Closed Cycle High-Temperature Central Receiver Concept for Solar Electric Power

EPRI

EPRI ER-403-SY
Project 377-1
Summary Report
August 1976

Prepared by
Boeing Engineering and Construction
Seattle, Washington

ELECTRIC POWER RESEARCH INSTITUTE

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(Research Project 377-1)

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Prepared by

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Abstract

The Electric Power Research Institute (EPRI) awarded Boeing a contract to examine the technical feasibility of a high temperature, gas cooled central receiver for producing electric power from solar energy using a closed Brayton helium cycle. Feasibility was examined in terms of system life, efficiency, cost, and technology requirements. These considerations have been implemented into the conceptual design of a receiver for utilization in a 100 megawatt output solar plant. The rationale is provided which supports the configuration, equipment arrangement, and material choices. Thermal cycling tests simulating a 30-year lifetime of the receiver's heat exchangers at operational temperatures to 816°C (1500°F) were performed to select materials. Preliminary design considerations were made for a 1 megawatt bench model receiver to verify the full scale receiver. Preliminary planning was developed for a 10 megawatt electrical pilot plant to follow the receiver verification.

The scope of the study also included system/subsystem definition for employing the central receiver design in a solar plant and predicting plant performance. Conceptual designs of several thermal energy storage devices were defined, integrated into plant performance and operational models, and evaluated with energy cost as the criteria.

The information developed during the study is highlighted in this final summary report. A final technical report has been prepared which provides additional detail. The final technical report, and interim reports are available from EPRI.

Summary

The 18-month study of the closed cycle, high temperature central receiver and its integration into a 100 MW_e solar plant confirmed the predicted potentialities of the concept. Technical feasibility of a cavity-type receiver employing closed cycle helium has been reinforced in areas of design, materials, performance, and integration into commercial plant operations. A singular receiver design has been accomplished which can be used in a stand-alone (solar only) plant with thermal storage capability or in a hybrid (solar plus fossil-fuel backup) plant. Receiver materials have been selected to meet the high temperature capability demanded of the concept. Receiver heat exchangers operate at 816°C (1500°F) and 3.45 MN/m^2 (500 psi). Two superalloys, Inconel 617 and Haynes 188, have been evaluated and tested by thermal cycling to 816°C (1500°F) at operational pressure through 10,500 cycles (equivalent to 30 years of diurnal cycling). Performance and operational studies show the receiver operates simply and effectively over a wide range of environmental and operational conditions. All the required technology for receiver design and implementation currently exists. Preliminary planning and design has been accomplished for verifying the receiver by bench model tests, and incorporating the receiver into a pilot plant. Receiver costs, while slightly higher than postulated water/steam receivers are such that overall plant costs are equivalent to steam/Rankine cycle solar plants due to the effectiveness of closed cycle helium.

The use of a closed Brayton cycle with helium as a working fluid shows a potential for high conversion efficiency which results in reduced size and cost of all elements involved in collecting and processing the heat to be converted to electrical energy. Thermal cycle parameters (pressures, temperatures, and recuperator effectiveness) have been selected to attain a 0.44 cycle efficiency, and turbomachinery has been sized and costed. A closed air Brayton cycle has also been examined and may be utilized in a manner similar to helium with only slight impact on performance and cost.

Three thermal storage concepts (phase change, sensible heat, and thermochemical) with a six hour storage limit have been conceptually designed, costed, and integrated into plant operations. The phase change storage device has a cost advantage, but all devices show quite similar performance when compared on a seasonal or annual basis.

An overview of the study results is contained in this report.

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1. Introduction

Conversion of solar energy to electrical energy has assumed increasing significance resulting from our expanded energy requirements and the potential resource and cost constraints of conventional fossil-fuel sources. One method of converting solar energy to electric power is thermal energy conversion in conjunction with a turbine-generator set. Application of this process is the subject of this report.

The general availability, inexhaustible supply, and inherent cleanliness of solar energy as an energy source has prompted major sponsoring organizations such as the Electric Power Research Institute (EPRI) and the Energy Research and Development Administration (ERDA) to explore this potential. The National Science Foundation (NSF) previously has sponsored system and subsystem studies of conceptual designs for solar thermal power plants using conventional steam-turbine generation equipment. The Aerospace Corporation completed a mission analysis of solar thermal power plants that included siting considerations, central receiver and distributed collector systems, and their integration into existing electric utility systems. That effort has provided excellent background material for selection of the central receiver concept explored in this study. The ERDA has recently initiated preliminary conceptual design studies for a 10 MW_e central receiver solar thermal pilot plant using a water/steam Rankine cycle.

In December 1974, EPRI awarded Boeing a contract to examine the technical feasibility of a high temperature, gas-cooled, central receiver in conjunction with a closed Brayton helium cycle for collecting the receiver thermal energy and converting that energy to electrical power. These choices were based upon the following rationale: (1) previous studies have identified the central receiver system as the most economically attractive concept; (2) the Brayton gas cycle operation precludes the two-phase flow problems of water/steam Rankine cycles; (3) operation at high temperatures promises the highest power conversion efficiencies and lowest cost; and (4) the minimum cooling water requirements facilitate plant siting.

An overview of the information developed during the contract period is presented in this final summmary report. For readers desiring additional detail, a final technical report is available from EPRI.

Artist's Concept

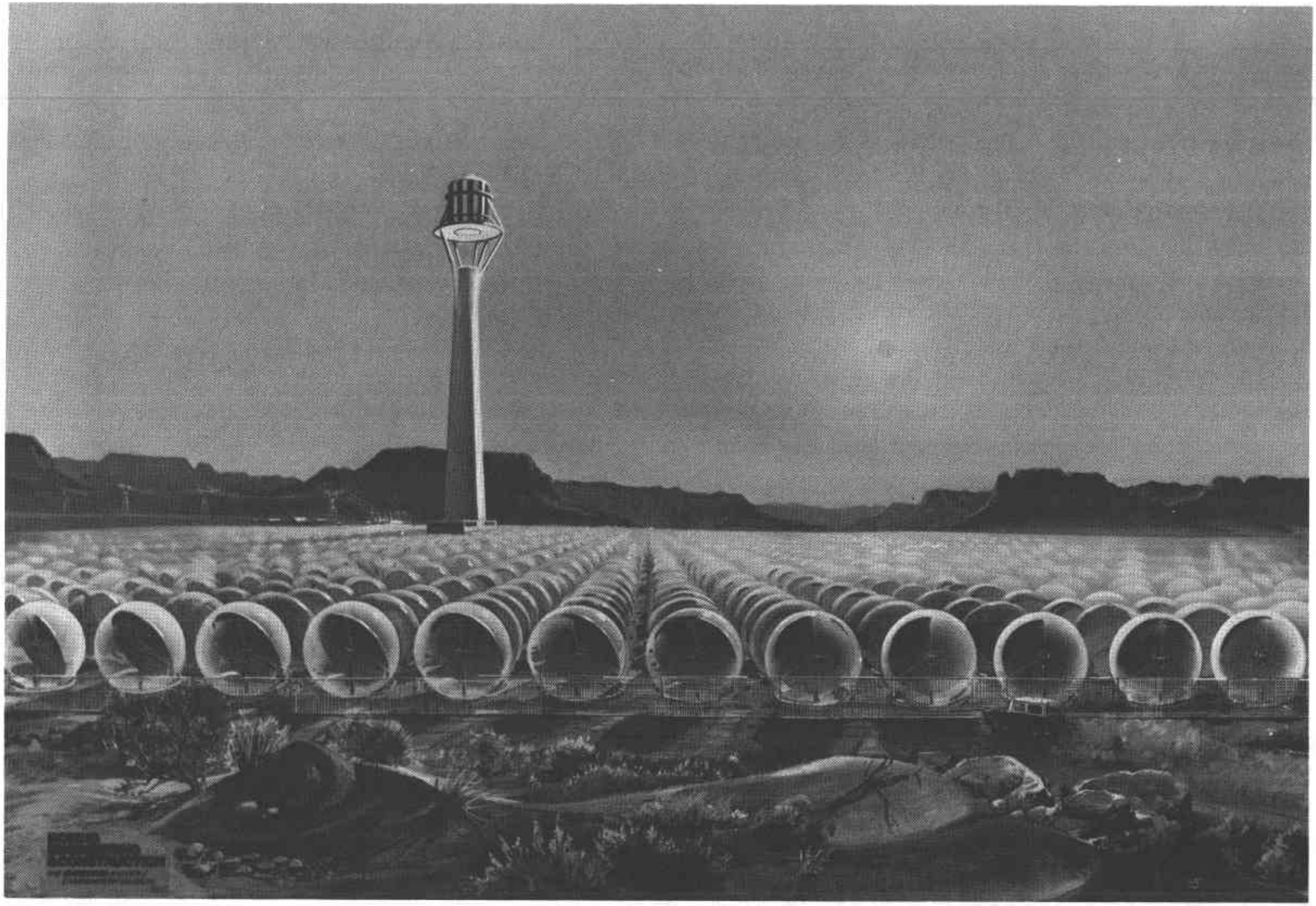
The facing page illustrates a central receiver system and shows a receiver mounted on top of a tower located centrally in a heliostat field. Reflected solar energy from the heliostats is directed through an aperture in the bottom of the receiver. The energy is reflected from the receiver walls onto high temperature heat exchangers through which the working fluid, helium, is circulated. The heated helium is transported to a turbine located either at the top or the base of the tower. The high temperature and thermal properties of helium combine to provide the potential for highly efficient conversion to electrical power by the turbine-driven generator.

Objectives

The study objectives include receiver conceptual design; system integration and costs; and a materials test program to verify the receiver design. Specifically, the study was directed to:

- Determine technical feasibility of a high temperature central receiver utilizing a closed cycle helium system considering lifetime, efficiency, cost, and technology requirements.
- Provide a general system definition and system performance parameters for a central receiver concept to produce 100 MW_e output.
- Provide a concept definition of a 1 MW_{th} test model receiver to simulate the 100 MW_e concept, including a development plan and cost estimate.
- Perform supporting thermal cycle tests of a representative receiver heat exchanger element to verify operational lifetime at high temperature.

These initial objectives were met during the study period, and the study was extended to include energy storage concepts and plant operations. Re-direction received from EPRI after the interim study phase modified the study to be in concert with other solar thermal conversion programs. This necessitated inclusion of both a stand-alone (solar only) plant with six hours thermal storage, and a hybrid (solar plus fossil-fuel backup) plant with one-half hour of thermal storage. The results obtained are summarized on the following pages. The project was completed in June 1976.



2. Design Requirements

Design Guidelines

Initial study requirements called for design of a high temperature, gas-cooled, central receiver system sized to generate 100 MW_e, using helium as the working fluid in a closed cycle mode. As the study matured, EPRI defined a plant model containing collector field characteristics, tower size, storage capability, megawatt output, and associated costs. These guidelines were incorporated and modified to reflect the results derived from the continuing studies of the receiver, the helium heat transport subsystem, storage subsystem, and turbomachinery. Performance and costs of these elements were assessed and entered into the plant model to permit EPRI to make a consistent system comparison with other receiver concepts.

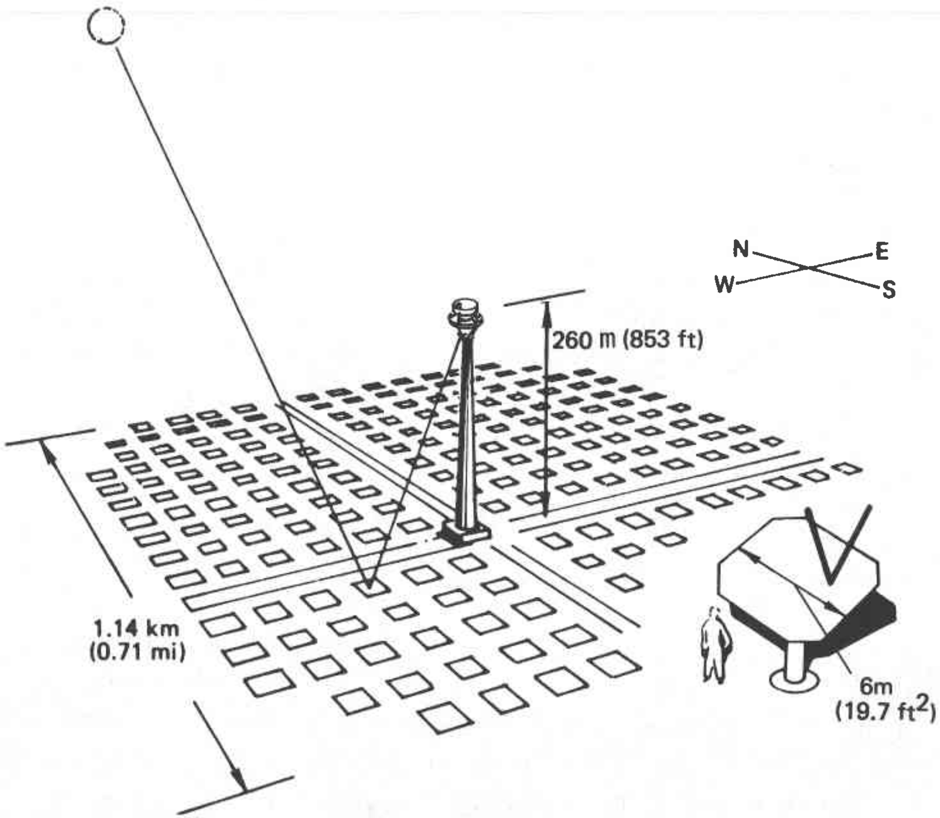
The central receiver modular concept for the EPRI "strawman" is shown on Figure 1. The two 100 MW_e intermediate plants defined for the study were the stand-alone plant consisting of two 50 MW_e plant modules with six hours of thermal storage, and the 100 MW_e hybrid plant consisting of one module with one-half hour thermal storage and fossil fuel back-up. Tower height for a plant module is 260 meters. Collector characteristics are depicted in the drawing. The collector efficiencies listed are annual averages of a heliostat configuration design for a winter-perturbed field. Performance efficiencies given for the elements of a typical Rankine cycle plant were replaced by the appropriate values for the closed cycle helium plant.

The "strawman" field described on Figure 1 completely determines the amount of insolation available to the receiver; however, the definition was not to be considered restrictive if the study results showed changes in field size would be beneficial to the receiver, balance of plant performance, and cost.

Environmental Requirements

The plant location specified was Inyokern California. Insolation profiles and seismic conditions prevailing in that area were used in the design. Winds to be accommodated were 18 meters per second (40 miles per hour) in operation; and for design survivability, 36 meters per second (80 miles per hour) steady with gusts up to 54 meters per second (120 miles per hour). A major design requirement was to use high temperature materials consistent with the state-of-the-art to achieve a 30-year equipment lifetime.

