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

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Prepared By
Boeing Engineering and Construction
Seattle, Washington

ELECTRIC POWER RESEARCH INSTITUTE

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

Interim Summary Report

Research Project 377-1

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Table of Contents

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. Two primary methods of converting solar energy to electrical power are direct energy production by solar (photovoltaic) cells, and thermal energy conversion in conjunction with a turbine-generator set. An application of the latter process is the subject of this report.

The attractiveness of solar energy as a source due to its availability, inexhaustible supply, and inherent cleanliness has prompted major sponsoring organizations such as the Electric Power Research Institute (EPRI), the National Science Foundation (NSF), and the Energy Research and Development Administration (ERDA) to explore this potential. NSF previously has sponsored system and subsystem studies of conceptual designs for steam-turbine generation solar thermal power plants. The Aerospace Corporation completed for NSF a mission analysis of solar thermal power plants that included siting considerations, central receiver and distributed solar plant systems, and integration into electric utility grids. That effort provides excellent background material for the study being reported. ERDA has recently initiated preliminary conceptual design studies for a 10 MW_e central receiver solar thermal plant.

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

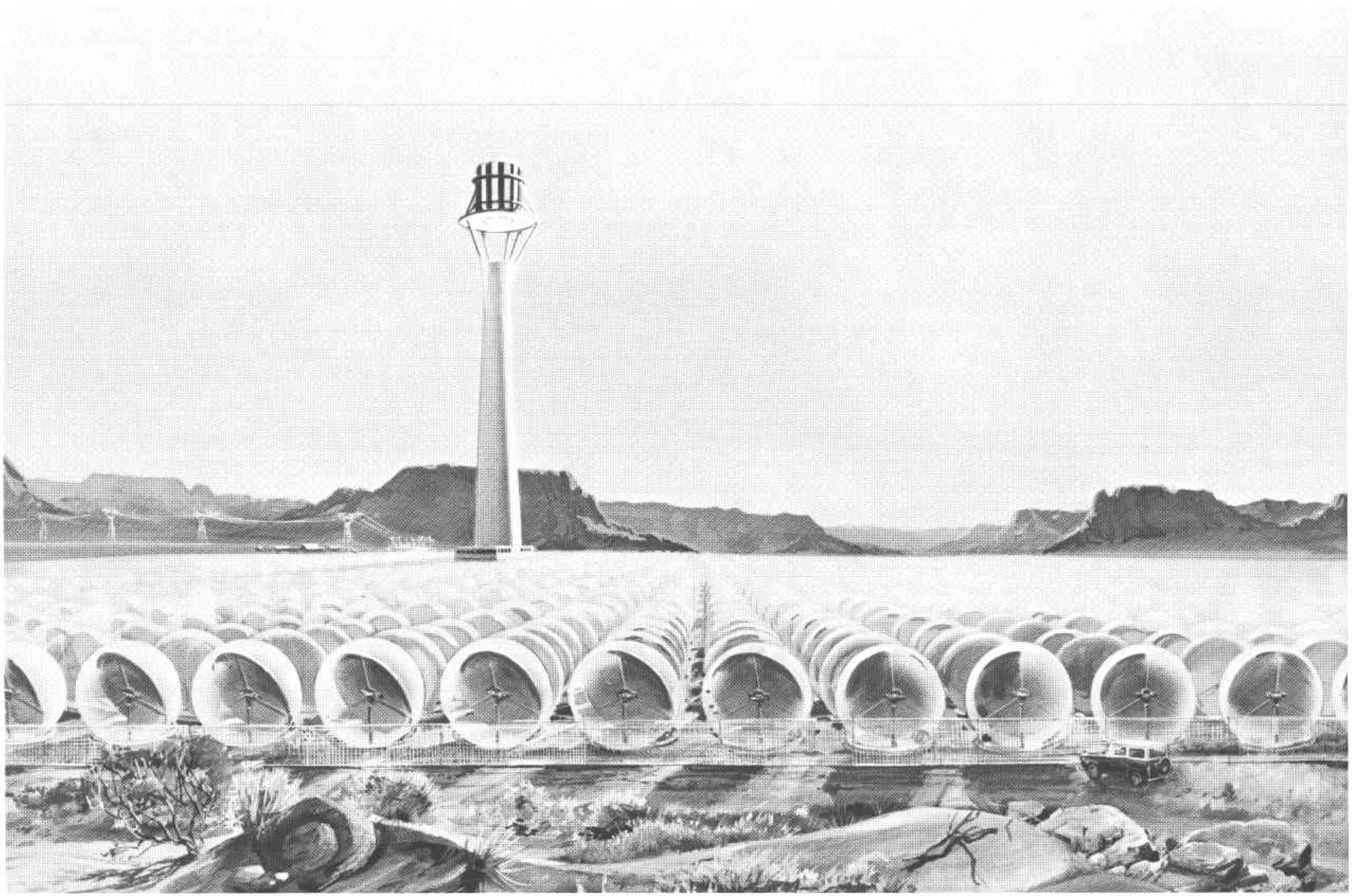
The information developed during the first nine months of a continuing contract period is presented in this interim summary report. An overview is provided of the results through August 1975. For readers desiring additional detail, a detailed interim technical report is available from EPRI.

