURE RANGE -- 81/48 °F
Figure D-76
FRIDAY, APRIL 12, 1974
PHOENIX TEMPERATURE RANGE -- 87/51 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1157 MW-HRS, 2516 BBL OIL EQUIV.
POTENTIAL SOLAR PLANT OUTPUT POWER
1046 MW-MAS. 2275 BBL OIL EQUIV.

SYSTEM LOAD

- - - HORS MWES HYDRO NO. 4
- - - BALANCE OF HYDRO UNITS

Pump Back

△ - + POTENTIAL SOLAR PLANT

Figure D-77
SATURDAY, APRIL 13, 1974
PHOENIX TEMPERATURE RANGE -- 82/83 °F
Figure D-78
SUNDAY, MAY 5, 1974
PHOENIX TEMPERATURE RANGE -- 92/62 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1143 MW-HRS. 2400 BBL OIL EQUIV.
Figure D-79

MONTAY, MAY 6, 1974

PHOENIX TEMPERATURE RANGE = 94/60 °F
PHOENIX TEMPERATURE RANGE -- 94/67 °F

Figure D-80
TUESDAY, MAY 7, 1974

POTENTIAL SOLAR PLANT OUTPUT POWER
978 MW-HRS. 2040 BBL OIL EQUIV.
Figure D-81
WEDNESDAY, MAY 8, 1974
PHOENIX TEMPERATURE RANGE -- 94/70 °F
Figure D-83
FRIDAY, MAY 10, 1974
PHOENIX TEMPERATURE RANGE -- 36/66 °F

Potential Solar Plant Output Power
1143 MW-HRS. 2486 BBL OIL EQUIV.
Figure D-84
SATURDAY, MAY 13, 1974
PHOENIX TEMPERATURE RANGE -- 99/66 °F
Figure D-85

SUNDAY, JUNE 2, 1974

PHOENIX TEMPERATURE RANGE -- 102/66 °F
Figure D-86

POTENTIAL SOLAR PLANT OUTPUT POWER
1271 MW-HRS. 2763 BBL OIL EQUIV.

- SYSTEM LOAD
- HORSE MESA HYDRO NO. 4
- BALANCE OF HYDRO UNITS
- COMBUSTION TURBINES
- POTENTIAL SOLAR PLANT

MONDAY, JUNE 3, 1974
PHOENIX TEMPERATURE RANGE -- °F

0.00 3.00 6.00 9.00 12.00 15.00 18.00 21.00 24.00
TIME (HOURS)
POTENTIAL SOLAR PLANT OUTPUT POWER
1306 MW-HRS, 2840 BBL OIL EQUIV.

SYSTEM LOAD

- O HORSE MESA HYDRO NO. 4
- O BALANCE OF HYDRO UNITS
- O COMBUSTION TURBINES
- + POTENTIAL SOLAR PLANT

Figure D-87
TUESDAY, JUNE 4, 1974
PHOENIX TEMPERATURE RANGE -- 101/68 °F

TIME (HOURS)
Figure D-88

WEDNESDAY, JUNE 5, 1974

PHOENIX TEMPERATURE RANGE -- 101/69 °F
POTENTIAL SOLAR PLANT OUTPUT POWER
1170 MW-HRS. 2544 BBL OIL EQUIV.

SYSTEM LOAD

- O HORSE MESA HYDRO NO. 4
- O BALANCE OF HYDRO UNITS
- Pump Back
- O COMBUSTION TURBINES

Ủ - + POTENTIAL SOLAR PLANT

Figure D-89
THURSDAY, JUNE 6, 1974
PHOENIX TEMPERATURE RANGE -- 103/70 °F
Figure D-90
FRIDAY, JUNE 7, 1974  100/70
PHOENIX TEMPERATURE RANGE --  °F
Figure D-91
SATURDAY, JUNE 3, 1974
PHOENIX TEMPERATURE RANGE -- 83/72 °F

POTENTIAL SOLAR PLANT OUTPUT POWER
1181 MW-HRS. 2568 BBL OIL EQUIV.

- ○ SYSTEM LOAD
- ○ HORSE MESA HYDRO NO. 4
- ○ BALANCE OF HYDRO UNITS
- ○ COMBUSTION TURBINES
- ± POTENTIAL SOLAR PLANT

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