Figure 1 Design Guidelines



PLANT MODULE

100 MWe INTERMEDIATE PLANTS

STAND-ALONE PLANT	2 MODULES (6 HOURS STORAGE)
HYBRID PLANT	1 MODULE (1/2 HOUR STORAGE)

PLANT MODULE CHARACTERISTICS

TOWER HEIGHT	260m (853 ft)
COLLECTOR AREA	0.5 km^2 (0.19 mi^2)
AREA UTILIZATION	38.6%
TOTAL LAND AREA	1.3 km^2 (0.5 mi^2)
NO. OF COLLECTORS	15,400
SIZE OF COLLECTORS	32.4 m^2 (349 ft^2)

COLLECTOR EFFICIENCIES:

TRACKING, AIMING, SHADING, BLOCKING	0.703
REFLECTIVITY	0.880

3. Results

Study results show a high-temperature central receiver employing closed cycle helium to be a promising choice for solar thermal conversion plants. The concept is technically feasible and shows promise of being cost-effective because of the high thermal efficiency obtainable with a closed cycle helium system. The concept and significant results for each of the major feasibility criteria are summarized in this section.

Receiver Concept

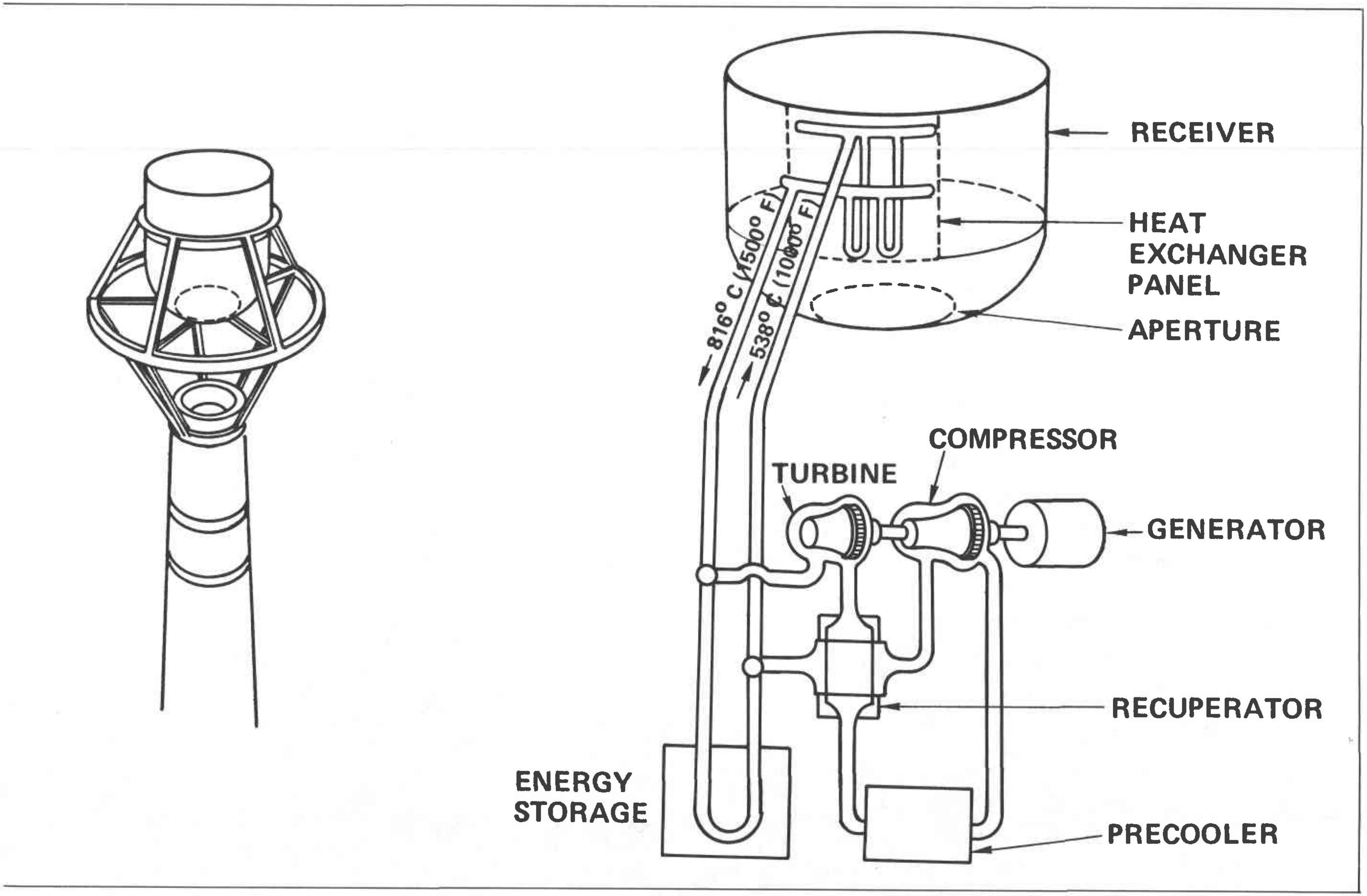
The most promising configuration is shown in Figure 2. The picture at the left illustrates the selected central receiver supported above the tower. The receiver configuration has a hemispherical lower section and a cylindrical upper section. An aperture at the receiver bottom admits the reflected energy from the collector field into the receiver interior. The schematic on the right identifies the major components of the solar plant. Heat exchanger panels are mounted on the interior of the upper cylindrical section to transfer heat from the receiver to the circulating helium.

Helium inlet and outlet temperatures are 538°C (1,000°F) and 816°C (1,500°F), respectively. The upper limit of 816°C (1,500°F) was chosen to increase cycle efficiency, yet remain within the state-of-the-art of high temperature metals. The associated energy conversion and helium processing equipment shown below the receiver would be located at the base of the tower, along with the thermal energy storage equipment (or at the top of the tower for the hybrid concept).

Receiver Lifetime

Materials are available to withstand the high temperatures encountered under repeated thermal cycling of the receiver and heat exchanger tubes during lifetime operation. Thermal cycling tests simulating 30-year lifetime at expected temperatures and pressure have been completed on Haynes 188 and Inconel 617 alloys. No adverse effects were detected for the design temperature limitation of 816°C (1,500°F) and the internal pressure of 3.45 MN/m^2 (500 psi) on either material. Final selection will be based on cost and availability. Stress-rupture tests, at temperatures considerably above the planned operating maximum, confirmed material capability to handle accidental temperature extremes. Test ruptures occurred at 1037°C (1,900°F) or higher under pressure and were non-catastrophic.

Figure 2 Receiver Concept/Schematic



Receiver Configuration

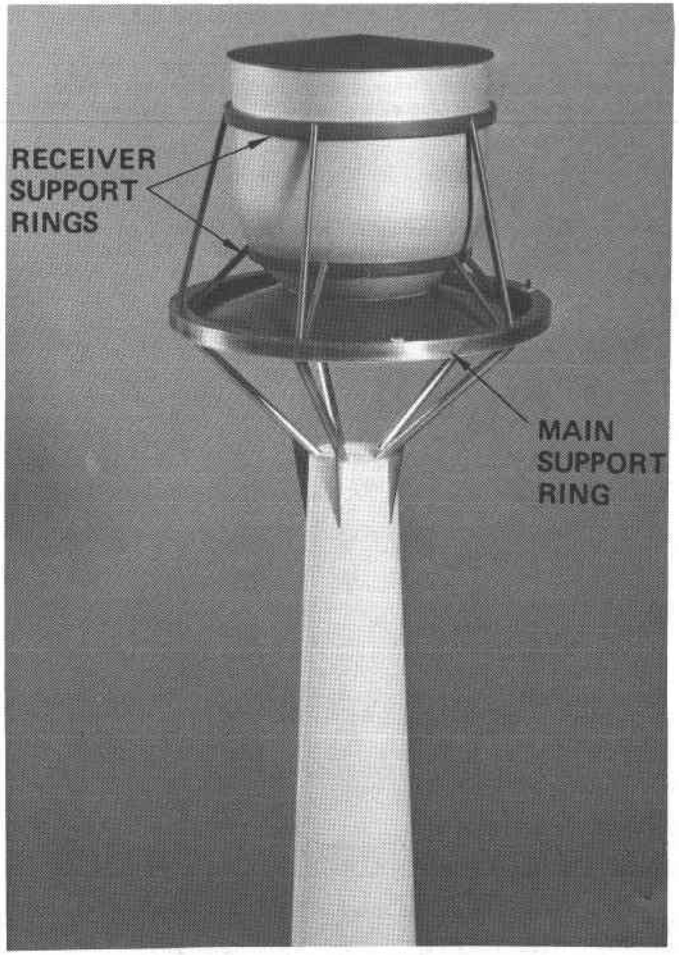
Two views of the design are contained in the photos of a scale model shown on Figure 3. The receiver, its main supports, and the upper tower are shown in the picture at the left. The receiver is mounted approximately 30 meters (98 feet) above the tower top to allow reflected solar energy to enter the aperture at the bottom of the receiver. The aperture is 19 meters (62 feet) in diameter and is located approximately 260 meters (850 feet) above ground level.

The receiver is supported by five support struts which extend out from the tower top to the main support ring. These supports are located away from the aperture to minimize heating by direct radiation and to reduce blockage of incoming energy. Vertical members extend from the main support ring to the receiver support rings. The size of these vertical support members is such that a helium riser or downcomer will be contained within individual supports. There are two risers and two downcomers.

The receiver shape is a composite of a hemispherical lower section, to minimize reflection losses, and a cylindrical upper section, which facilitates mounting of the heat exchanger panel modules. The photo on the right in Figure 3 shows the mounting arrangement of heat exchanger panels in the upper half of the receiver. There are 3 rows of these panels with 70 panels in each row. The lower hemispherical section has insulation panels which reflect and reradiate the energy to the heat exchanger panels and also reduce conductive heat loss.

Receiver interior dimensions are approximately 39 meters (128 feet) in diameter and 39 meters high. The receiver, supports, risers, and downcomers weigh an estimated 1.5 million kilograms (3.3 million pounds).

Figure 3 Preferred Receiver



RECEIVER AND UPPER TOWER

RECEIVER WITH ROOF REMOVED EXPOSING HEAT EXCHANGER PANELS

Receiver Performance

The baseline receiver collects the heat required for both electrical power production and storage for deferred power production in each 50 MW_e-rated stand-alone plant module. For the 100 MW_e-rated hybrid plant, the heat is used for direct electrical power generation requiring only one such module.

Receiver inputs for four representative days out of the year are shown on the graph on Figure 4. These daily input curves are derived from applying the appropriate field efficiency factors to the Inyokern, California, insolation profiles averaged for 30 days on both sides of the dates shown. The table shows the receiver thermal heat balance at noon for summer and winter days. The largest loss at both time periods is due to re-radiation of the energy back out of the aperture, an unavoidable circumstance for high temperature cavities. Reflection losses through the aperture have been minimized by the receiver shape. Convection and conduction losses are small. The net receiver efficiency in transferring heat to the helium is approximately 82%. The turbine requires 230 MW_{th} to produce 100 MW_e.

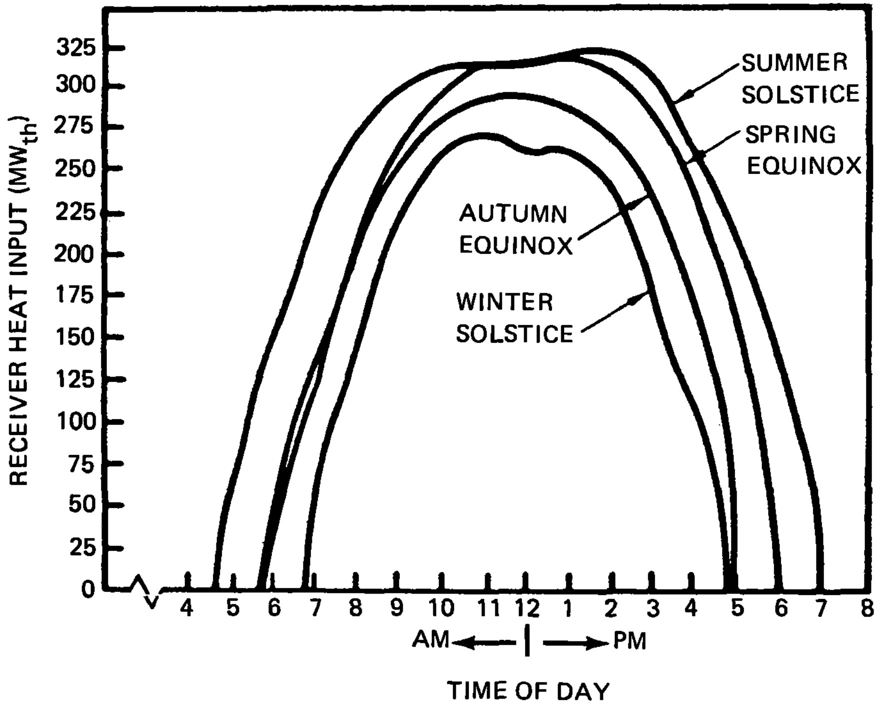
Receiver Cost

The baseline receiver cost is estimated at slightly over $7.8 million (1975 dollars) for the 100 MW_e hybrid plant configuration. For the 100 MW_e stand-alone plant with 6 hours thermal storage, which requires two receivers, the total capital cost is $15.7 million or $157/kilowatt of rated output. These costs have been based on use of Inconel 617 for heat exchanger panel tubing and the adjacent helium distribution lines.

Receiver Verification

The conceptual design of a 1 MW_{th} bench model receiver which simulates the 100 MW_e receiver concept has been completed. It has been primarily designed for the ERDA Solar Test Facility at Albuquerque, New Mexico, but is adaptable to the Centre de la Recherche Scientifique (CNRS) Solar Energy Laboratory at Odeillo, France. The development schedule recommended requires that the 1 MW_{th} bench model receiver be available for testing in early 1978.

Figure 4 Receiver Performance



RECEIVER HEAT BALANCE (MW_{th})

	SUMMER		WINTER	
SOLAR INPUT TO RECEIVER		315.0		259.0
RECEIVER LOSSES:				
REFLECTION OUT APERTURE	11.9		9.7	
RERADIATION OUT APERTURE	33.1		29.5	
CONVECTION TO AIR	2.5		2.4	
CONDUCTION THROUGH WALLS	6.0		5.7	
TOTAL LOSSES		53.5		47.3
HEAT REMOVED BY HELIUM		261.5		211.7

Thermal Cycle

The study of closed gas cycles for a solar plant was predicated on the advantages offered by the higher power conversion efficiencies of those systems and the resulting plant cost economies. This has been exemplified in the closed helium cycle selected for the receiver and plant concept summarized herein. Similar considerations apply to closed air cycles as well.