The facing page illustrates a central receiver mounted on a tower in a collector field. Reflected solar energy from the collectors 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 passed through a turbine located within the tower. The high temperature and cycle properties of helium combine to produce a most efficient conversion to electrical power by the turbine-driven generator.

The study objectives include central receiver conceptual design, as well as considerations of the system, materials, test model, and costs to implement the receiver. Specifically, the study was directed to:

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

Significant progress towards meeting these objectives has been made during the initial study period and is summarized in the following pages. The study continuation extends and adds definition to the above objectives. Redirection received from EPRI after the interim study phase modifies the continuing study to be in concert with other solar thermal conversion programs. The nature and impact of the redirection has been included in the text. The project will be completed in May 1976.



2. Design Requirements

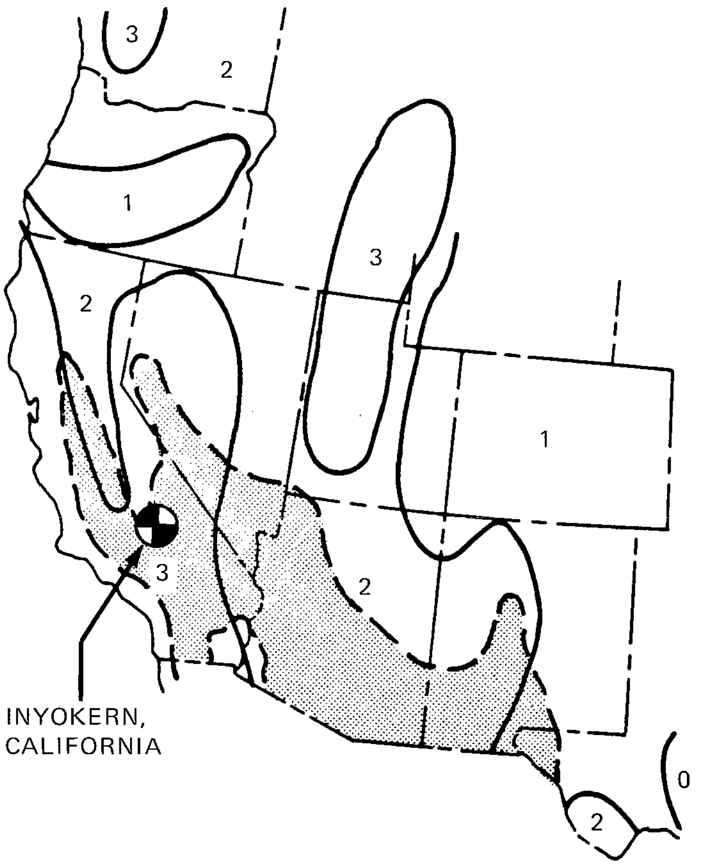
Initial study requirements called for design of a high temperature central receiver appropriate to 100 MW_e output and the use of closed cycle helium as the working fluid. The Solar Thermal Conversion plant site was specified as Inyokern, California, based on the availability of isolation data from the predecessor Aerospace Corporation study. Figure 1 shows environmental data for Inyokern in terms of seismic risk map and isolation profiles. The solid lines of the left side of the figure show Inyokern is in the highest seismic risk zone, Zone 3. The design impact of Zone 3 is to double tower weight and enlarge the volume of the tower base to accommodate shear loads greater than those required for Zones 0-2. Increased strength is required for the tower, receiver, supports, and mounted equipment. The shaded area indicates that Inyokern is in the region with the highest number of sunshine hours per year. The right side of the figure shows averaged insolation for four specific days. The interim study used the conditions for noon on June 21, to size the receiver/plant to produce 100 MW_e.