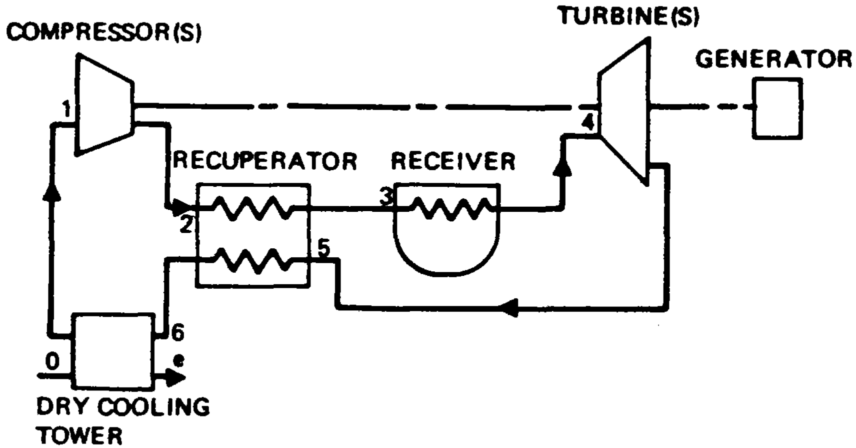
The cycle selected as a design baseline provides the maximum efficiency available with current technology (44%). This system was chosen because analysis indicated it resulted in minimum total plant costs. The schematic and design conditions are displayed on Figure 5. The turbine inlet temperature of 816°C (1,500°F) corresponds to the helium temperature from the receiver and the receiver tubing material limit for a 30-year lifetime. The compressor inlet temperature was selected as 49°C (120°F) to keep the size of the precooler (dry cooling was specified) to a minimum. The compressor pressure ratio of 1.9 has been selected on the basis of cycle efficiency sensitivity studies. The helium pressure level of 3.45 MN/m^2 (500 psi) improves heat exchanger performance, reduces the size (and therefore, cost) of the system and corresponds to a near-optimum pressure level for a 100 MW_e-rated turbine running at synchronous speed (3600 RPM).

Recuperator effectiveness was found to be the dominant factor in both cycle efficiency and cost. Recuperator costs increase disproportionately with increase in recuperator effectivity. However, the additional costs to achieve a 0.94 effectivity are offset by the reduction in heliostat field costs resulting from the high thermal cycle efficiency.

Performance comparisons were made with closed air, open air, and steam cycles. The efficiency advantage of the closed helium cycle is illustrated by the graph on Figure 5. As can be seen, the closed air cycle also performs well. For plant operations, closed cycles also have the potential to provide higher flexibility in control and output than other cycles. The ability to control the gas inventory is an important advantage due to the diversity of environmental and operational conditions possible in a solar plant.

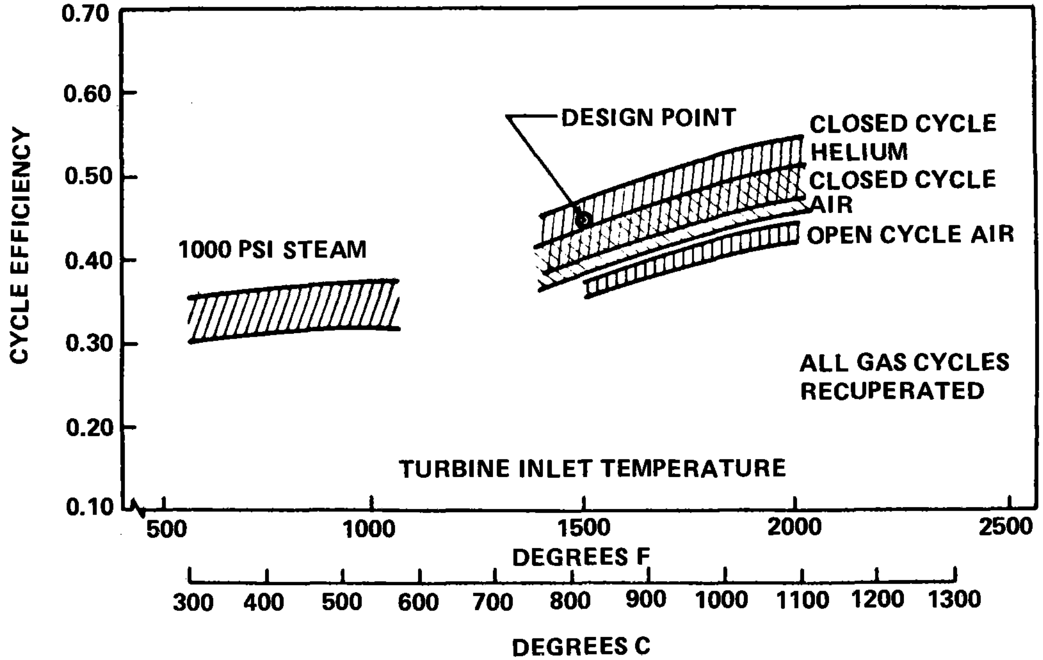
Another area of concern, that of closed cycle turbomachinery availability, has been addressed throughout the study with results indicating that the technology and the machinery can be made available for a reasonably-paced commercial plant development schedule.

Figure 5 Selected Thermal Cycle



BASELINE:

T_4 – TURBINE INLET TEMP	816°C (1500°F)
T_1 – COMPRESSOR INLET TEMP	49°C (120°F)
T_0 – AMBIENT TEMP	27°C (80°F)
P_2/P_1 – PRESSURE RATIO	1.9
P_2 – MAXIMUM PRESSURE	3.45 MN/m^2 (500 psi)
η_e – RECUPERATOR EFFECTIVENESS	0.94
η_{TC} – CYCLE EFFICIENCY	0.44



Plant Performance

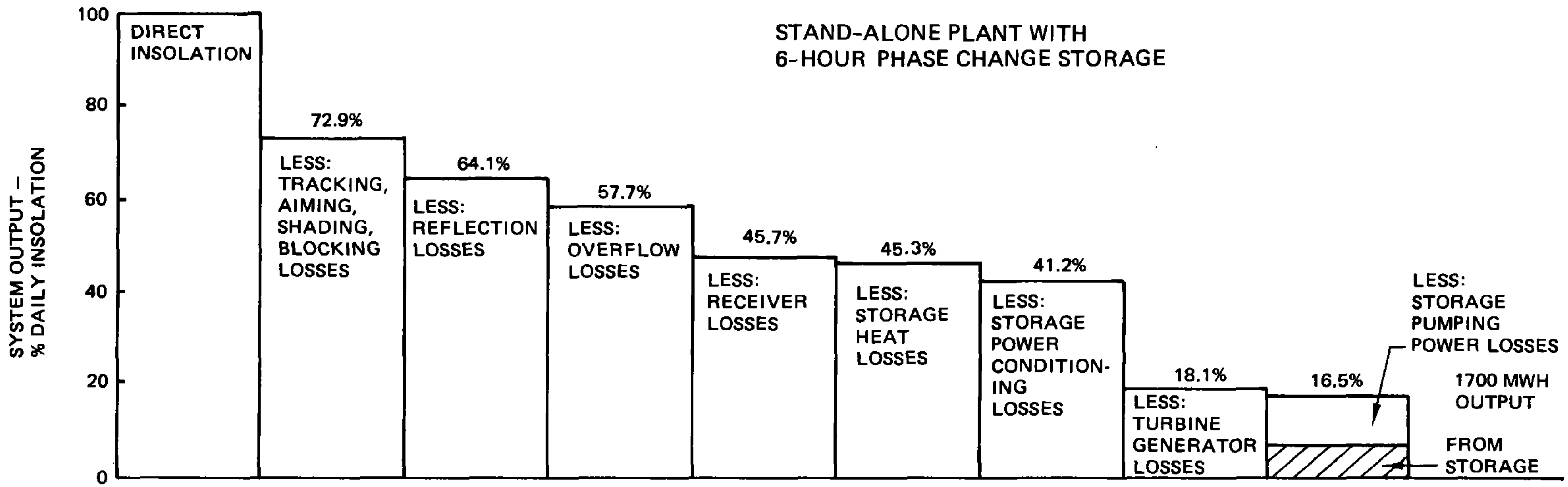
The baseline solar plant subsystems and the selected thermal cycle were used to determine plant performance and operational characteristics. Performance results are typified by the two bar charts shown on Figure 6. Both charts show the efficiency losses in going from the direct daily insolation energy through the plant subsystems to the generator MWH_e output for a summer daily cycle.

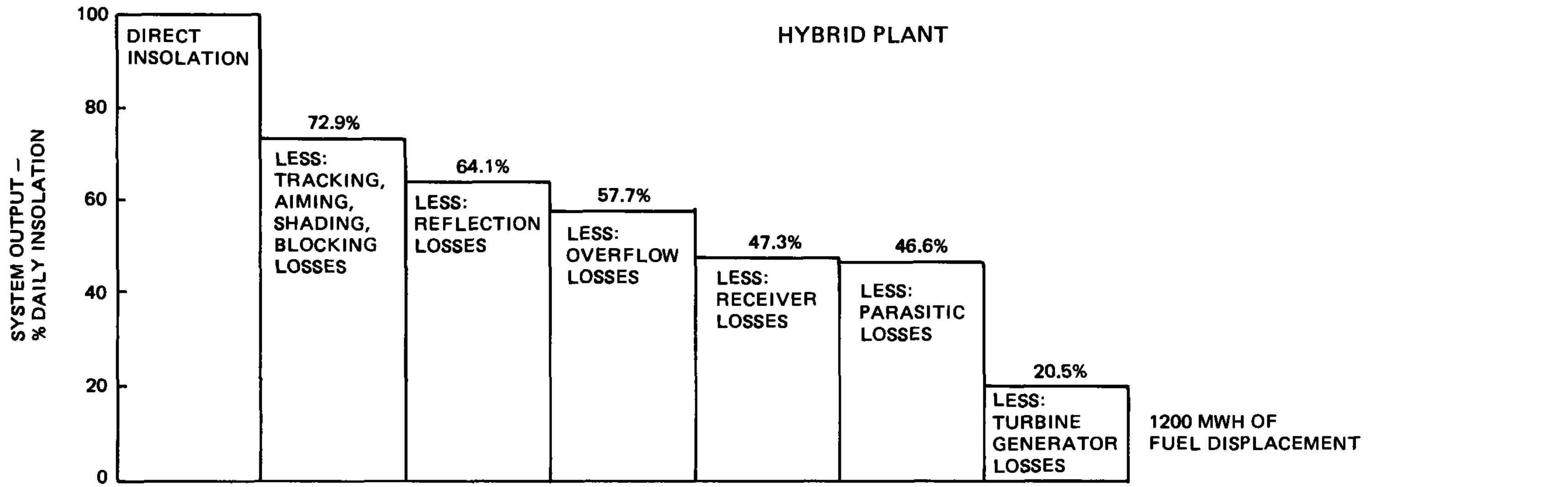
The upper chart depicts performance for the two module intermediate load stand-alone plant with six hours of thermal storage. A phase change thermal energy storage device is used here but results are available for sensible heat and thermochemical devices as well (see Figure 19). Performance has been predicted for winter, fall, spring and summer days. This daily performance data has been used to estimate the yearly energy output of the plant. The tracking, aiming, shading, and blocking losses shown in the second bar are derived from the "strawman" field performance supplied by EPRI with modifications to include solar intensity and collector field performance variation over the daily cycle.

The integrated insolation from the "strawman" field over a summer day exceeds the requirements for direct energy production and the six hour storage limit by nearly 300 MWH_e. The plant produces approximately 1700 MWH_e of electrical power for a capacity factor of 71%. Peak plant efficiencies in excess of 27.5% are achieved during periods of direct solar thermal production of electricity. Integrated daily energy conversion efficiencies of 16.5% are typical for summer day performance with thermal storage devices. Conversion efficiencies drop to 14-15% for winter operation with storage availability reduced to 3-4 hours and output reduced to approximately 1200 MWH_e.

The lower chart represents the performance of a hybrid plant with fossil-fuel backup over a summer day. The chart is similar to the upper chart prior to taking the receiver losses. Operating from insolation, the single module plant produces 1200 MWH_e of fuel displacement for a capacity factor of approximately 50%. 500 MWH_e would have to be added by the fossil-fuel heat source to match the energy production of the stand-alone plant with six hours thermal storage. The hybrid system offers the advantage of dependable capacity over a range of plant operating conditions.

Figure 6 Plant Performance–Summer Daily Cycle





4. Receiver Characteristics

Configuration Selection

Early configuration work indicated that the most desirable receiver for the high temperature concept was one supported above the tower, and with the receiver having a bottom circular aperture to admit reflected energy from the collector field. Figure 7 presents the three bottom-aperture receiver shapes analyzed during the study. The composite shape finally selected (extreme right) combines portions of the cylindrical and spherical shapes used as the initial baseline and alternative.

The table below the illustration summarizes the selection rationale. The distribution of reflected and absorbed energy was an important factor in concept selection. Detailed thermal analyses were performed first on the initial cylindrical receiver and an alternative spherical receiver shape. The absorbed heat flux and temperature on the lower interior walls of the cylinder reached much higher (and intolerable) levels than on the walls of a sphere. A careful ray-tracing analysis also indicated reflection losses from the cylinder to be 13% compared to under 4% for the sphere. A cylinder to accommodate this reflection difference would be proportionately larger.

The selected composite shape has reflection losses comparable to the spherical shape, due to the incidence of incoming sunlight on the similar lower geometry. The first reflection on a true cylinder is also low on the wall, but the field of view from this position to the aperture is significantly greater, causing greater losses.

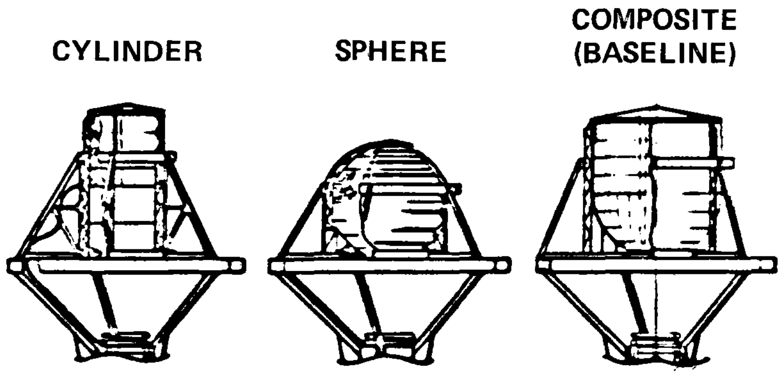
Thermal considerations dictated that the heat exchanger panels be mounted in the upper half of the receiver to escape direct solar impingement and consequent overheating. The ability to build and hang heat exchanger panel modules on a vertical wall offers technical and economic advantages. Structural assembly of a cylinder is much easier than a sphere, and the choice selected eliminated more than half of the complex curvature (and cost) of a spherical shape.

Receiver dimensions are approximately 39 meters (128 feet) high and 39 meters (128 feet) in diameter. Aperture diameter is 19 meters (62.3 feet).

Figure 7 Receiver Evaluation

BASIS

EQUAL APERTURE SIZE
EQUAL HEAT PRODUCTION FOR POWER
SAME HEAT EXCHANGER MODULE
MAXIMUM SOLAR POWER INPUT



CONSIDERATIONS FOR SELECTION	RATING		
HEAT DISTRIBUTION	POOR	BEST	GOOD
REFLECTION LOSSES (%)	13.1	3.7	3.8
OVERALL THERMAL EFFICIENCY (%)	74	87	83
ESTIMATED COST RATIO	1.0	2.0	1.5

Receiver Support Structure Design

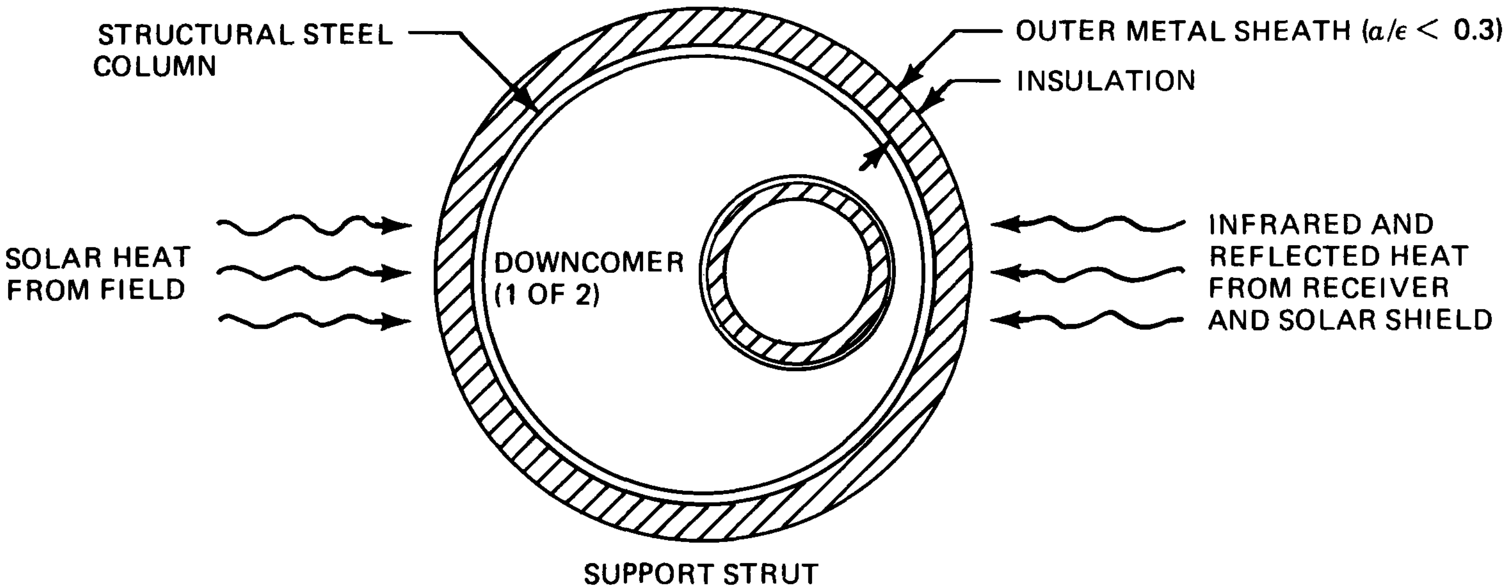
The receiver support struts located below the receiver aperture plane are heated by the solar heat from the field and also by the radiated and reflected heat flux from the receiver back through the aperture. Figure 8 illustrates the heating situation and the selected strut design. The structural steel column is protected by 5 centimeters (2 inches) of insulation covered by an outer metal sheath with a low absorptance-to-emittance ratio. The local maximum temperatures on either side of the support strut are presented in the data on Figure 8.

The risers supplying 518°C (1,000°F) helium to the receiver and the downcomers returning 816°C (1,500°F) helium to the tower must also traverse the stand-off distance between the receiver and the tower top. An initial concept had a riser-downcomer pair partially sheltered from the field by being located behind each of the five support legs to the receiver, or 5 sets in all. The concept was unsatisfactory due to the multiplicity of plumbing connections, and the amount of insulation and shields required to protect the risers and downcomers from the cavity heat and some field heat. To alleviate these problems and the associated costs, the number of risers and downcomers was reduced to two each of a larger size and with placement of each individual pipe within a separate support leg.

A cross-section of a typical downcomer is shown within the support strut on Figure 8. The design concept is to insulate the pipe containing 816°C (1,500°F) helium on the interior so that a carbon steel pipe may be used instead of the more expensive Inconel 617. The risers with 518°C (1,000°F) helium will be externally insulated, and use of Inconcel 617 is again unnecessary.

The vertical height of the support structure columns has been utilized to produce natural draft cooling of the interior structure. The heat dissipated by risers and downcomer pipes and the heat leak through the structure heat shield combine to produce internal air temperatures 30 to 60°C (54 to 108°F) above ambient. With adequate venting, the warmed buoyant air circulates upward drawing in ambient air at the bottom.

Figure 8 Receiver Support Structure Design (Typical)



LOCAL MAXIMUM TEMPERATURES	SOLAR HEATED SIDE	CAVITY HEATED SIDE
OUTER SHEATH	816°C (1500°F)	610°C (1130°F)
STEEL COLUMN WALL	370°C (700°F)	195°C (384°F)

NOTE: STRUT INTERIOR COOLED BY NATURAL AIR DRAFT

Interior Arrangement

Successful receiver operation depends on the effectiveness of heat transfer to the helium. The heat exchangers for the preferred receiver are located (see Figure 3) in the upper half of the receiver interior. In this position, they escape the direct energy impingement which would cause local hot spots. Three rows of heat exchanger panels with 70 panels per row constitute the effective heat transfer surface.

Figure 9 illustrates a heat exchanger panel module and its structural supports. Each panel consists of two offset columns of heat exchanger tubing, insulation, tubing loops to the helium manifolds, and the support structure to the outside wall. Panels are designed to be removable to facilitate easy maintenance.

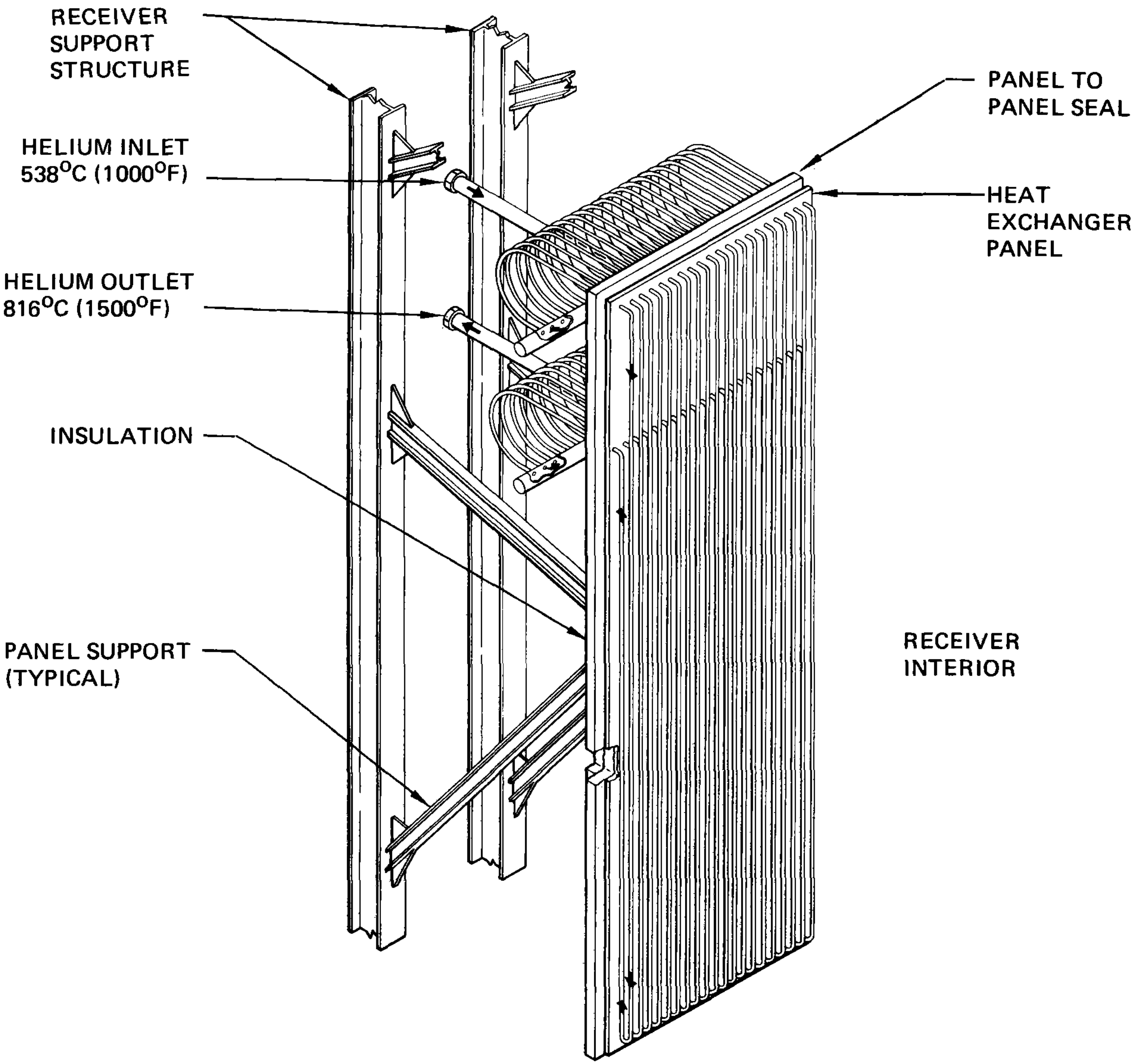
There are 20 tubes on a panel module. Each tube is U-shaped with one leg from the inlet down to the bend, and the other leg back close to the insulation up to the outlet. The difference in path length and the loops behind the insulation are to provide tube expansion during thermal cycling to keep the tube configuration in a stable position.

Material for the tubes can be either Haynes 188 or Inconel 617 alloy. Both materials were successfully thermal-cycled over a simulated 30-year lifetime. Tube dimensions are 2.54 centimeters (1 inch) outside diameter with a 0.17 centimeter (0.062 inch) wall and 9.5 meters (31 feet) length. Exposed tube surface area per panel is 17 square meters (182 square feet). Panel surface area is 9.5 square meters (103 square feet). Total panel weight is 1200 kilograms (2650 pounds).

Panel insulation behind the tubes consists of three successive layers of alumina-silica blanket, alumina-silica block and mineral wool block for a total thickness of 0.15 meters (6 inches). The panels for the lower hemispherical section (without heat exchangers) have the same insulation materials and thickness.

The helium flow rate for each panel is established nominally at 0.9 kilograms/second (2 pounds/second) to insure turbulent flow in each tube and the best heat transfer characteristics. At these conditions there is only a modest temperature difference of about 36^{o}C (65^{o}F) between the tube wall and the helium, and a tube pressure drop of 0.034-0.048 MN/m^2 (5-7 psi).

Figure 9 Interior Heat Exchanger Panel Arrangement



Receiver Temperatures

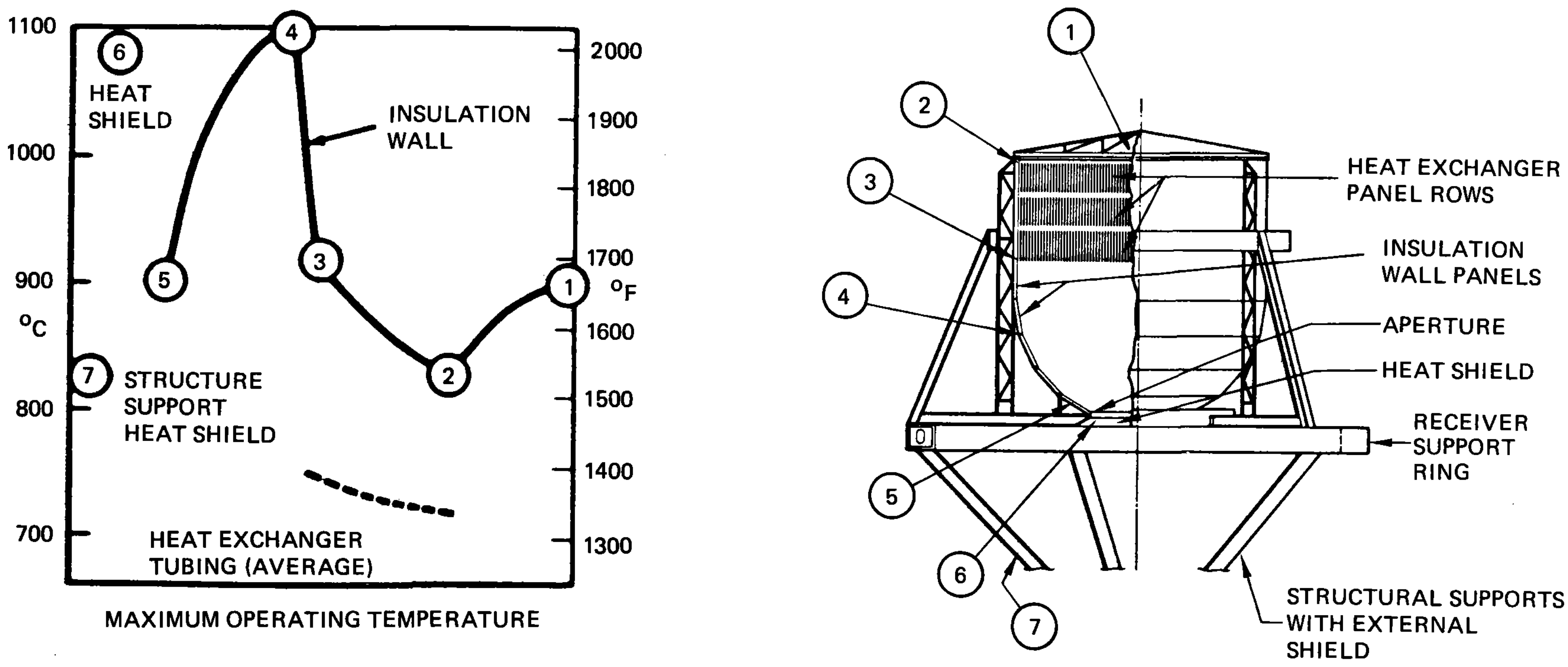
Detailed thermal analyses were performed on the baseline receiver design to determine maximum operating temperatures at key locations and to ensure that design materials had the capability to sustain these temperatures. Figure 10 shows a composite of temperatures at various receiver locations.