High temperature materials consistent with the state-of-the-art were to be used to achieve long equipment lifetimes, and winds of 80 mph, with gusts to 120 mph were to be accommodated. Major study assumptions were to keep tower height below 315 meters (1,000 feet), and to assume energy storage was available, but not to be used as a factor for interim design.

The above requirements and assumptions have been modified for the study continuation to design for 50 MW_e with 6 hours storage for each intermediate plant module (2 modules/plant), and for 100 MW_e with 0.5 hours storage for a hybrid plant. The associated tower height has been specified at 260 meters (855 feet).

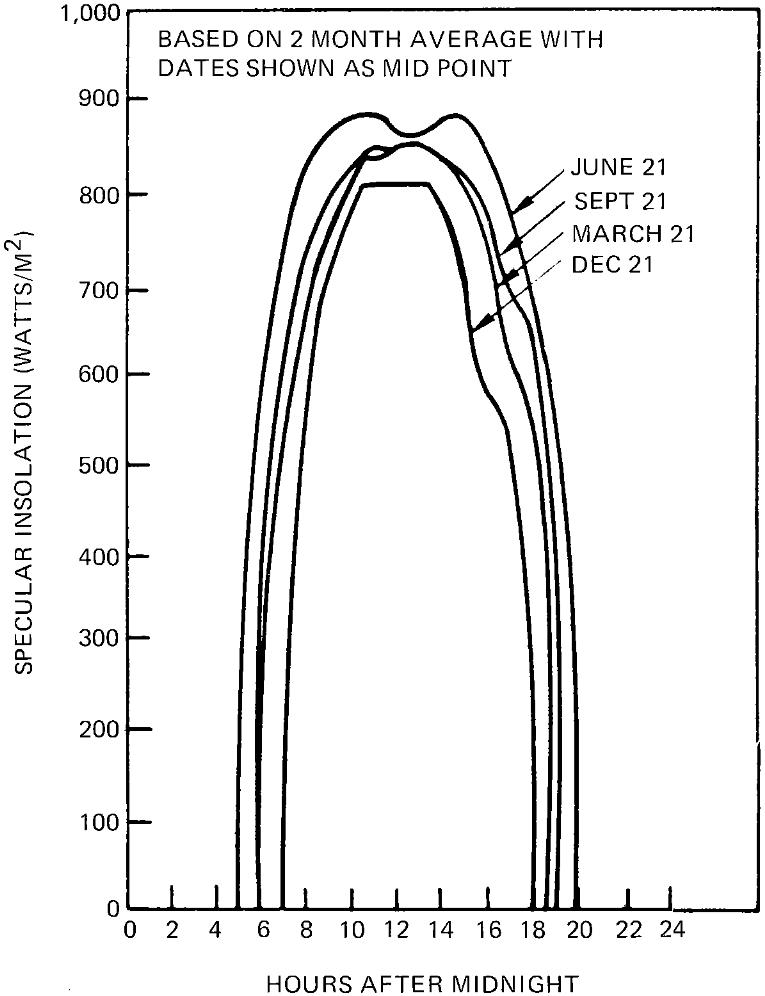
Figure 1 Environmental Requirements

SEISMIC RISK MAP – ALGER MISSEN (1969)



SHADED AREA 3,400 TO 4,000 HOURS OF SUNSHINE/YEAR

INSOLATION PROFILE –
INYOKERN, CALIFORNIA



3. Summary of Results

Preliminary results show that a high-temperature central receiver, employing closed cycle helium, is an excellent choice for solar thermal conversion plants. The concept is technically feasible and shows promise of being cost-effective due to the high thermal engine efficiency obtainable with helium and the simplicity of associated plant equipment. The concept and the significant results for each of the major feasibility criteria are summarized in this section.