For the receiver interior, if the encircled numbers are followed sequentially from the roof center down the receiver walls, it is noted that the roof insulation and the insulation behind the heat exchangers (numbers 2, 3) have temperatures of 920°C (1690°F) or below.

The temperature-reducing effect of the heat exchanger tubing, shown as an average, is evident. The high temperature of 1100°C (2000°F) is sustained on the hemispherical wall where the incoming heat flux is maximum and there are no tubes for heat removal. The temperature on the hemispherical wall drops to 900°C (1650°F) near the aperture.

Outside the receiver, the temperature on the heat shield extending out from the aperture is about 1090°C (1975°F). Maximum temperatures on the receiver support structure crossing the field radiation is 816°C (1500°F) as was shown on Figure 8.

Figure 10 Receiver Temperatures



5. Material Selection and Tests

Material Selection

A number of metal alloys were chosen as candidates for high temperature tubing applications based upon present manufacturing capability, performance capability, and economic considerations. The high temperature limit was established at about 816°C (1500°F) to meet the 30-year lifetime requirement at repeated thermal cycles and stresses. Detailed screening of major property data (stress-rupture strength, creep, oxidation resistance, and metallurgical stability) resulted in selection of Haynes 188 and Inconel 617 alloys for further evaluation.

Thermal Cycling Tests

Test specimens of Haynes 188 and Inconel 617 were fabricated and subjected to 10,560 thermal cycles, while under pressure, to simulate a 30-year lifetime. The arrangement and dimensions of each test specimen are shown on the left-hand side of Figure 11. Each specimen was pressurized to 3.45 MN/m^2 (500 psi) helium pressure, and tube temperatures were cycled between 482°C (900°F) and 830°C (1525°F). The graph on the right-hand side of Figure 11 shows test conditions for each weekly run. The bottom sketch shows a schematic of the test setup featuring quartz lamp heaters and a regulated helium bottle supply.

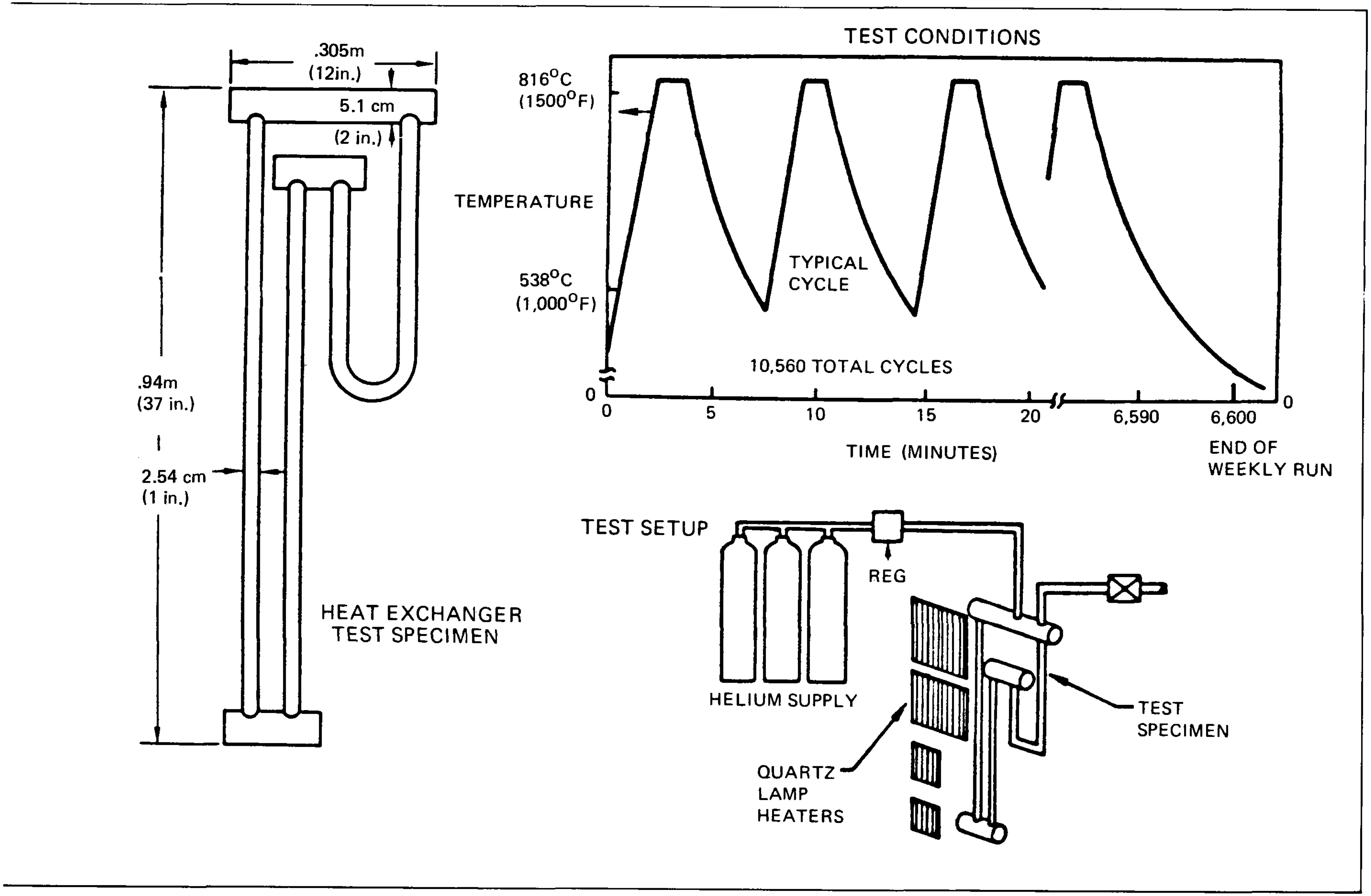
Test Results

Physical, mechanical, and metallurgical evaluations after the tests gave the following results. External and internal surfaces of the tubes were coated with a thin, tightly adherent, dark green scale. Optical measurements of wall thickness showed no evidence of material loss, indicating excellent scaling resistance. Ultimate tensile strength after test was the same as before test, while yield strength showed some drop-off. There was a larger reduction in elongation properties. The increased hardness of base metal and the weldments showed clear evidence of aging both in the mechanical testing and in the intensive micro-structure comparisons. Microscopic examination of defects in manual welds revealed no crack propagation during the tests. Sections of the scaled tube surfaces showed little, if any, intergranular oxide penetration.

Test Summary

The thermal cycling tests verified Haynes 188 and Inconel 617 as excellent materials for central receiver high temperature applications. The performance of either material was such that a final material selection could be based on economics and quantity availability.

Figure 11 Thermal Cycling Tests



Elevated Temperature Rupture Tests

Materials specialists at EPRI requested that elevated temperature tests-to-rupture be performed to determine the capability of Haynes 188 and Inconel 617 tubes to sustain temperatures in excess of the proposed service maximum of 830°C (1525°F). The information was desired to evaluate the safety hazard of accidental or intentional overheating and to increase the body of information on high temperature material behavior.

Accordingly, single tubular specimens of Haynes 188 and Inconel 617 alloys, both in the new and after-thermal cycling condition, were pressurized with helium at 3.45 MN/m^2 bar (500 psi) and thermally cycled at successively higher temperatures until stress-rupture failure occurred. The upper photo of Figure 12 shows the four subject test specimens. Test temperature levels were 871°C (1600°F), 926°C (1700°F), 982°C (1800°F), 1037°C (1900°F), and 1092°C (2000°F). Fifty cycles were performed at each test temperature level the material could sustain. Each cycle was 50 minutes at temperature with 10 minutes between cycles. Between each test temperature level, tube diameters were measured and recorded.

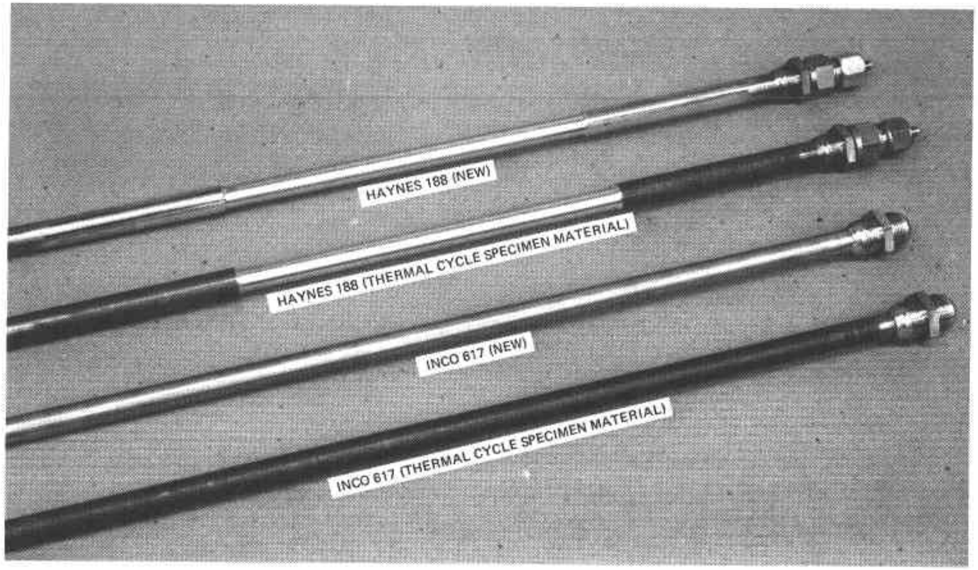
Test Results

All ruptures of test specimens occurred in the 1037°C (1900°F) to 1092°C (2000°F) range, with the Haynes 188 and Inconel 617 tubes which had been exposed previously to thermal cycling surviving the longest. All failures were noncatastrophic with small fissures occurring in the tubes permitting helium leakage without explosions or fast crack propagation. Successively more ballooning of material occurred with higher temperatures and was more evident for Haynes 188 than for Inconel 617. The lower photo on the left of Figure 12 shows the ballooning of a new-material Haynes 188 tube. The photo to the right shows the rupture site on the same tube and is typical of all ruptures. No ruptures occurred on the weld lines of the tubes.

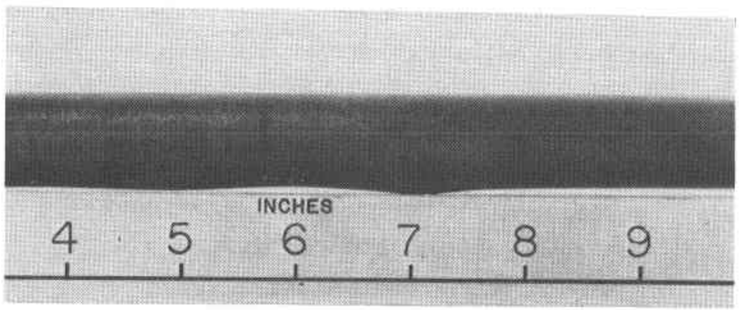
Test Summary

The tests showed material capability to withstand much higher temperatures than 830°C (1525°F) for a period of time, and that this capability is improved by prior exposure to lower temperatures. The noncatastrophic failures show the safety hazard to be small and the operational effect of a single tube failure to be isolated to itself. Welded tubing of the quality tested would probably perform as well as seamless tubing in the receiver heat exchanger application.

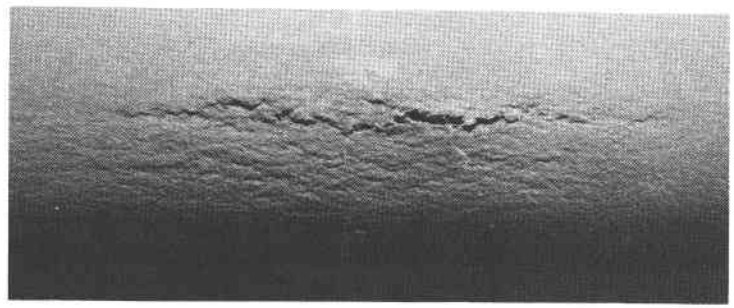
Figure 12 Elevated Temperature Rupture Tests



TEST SPECIMENS



HAYNES 188 TUBE BALLOONING
1037°C (1900°F)



HAYNES 188 RUPTURE SITE
1037°C (1900°F)

6. Cycle Analysis/Turbomachinery

Cycle Analysis

The thermal cycle selected as a result of detailed analysis has been presented on Figure 5. The high cycle efficiency of 0.44 attainable with helium reduces the sizes (and costs) of the collector field and receiver. At the defined turbine inlet temperature of 816°C (1500°F), this efficiency requires a high effectiveness (.94) in the recuperator necessitating a large surface area for heat transfer. While the recuperator cost is increased, the net plant cost is reduced.

Cycle characteristics were examined for their impact on plant operations, and showed the inherent advantages of using closed cycle gas. With a storage system, turbine inlet temperatures are normally reduced below the 816°C (1500°F) temperatures supplied by the receiver during nominal operations. For closed cycles, the efficiencies can be sustained at a high level by controlling the inventory of gas in the system. This type of control is also available for changes in solar input.

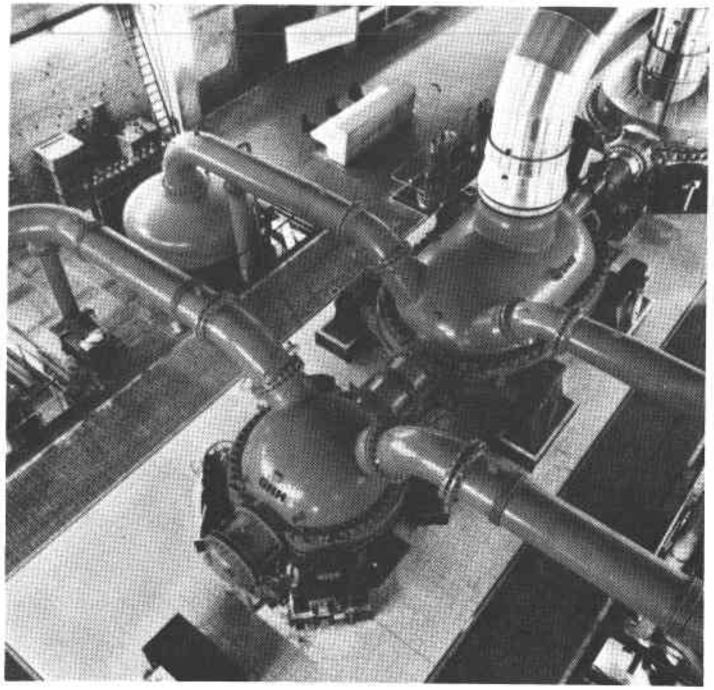
Turbomachinery Availability

Availability of helium turbomachinery in the 50 MW_e to 100 MW_e range is an important consideration in the feasibility of a closed cycle helium plant. Extensive interchanges with domestic and foreign suppliers has increased confidence that there does not appear to be any insurmountable technical problems of closed cycle turbomachinery for the commercial use intended. The design is straightforward and costs are expected to be consistent with similar air-driven turbines; however, the lead time required is increased approximately 12 months.