Receiver Concept

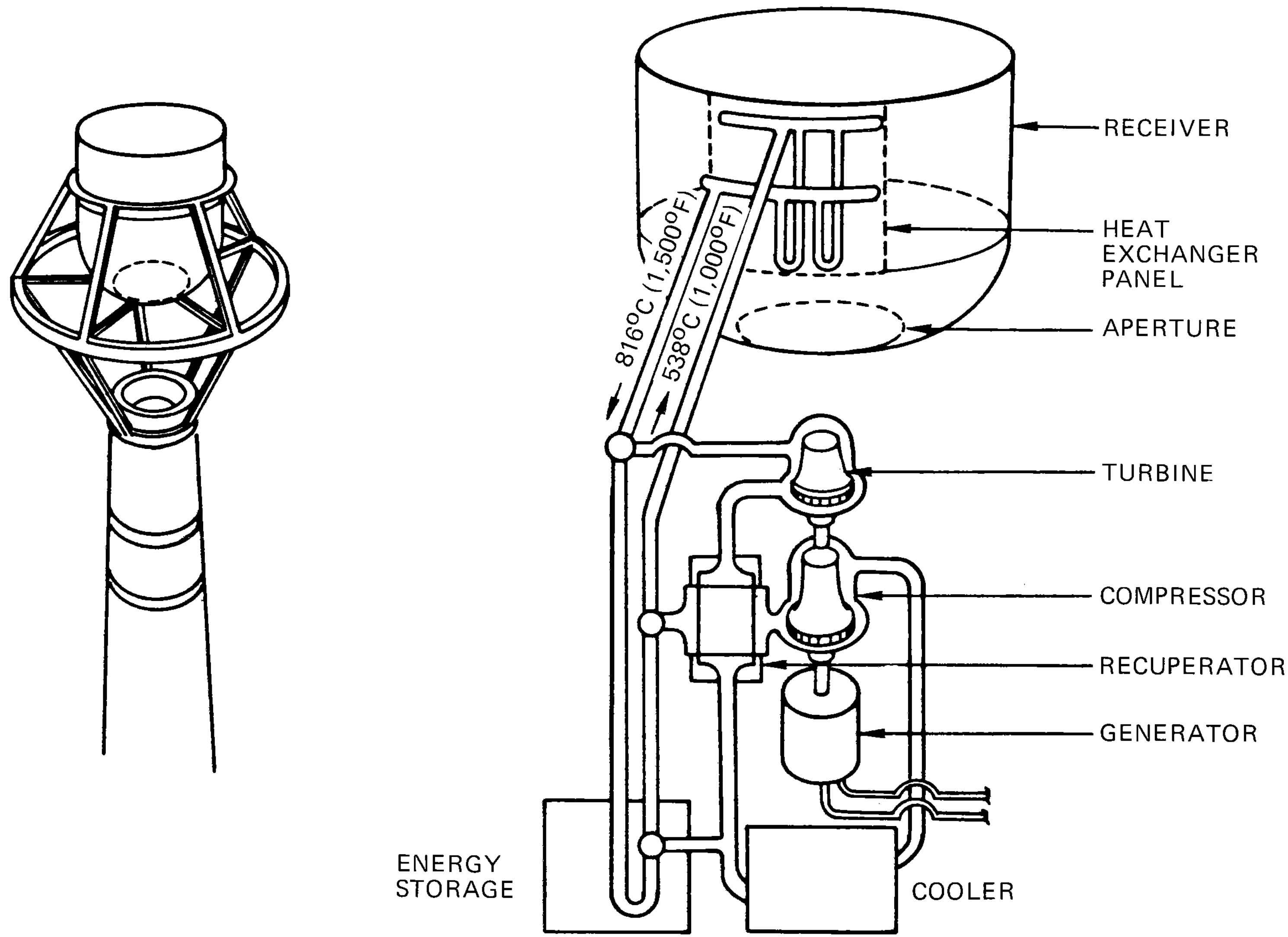
The most promising concept is displayed on Figure 2. The picture at the left illustrates the selected central receiver supported above the tower. An aperture at the receiver bottom admits the reflected energy from the collector field into the receiver interior. The receiver configuration has a hemispherical lower section and a cylindrical upper section. The schematic on the right identifies the major equipment of the solar plant. Heat exchanger panels are mounted on the upper cylindrical section and facilitate heat transfer 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 as high as possible to elevate cycle efficiency, yet remain within the state-of-the-art of high temperature metals. The associated conversion and helium processing equipment shown below the receiver would be contained within the tower. An alternative thermal energy storage loop is shown, but was not included in receiver capability during the interim study.