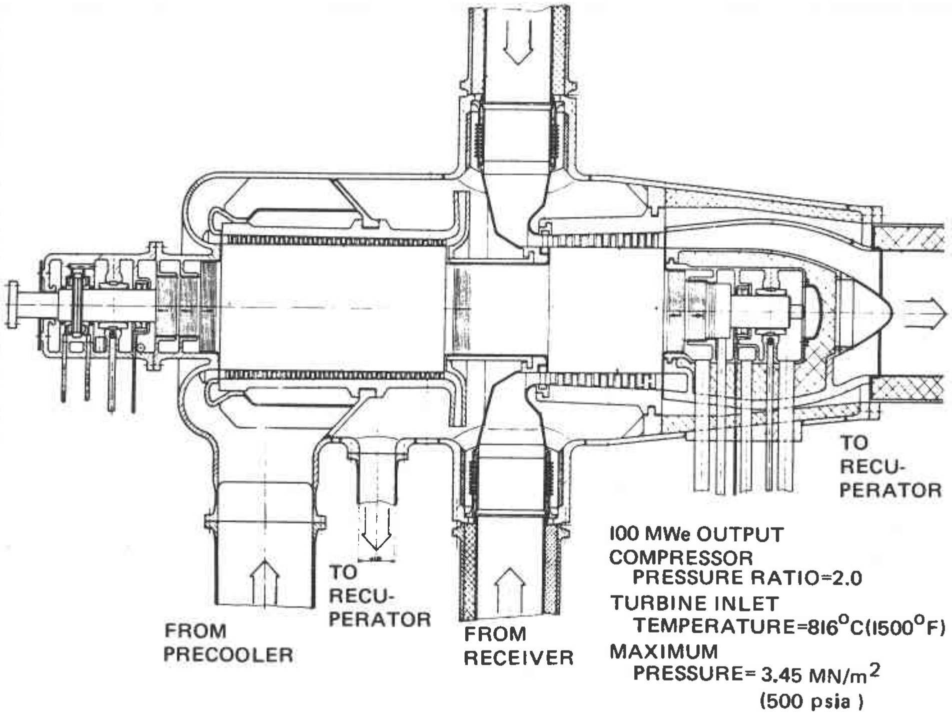
The development of helium turbomachinery has proceeded in Europe. Gutehoffnungshutte Sterkrade, A. G. (GHH) has developed a 50 MW_e turbine which is under test at a public utility in Oberhausen, Germany. A photograph of the installation is shown on the left side of Figure 13. GHH has promised support in future developmental efforts.

The drawing to the right of Figure 13 shows a specific design received from Brown, Boveri and Company, Ltd. (BBC) of Switzerland, for a 100 MW_e rated turbogroup for possible use in the baseline solar plant. Estimated availability for the unit would be 3-4 years from date of order.

Figure 13 Helium Turbomachinery Availability



HELIUM TURBINE INSTALLATION
AT OBERHAUSEN, GERMANY
HELIUM TURBINE FROM GUTEHOFFNUNGSHUTTE
STERKRADE, AG (GHH)



PROPOSED 100 MWe SOLAR PLANT
TURBOGROUP (BROWN, BOVERI AND CO.)

Alternative Cycle Comparisons

Cycle analysis was extended to making comparisons between closed helium, closed air and open air cycles. These cycles were examined for performance, cost, and qualitative considerations. A performance comparison based on efficiency has been shown on Figure 5. The efficiency design range determined for helium is 0.43-0.45 followed closely by closed cycle air at 0.40-0.42. Open cycle air would be 0.35-0.36. For comparison, the "strawman" Rankine cycle efficiency is 0.36.

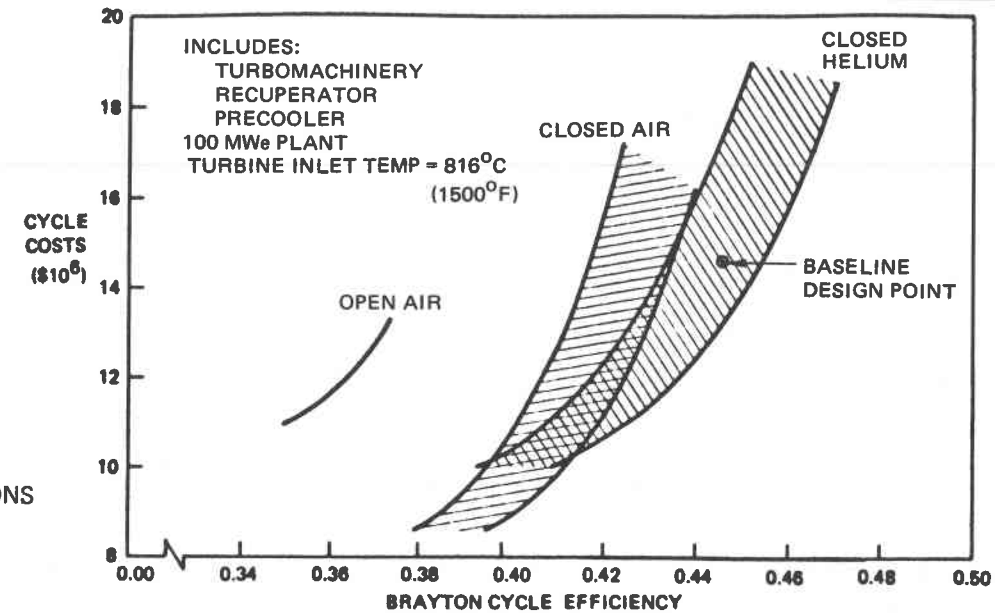
A cost comparison of equipment for the alternative cycles is shown on Figure 14. Examination shows that the closed cycles offer higher efficiencies than an open cycle for the same cost. Selection of the gas for a closed cycle that gives the highest efficiency/minimum cost is a function of efficiency level. The recuperators are larger and more costly at higher efficiencies; therefore, helium is more cost effective because of its smaller recuperator.

At lower efficiencies, the cost advantage of the helium recuperator is not enough to offset the increased cost of the helium turbomachinery, so air is the most cost effective.

The table on Figure 14 lists some of the qualitative considerations for cycle selection. Closed cycle helium and air have been grouped together due to their inherent similarities. Closed cycle systems permit operational flexibility through utilization of gas inventory control. This enables the turbomachinery to operate at nominal efficiency over a broad range of load factors. However, there is no operational experience in the United States (outside of very small units), such as with open cycle and steam cycles. The open cycle is a simple system requiring no precooler but operational flexibility is limited and efficiency is slightly lower. Steam cycles are familiar to utilities and reliability is known for many applications. The solar plant usage adds additional complexities in the high heat flux steam receiver design and in the control complexity to handle variable solar input.

Figure 14 Alternative Cycle Comparisons

COST COMPARISONS



QUALITATIVE CYCLE COMPARISONS

System	Advantages	Disadvantages
Closed Cycle	■ High efficiency ■ Smallest turbomachinery ■ Operational flexibility ■ Good bottoming cycle potential	■ No operating experience in U.S. ■ Large precooler ■ High Brayton cycle costs
Open Cycle	■ No precooler required ■ Similar to combustion turbine design ■ System simplicity	■ Limited operational flexibility ■ Large turbomachinery ■ Large recuperator
Steam	■ Proven systems available ■ Utility familiarity ■ High heat flux receiver	■ Large precooler ■ Lower efficiency ■ Complex system

7. Energy Storage

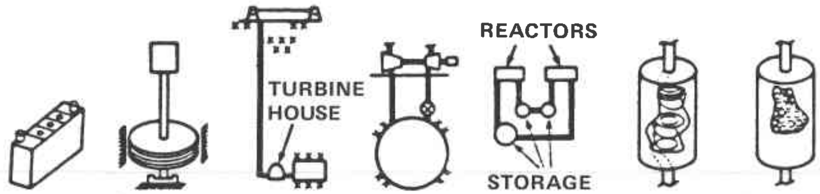
Storage Concepts

The requirements for providing 6 hours of storage for a stand-alone plant and one-half (½) hour of storage for "thermal buffering" for a hybrid plant led to a review of energy storage devices used in other applications. Public Service Gas and Electric of New Jersey (PSG&E), under joint funding from ERDA and EPRI had completed such a technical and economic assessment for electric utility applications. The PSG&E cost data for load side energy storage devices (batteries, flywheels, pumped hydro, and compressed air) is displayed on the upper chart of Figure 15. The need for thermal control as well as storage in the solar plant dictated examining energy storage devices on the source side of the generator. Three such thermal energy storage devices were examined in detail for performance and costs. Chemical, phase change, and sensible heat storage devices were conceptually designed and integrated into solar plant operations. Resultant cost estimates have been included in the table on Figure 15. The cost advantage of phase change storage is apparent for the 6-hour requirement.

Storage/Plant Cycles

The illustrations on the lower half of Figure 20 show how the thermal energy storage concepts are integrated into plant operation. The storage charge cycle is on the left, the discharge cycle on the right. The heavier lines indicate the primary fluid loops. In the charge cycle, some helium heated in the receiver is by-passed to go through the energy storage media and is returned to the circuit. When discharging without any insolation load, the receiver is by-passed with the helium receiving heat in the storage device and passing it to the turbine. For partial insolation, the receiver and the storage device can work together as shown by the dashed line in the discharge cycle. Representative situations where this occurs are for start-up, thermal buffering for insolation blockage, and switchover to storage.

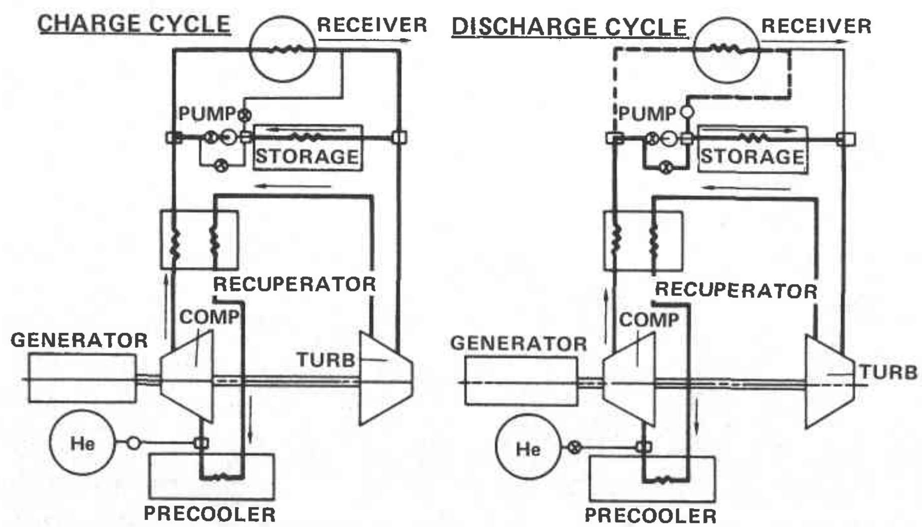
Figure 15 Storage Concepts/Cycles



	BATTERIES	FLY-WHEELS	PUMPED HYDRO	COM-PRESSED AIR	CHEMICAL	THERMAL PHASE CHANGE	THERMAL SENSIBLE HEAT
6-HOUR STORAGE	405 *	795 *	154 *	174 *	472	164	336
1/2-HOUR STORAGE "THERMAL BUFFERING"	N/A	N/A	N/A	N/A	444	74	68

* Reference RP-225, Technical and Economic Feasibility of Energy Storage for Conventional Utility Applications, Public Service Gas & Electric of New Jersey

ENERGY STORAGE INVESTMENT COST COMPARISON($/kW)



THERMAL ENERGY STORAGE/PLANT SCHEMATICS

Thermal Phase Change Storage

Energy storage in the latent heat of fusion of molten salts is an attractive concept because of the very high energy storage density and the relatively low cost. The baseline system to provide 6 hours storage has a system weight of approximately 4 million kilograms (8.8 million pounds) and a system cost, as shown in the preceding table, of \$164/kW$_e$.

Fusible salts and eutectic mixtures of those salts with substantial heat of fusion are available commercially at virtually any melt temperature. For applications with closed cycle gas turbine plants, the melt temperature range of interest is 600°C (1110°F) to 900°C (1650°F). Fluoride salts operate in this range and have high heats of fusion. They are abundant, inexpensive, and are chemically and thermally stable. The salt selected, $7C_aF_2/54KF/39NaF$, melts at 682°C (1260° F), and has a heat of fusion of 0.156 kWh/kilogram (241 BTUs/pound). The melt temperature was chosen to give the highest storage round-trip efficiency. Round-trip efficiency is the ratio of total energy (electrical equivalent) out of the storage device to that used to charge and operate the device.

A schematic diagram of the design concept in the charging mode is shown on the left side of Figure 16. The vertical arrangement maintains the liquified salt above the solid salt. The helium flow path is reversed during discharge.