Receiver Lifetime

Materials are available to withstand the high temperatures encountered under repeated thermal cycling of the receiver and heat exchanger tubes under daily operation. A thermal cycling test simulating a 30-year lifetime at expected temperature extremes and high pressure has been completed on one candidate material, Haynes 188 alloy. No adverse effects were detected for the design temperature limitation of 816°C or the internal pressure of 34 bar (500 psi). Another candidate material, Inconel 617, will be tested in the study continuation.

Figure 2 Receiver Concept/Schematic



System Performance

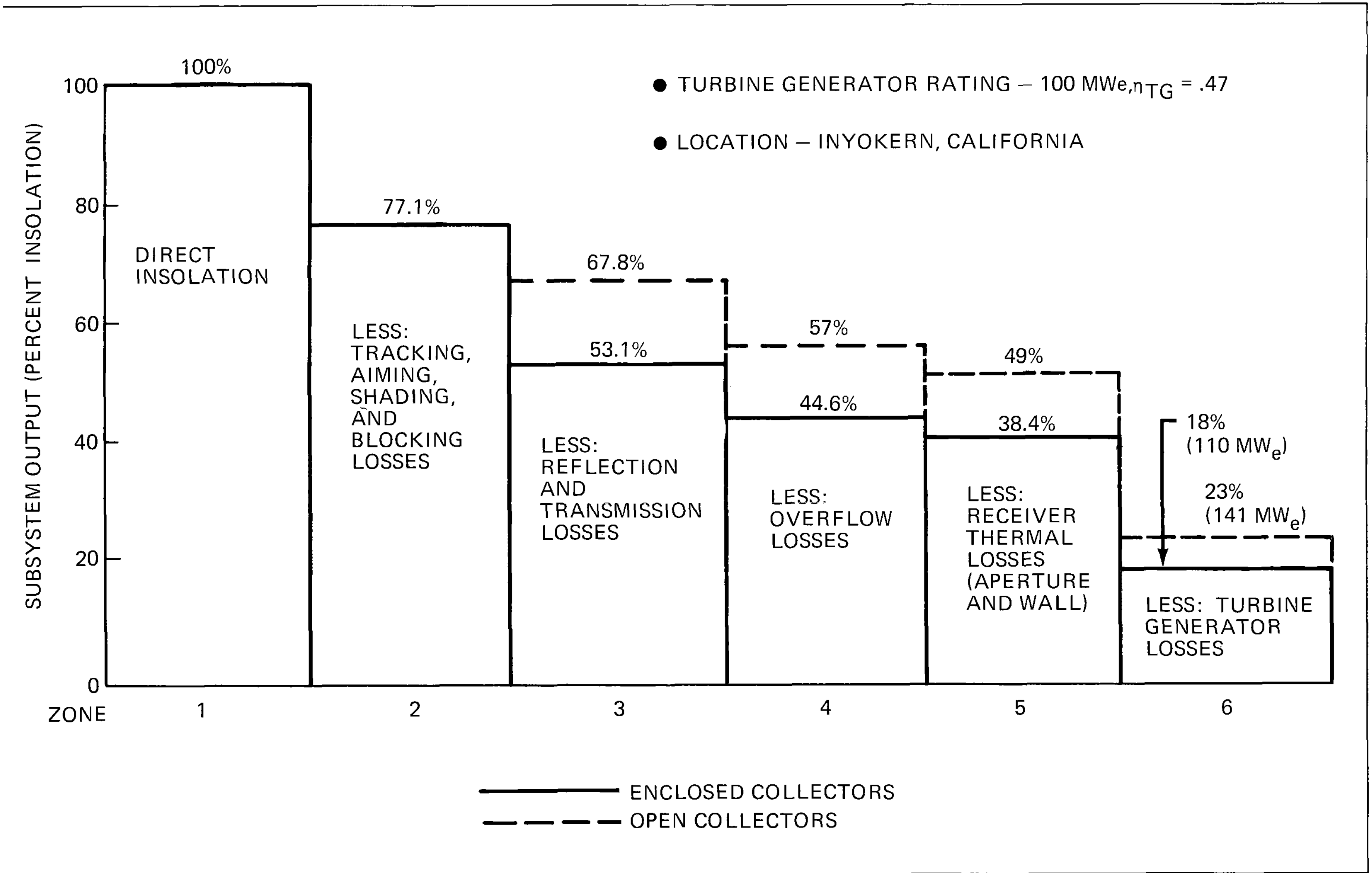
A hybrid system study requirement to produce 100 MW_e at the generator was fulfilled with the selected receiver concept and an arbitrary collector field. Performance is illustrated in Figure 3 for a summer-oriented field and mean summer insolation at noon. The solid horizontal lines across the chart illustrate subsystem efficiency losses for enclosed metallized-plastic reflectors (Zones 1-4), the receiver concept (Zone 5), and the helium cycle (Zone 6). Generator output exceeds 100 MW_e. Transmission losses in Zone 3 are the result of the dual passage of solar energy through the enclosures used in the interim study. Overflow losses in Zone 4 are due to far field reflectors whose reflected image is larger than the aperture. The helium cycle efficiency (Zone 6) is 0.47.