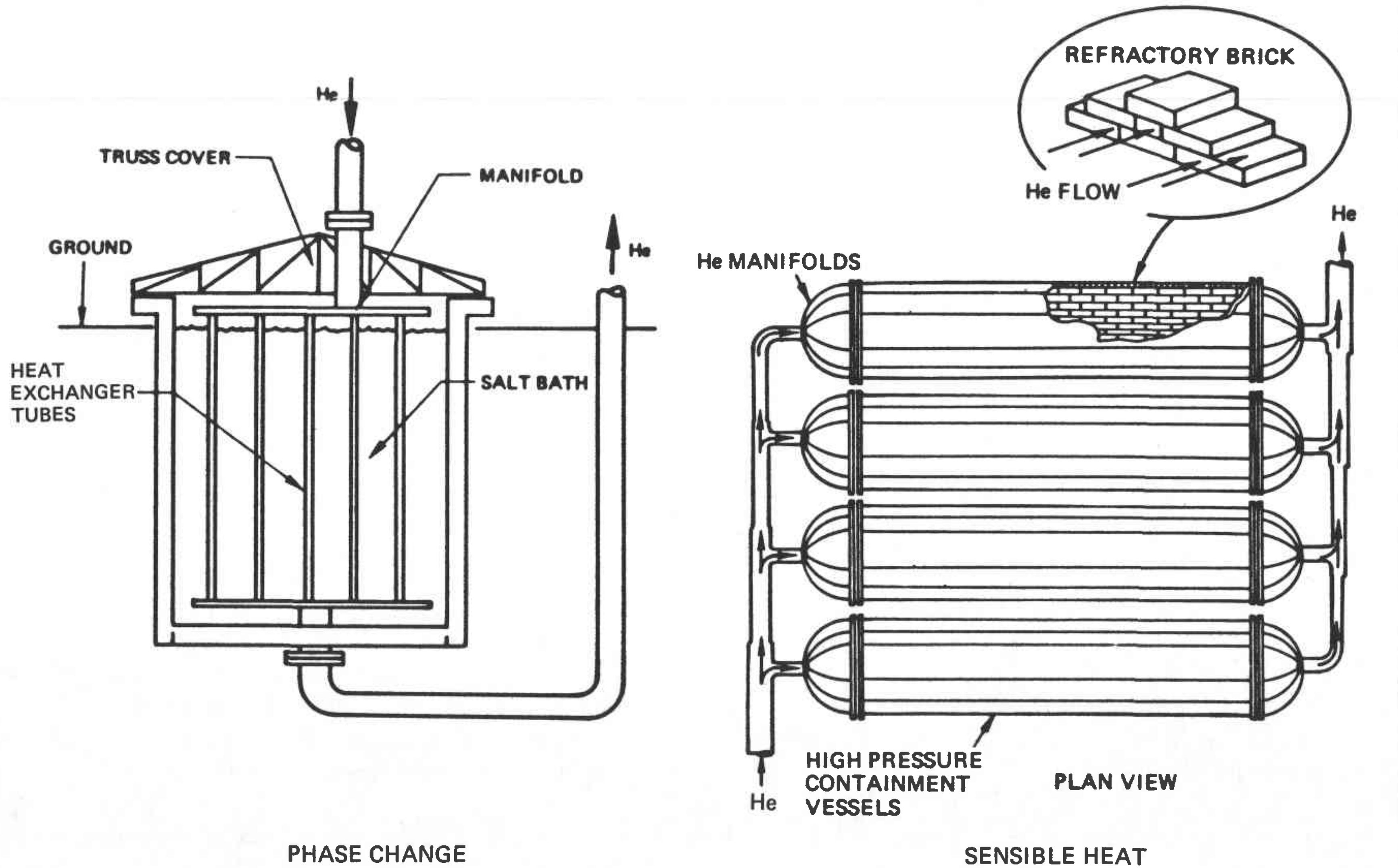
Thermal Sensible Heat Storage

Energy storage as sensible heat in materials is attractive because it is a state-of-the-art technology. Material cost per pound is small but their low specific heats result in a low energy storage density. The 6-hour sensible heat storage device most attractive in the short term has a system weight of approximately 10 million kilograms (22 million pounds) with a cost of \$336/kW$_e$.

Two design concepts were investigated in detail: a liquid NaOH bath/tube arrangement; and a solid porous media/pressure vessel arrangement. Potential corrosion problems and the relatively low thermal conductivity of molten NaOH resulted in choosing MgO refractory bricks as the storage media. MgO has superior thermal conductivity and moderate values of density and cost when compared to other refractories. High strength is retained at elevated temperatures, spalling resistance is excellent, and large quantities of MgO brick are readily available.

The refractory brick is contained in horizontally placed, insulated, cylindrical pressure vessels as is illustrated by the schematic on the right side of Figure 16. The helium flow is distributed and collected by a piping system to each vessel.

Figure 16 Thermal Energy Storage Concepts



Thermochemical Storage

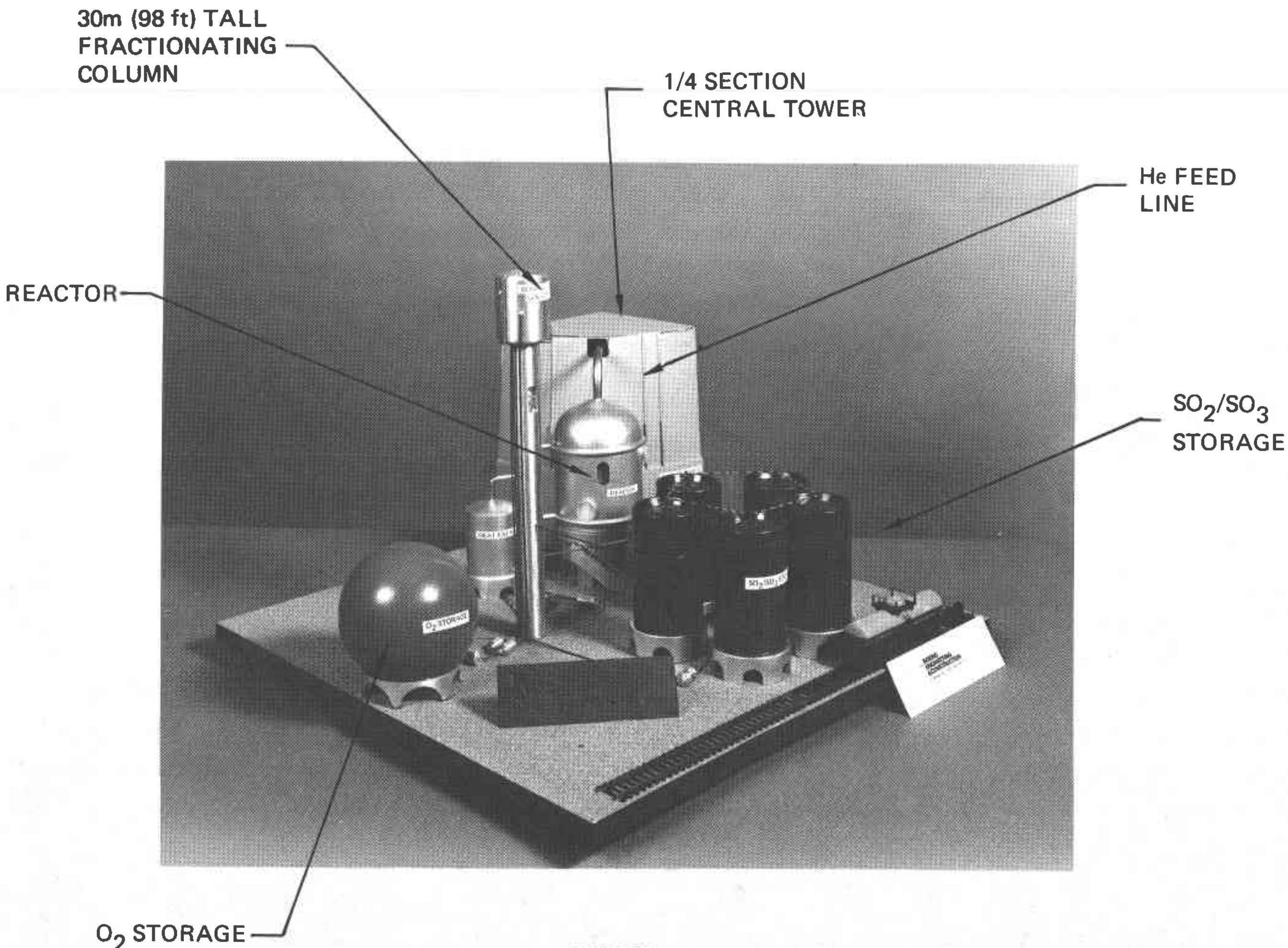
Thermal energy storage in reversible thermochemical reactions was also examined in detail because such systems offer the highest energy storage density of any of the concepts studied. The concept/reaction chosen resulted in a weight of 3.6 million kilograms (8 million pounds). Costs were higher than other concepts at \$472/kW$_e$. The technology, while considered long-term for the high temperature solar plant application, is based on well understood commercial chemical processes.

The Rocket Research Corporation of Redmond, Washington, carried out the investigation of thermochemical storage under subcontract to Boeing. The reaction selected is based on the reversibility of the dissociation of sulfur trioxide into sulfur dioxide and oxygen. The reaction involves absorption of 0.190 kWh/kilogram (532 BTUs/pound) in dissociating sulfur trioxide in the charging process, and release of the same amount of heat when SO_2 and O_2 are recombined in the discharge process.

These processes occur in a reactor where a catalyst is required to make the reactions proceed. Catalyst selection remains a critical materials question. An aggressive development program is required to develop a low cost, high temperature resistant catalyst. Two of the three reaction constituents are conveniently stored at ordinary pressures as liquids, and the third constituent, O_2, is commonly processed and stored as a gas. The option exists for cryogenic storage of the O_2 constituent. The reaction temperature is controllable and all constituents will remain in frozen equilibrium in the absence of the catalyst.

A photograph of a thermochemical storage model is shown on Figure 17 with key elements identified. The fractionating column separates the dissociation or reaction products so that the undissociated or unreacted species can be returned to the reactor. The fractionating efficiency depends on the difference of boiling points of the compounds to be separated. In the case of the selected thermochemical storage system, the boiling points are spread apart far enough to allow complete separation of all constituents.

Figure 17 Thermochemical Storage Concept



MODEL

8 Plant Operation and Cost

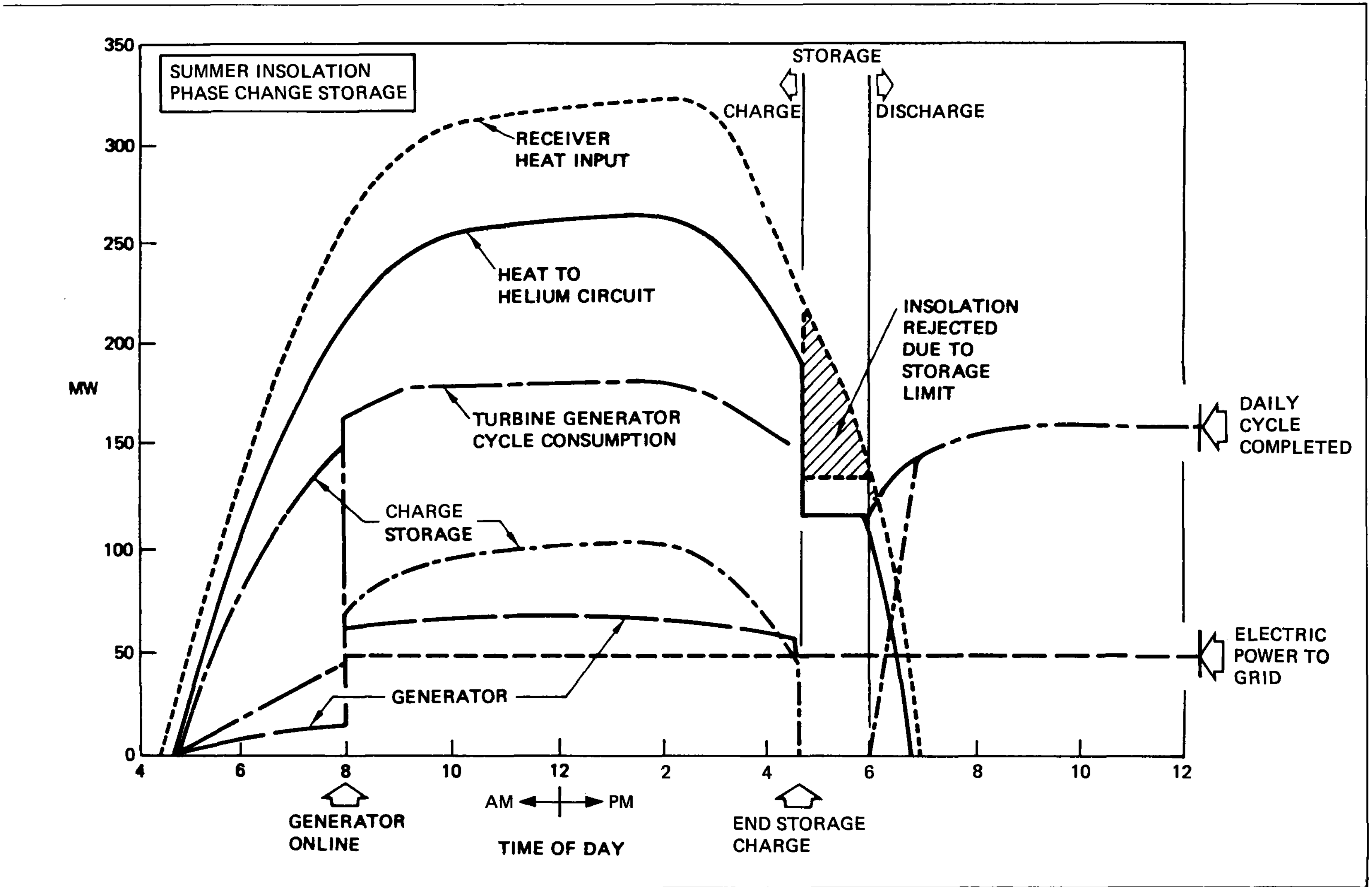
Plant Operation

Prior sections have considered major plant subsystems; namely, the receiver, thermal cycle, and thermal energy storage. A significant portion of the study was devoted to integrating these subsystems into a solar plant and determining its operational modes and performance. A math model was developed to support plant design and performance analysis work. Figure 18 is typical of the results obtained for a solar plant operating over a representative daily cycle. The example chosen is for a summer day and a plant with a phase change storage device.

The upper curve shows the receiver heat input as it comes from the collector field. The solid curve below it is the amount of heat absorbed in the helium circuit. This heat is divided into components (also indicated) for direct use in the turbine generator cycle and to charge storage. The particular mode of operation shown uses residual energy in storage to start the system. The first few hours insolation are used to charge storage with a limited amount of heat used in the turbine-generator to produce the power to run the storage mode. At 8 AM, in this example, the generator is put on-line to furnish 50 MW_e to the grid from the plant module. Two plant modules thus furnish the required 100 MW_e.

The plant module runs uninterrupted until shortly before 5 PM when the 6-hour storage limit is achieved. At this time, and until 6 PM the amount of heat from the collector field is excessive and some heat must be rejected (cross-hatched region). At 6 PM, the storage system begins discharging as the receiver output drops off until finally the plant is running only on heat from the phase change storage device. The daily cycle in summer is completed shortly after midnight, the plant having generated power for over 16 hours.

Figure 18 Plant Operation–Representative Daily Cycle



Plant Module Seasonal Performance

The upper chart of Figure 19 summarizes plant module performance for 4 seasonal days for each of the three thermal energy storage devices incorporated into the plant. For a given seasonal day, there is little variation between the three devices in plant operational hours or daily storage hours. Storage time during winter is curtailed by the available insolation hours.