The dotted lines on Chart 3 indicates system performance for unenclosed reflectors or plane glass mirrors. Deletion of the transmission loss of Zone 4 has a large effect, raising the power production level to 141 MW_e and raising the overall efficiency to 23%. The selected receiver concept size is incapable of handling 141 MW_e but the field could be reduced considerably ($\sim$1/3) to produce over 100 MW_e with the receiver as designed at a 23% system efficiency. In fact, the reduced size field of uncovered reflectors matches almost exactly the field prescribed by EPRI for the study continuation.

Technology Requirements

All the required technology is available for the high temperature central receiver. Technology problems of the collector field will be resolved by other contracts. The primary problem is the availability of helium turbomachinery of 50 – 100 MW_e capacity. Resolution can come from a development program which 1) scales up from an existing 50 MW_e helium turbine, such as is operating in Oberhausen, Germany; or 2) scales down from potential helium turbomachinery for nuclear plants; or 3) utilizes the existing 50 MW_e turbine in a reduced size (intermediate) plant module. The latter option is the best possibility when energy storage is added, and receiver thermal energy must be partitioned between immediate and deferred production. Storage is to be included in the study continuation. Other turbomachinery areas requiring technology effort concern installation options, operation, and control.

Figure 3 Hybrid Plant System Performance



Receiver Performance

Predicted receiver performance is summarized in Figure 4. The 231 megawatts removed by the helium give the receiver an 85% efficiency. The largest loss is by reradiation out the aperture. This is a direct function of the high operational temperature of the receiver interior. Any high temperature receiver must face this loss and the associated heating effect on nearby structures. Severity and protective measures are to be examined during the remaining contract period. The reflective loss is due to reflection of energy within the cavity, a portion of which is never absorbed before it escapes through the aperture. Convection and conduction losses are small. The 231 megawatts absorbed by the helium exceeds the design minimum required to enter the turbine (211) to generate 100 megawatts of electrical power (47% cycle efficiency).

Receiver Cost

All the receiver cost drivers have been isolated for the concept selected. These include all structural materials, construction, and rigging. The first two items biased the preliminary compilation of costs to higher values than expected. These will be examined in depth in the study continuation.

Receiver Verification

An initial design of a 1 MW_{th} model receiver has been completed which simulates the 100 MW_e receiver concept. A preliminary processing schedule has been identified for model tests in the 1977-1978 time period. Model size was limited in the interim study by the capabilities of the Centre de la Rechérche Scientifique (CNRS) Solar Energy Laboratory at Odeillo, France.