The sensible heat storage subsystem has slightly higher plant efficiencies than the other two storage subsystems due to lower parasitic load requirements and a better round trip storage efficiency. Energy losses accounted for in the system are those due to pressure, conduction, availability (supply temperature variations), and parasitic power. Parasitic requirements for the thermochemical subsystem are the highest, but inventory (mass flow) requirements through the turbine and storage are much reduced.

Plant Module Yearly Average Performance

The lower chart of Figure 19 shows some of the key performance factors for solar plants utilizing thermal energy storage devices when averaged over a year.

The overall plant conversion efficiency is the result of going through the plant subsystem efficiency chain. Solar availability is the percent of operating time available during the year. The excess insolation percentage is the amount of energy not used due to limiting the storage devices to six hours. The storage utilization factor is the fraction of six hours storage used over the year. The sensible heat storage device outperforms the phase change and thermochemical devices in all the categories listed in the table. The thermochemical device, being less subject to equipment capacity limits, has excellent potential for longer storage periods. However, as shown on Figure 15, the higher costs of the sensible heat and thermochemical subsystems over that of the phase change subsystem makes the latter a preferred current choice.

Figure 19 Plant Module Performance Summaries

	SEASON	PLANT OPERATION (HOURS/DAY)	*PLANT EFFICIENCY η_p(%)	MAX-MIN GENERATOR CAPACITY (MW_e)	MAXIMUM RECEIVER MASS FLOW (kgm/sec)	MAXIMUM TURBINE MASS FLOW (kgm/sec)	MAXIMUM STORAGE MASS FLOW (kgm/sec)	CHARGE TO DISCHARGE RATIO	ROUND TRIP STORAGE EFF. (%)	DAILY STORAGE (HOURS)
PHASE CHANGE STORAGE	WINTER	11.7	14.9	63-50	196	195	195	0.49	60	3.1
	SPRING	16.3	16.3	70-50	256	195	195	0.72	64	6.0
	SUMMER	16.8	16.5	71-50	259	195	195	0.72	62	6.0
	FALL	14.3	16.5	67-50	233	195	195	0.63	64	5.3
SENSIBLE HEAT STORAGE	WINTER	12.2	15.4	58-50	168	195	195	0.58	68	3.5
	SPRING	16.4	17.3	66-50	225	195	195	0.76	74	6.0
	SUMMER	16.9	17.7	57-60	231	195	195	0.74	73	6.0
	FALL	15.1	17.5	62-50	192	195	195	0.64	74	6.0
THERMO-CHEMICAL STORAGE	WINTER	10.6	14.7	72-50	179	133	92	0.55	56	2.9
	SPRING	15.8	15.6	81-50	212	149	92	0.76	58	5.8
	SUMMER	16.9	16.1	82-50	215	151	92	0.78	57	6.0
	FALL	13.8	15.9	77-50	196	142	92	0.66	57	4.8

SEASONAL PERFORMANCE

$$^*\eta_p = \frac{\text{NET GENERATOR ENERGY}}{\text{SPECULAR INSOLATION ENERGY}}$$

	OVERALL PLANT CONVERSION EFFICIENCY * (%)	SOLAR AVAILABILITY (%)	EXCESS INSOLATION (%)	STORAGE ROUND TRIP EFFICIENCY (%)	STORAGE UTILIZATION FACTOR (%)
PHASE CHANGE CONCEPT	15.9	61.8	4.6	62.1	84.9
SENSIBLE HEAT CONCEPT	16.9	63.4	8.5	72.1	89.8
THERMOCHEMICAL CONCEPT	15.5	60.5	3.4	57.2	81.4

YEARLY AVERAGE PERFORMANCE

Plant Cost Comparisons

The EPRI "strawman" included cost accounts for intermediate stand-alone and hybrid plants based on a central receiver concept using a steam/Rankine cycle. These "strawman" plant accounts were furnished so side-by-side cost comparisons could be made with central receiver solar plant designs using alternative cycles. Relevant account items for closed cycle helium plants have been determined, and results are shown on Figure 20. Total costs for the two concepts are comparable for the stand-alone solar plants. The hybrid plant utilizing the helium cycle is about 11% more costly than the steam/Rankine cycle "strawman" hybrid plant.

The "strawman" plants were used as a baseline for the cost comparison; i.e., the costs of the higher performance helium system were adjusted so the plants would have the same direct electrical power production. Because the helium cycle efficiency of 0.44 exceeds the 0.36 of the "strawman" plants, major adjustments occurred in field size and cost. These changes are displayed within the horizontal bar in Figure 20. This comparison also shows that a significant increase in the cost of collectors would give the helium system a definite cost advantage. Other major differences are in the receiver, turbine equipment, and miscellaneous plant equipment accounts. The cost of helium risers and downcomers between tower top and bottom is included in the miscellaneous plant equipment account.

Thermal energy storage using the phase change concept for the helium system verified the "strawman" estimate for the stand-alone plant. Other storage concepts would have yielded much higher costs. The major difference in the hybrid plant totals is in the provisioning for one-half hour storage. Prorating the six-hour storage to one-half hour storage in the "strawman" plants ($180/kW_e to $15/kW_e) is probably not realistic.

Comparisons of total costs for these intermediate plants are reasonably close and provide an encouraging basis for continued consideration of closed cycle gas plants.

Figure 20 Plant Cost Comparisons ($/kWe)

PLANT TYPE	STAND-ALONE		HYBRID	
	STRAWMAN	HELIUM	STRAWMAN	HELIUM
COLLECTOR AREA (km^2)	1.0	0.84	0.5	0.42
STORAGE TIME (HOURS)	6	6	0.5	0.5
ACCOUNT				
LAND	2	2	1	1
STRUCTURE AND FACILITIES	44	44	51	51
HELIOSTATS*	600	505	300	258
CENTRAL RECEIVER/TOWER**/HEAT EXCHANGER	95	197	68	98
STORAGE TANKS	180***	164	15***	74
BOILER PLANT	–	–	73	73
TURBINE PLANT EQUIPMENT	80	119	80	105
ELECTRIC PLANT EQUIPMENT	21	20	21	20
MISC PLANT EQUIPMENT	4	28	4	16
ALLOWANCE FOR COOLING TOWERS	20	15	20	15
TOTAL DIRECT COST	1,046	1,094	633	711
CONTINGENCY ALLOWANCE AND SPARE PARTS ALLOWANCE (5%)	52	55	32	36
INDIRECT COSTS (10%)	105	109	63	71
TOTAL CAPITAL INVESTMENT (1975)	1,203	1,258	728	818
INTEREST DURING CONSTRUCTION (15%)	180	189	109	123
TOTAL COST AT YEAR OF COMMERCIAL OPERATION (1975 DOLLARS)	1,383	1,447	837	941

*COLLECTOR COST–$60/m^2$

**TOWER HEIGHT–260m (2 and 1 TOWER(S), RESPECTIVELY)

***THERMAL STORAGE COST–$30/kWHe

9. Bench Model Receiver Test Program

Definition of the 100 MW_e receiver concept provides the performance, configuration guidelines, and materials to define a bench model receiver program. For such a test program, a 1 MW_{th} bench model receiver is to be designed and tested.

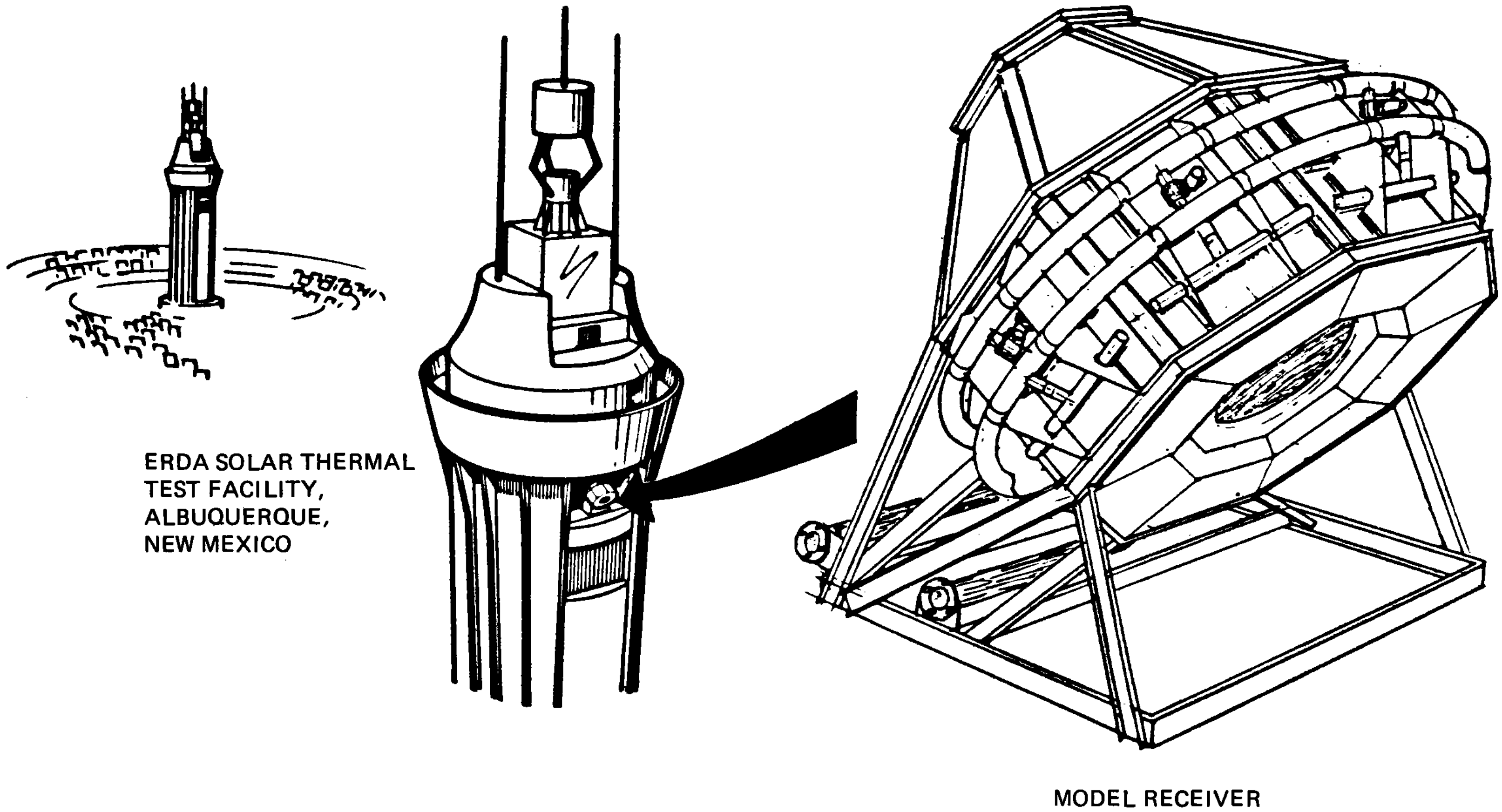
The purposes of scale model testing are to validate the technology for gas-cooled central receiver power plants, to verify the design concepts proposed for commercial size plants, and to gain test experience with the selected high temperature materials. The 1 MW_{th} bench model receiver design will duplicate the operational characteristics of the 100 MW_e receiver. In particular, the bench model will exhibit all the energy transport functions of the commercial receiver; and utilize as many of the materials, manufacturing processes, and design details as possible. Technology derived from the program will be applicable to either open or closed cycle, and various working fluids, specifically including helium.

Two program phases are planned. The first phase of approximately 16 months would include design definition and development, test planning, fabrication, and functional testing. The second phase of about 8 months would be comprised of test setup, checkout, test conduct and data evaluation.

The model receiver is configured for testing in the 5 MW_{th} ERDA Solar Thermal Test Facility at Albuquerque, New Mexico, as shown on Figure 21. With minor modifications the receiver concept shown might be capable of being tested in the CNRS Solar Energy Laboratory at Odeillo, France. A third test option using electric heat is also retained in the event that solar test facilities cannot be available in the appropriate time frame. Test of the model receiver at the 5 MW_{th} ERDA facility would be scheduled for the second quarter, CY 1978.

The preliminary design of the bench model receiver shows an octagonal shape with eight independently controlled heat exchanger panels around the inner periphery. Each heat exchanger panel has 48 Inconel 617 tubes in a U-shaped configuration. Receiver walls are steel and lined with 0.15 meters (6 inches) of high temperature insulation as in the commercial receiver concept. A flow control valve for each heat exchanger panel will regulate mass flow and gas temperature.

Figure 21 1 MW_{th} Bench Model Receiver Program



10.0 Recommendations

Successful completion of the closed cycle, high temperature central receiver study provides confidence that solar power plant development based on the study results should proceed. A development schedule is shown on Figure 22. The 1 MW_{th} bench model receiver design and test program shown has recently been initiated with an EPRI Contract (RP377-2) with Boeing. This program will verify the gas-cooled, high temperature receiver. The next step is the development, installation, and operation of a pilot plant to simulate and demonstrate commercial plant operation. For this pilot plant to be operational in 1981, preliminary design should be targeted for the last half of 1977. The initial pilot plant should be a 10 MW_e scale plant. As currently conceived, the plant would employ a quadrant of a collector field to make its output 2.5 MW_e. The option exists to expand the plant to full 10 MW_e capability by expanding the collector field. This is shown by the dashed lines on Figure 22.

Prior to initiating the 10 MW_e scale pilot plant preliminary design in late 1977, system design trades should be initiated in several areas to support the plant definition. These trades should consider the initial 2.5 MW_e plant and the growth version.

- Receiver–Evaluate orientation to a North facing quadrant field and to a circular ring quarter field. Evaluate whether the same receiver can be used at one pressure for a 2.5 MW_e plant and at a higher pressure for a full 10 MW_e plant.

- Turbomachinery–Define equipment and costs for closed cycle machinery applicable to 2.5 MW_e and 10 MW_e output. Examine methods to utilize same equipment in both applications.

- Collector Field Configuration–Determine impact of field configuration on tower height and receiver aperture sizing for the 2.5 MW_e plant and a 10 MW_e version.

- Precoolers–Examine both wet and dry cooling as to performance advantages and costs.

- Plant Operations–Define subsystem performance and predict operational performance in a pilot plant over seasonal daily cycles.

- Plant Control–Determine plant control schemes for maintaining plant operation with minimum impact of transient conditions.

In addition, long term creep rupture tests should be initiated on the superalloys, Haynes 188 and Inconel 617. The long term effects on these materials at sustained temperatures of 816°C (1500°F) should be determined.

Figure 22 Solar Power Plant Development Schedule