Figure 4 Receiver Performance

		MEGAWATTS
SOLAR INPUT TO RECEIVER CAVITY		273.0
RECEIVER LOSSES:		
REFLECTION OUT APERTURE	10.3	
RERADIATION OUT APERTURE	23.4	
CONVECTION TO AIR	2.3	
CONDUCTION THROUGH WALLS	6.1	
TOTAL LOSSES		42.1
HEAT REMOVED BY HELIUM		230.9

RECEIVER EFFICIENCY (231/273) = 85%

4. Receiver Characteristics

Configuration Selection

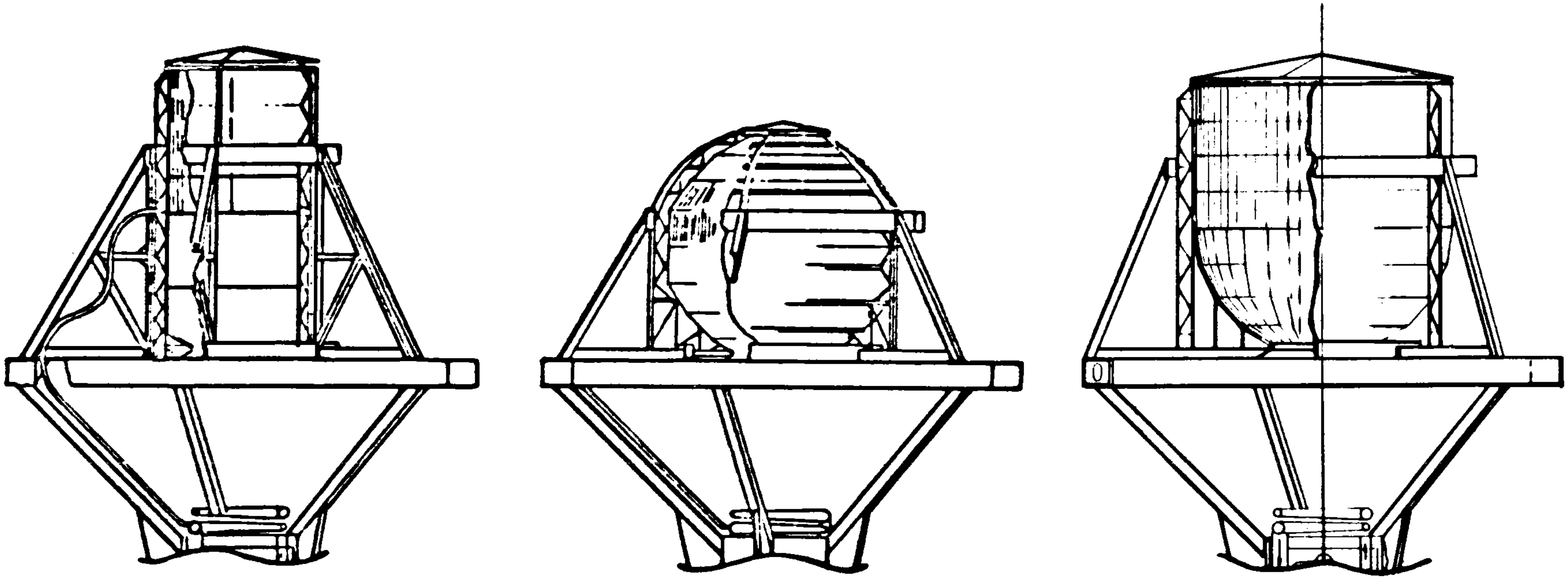
Early configuration work indicated that the most desirable receiver for the high temperature concept was one supported above the tower, and with the receiver having a bottom circular aperture to admit reflected energy from the collector field. Figure 5 presents the three bottom-aperture receiver shapes detailed in the interim study. The composite shape finally selected (extreme right) combines portions of the cylindrical and spherical shapes used in the study as the initial baseline and alternative.

The distribution of reflected and absorbed energy was an important factor in concept selection. Detailed thermal analyses were performed on the initial baseline 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 those of the sphere. A careful ray-tracing analysis also indicated reflection losses from the cylinder to be 13%, compared to under 4% for the sphere. The size of the cylinder to accommodate this difference was proportionately larger. The selected shape has reflection losses comparable to the spherical shape, due to similar incidence of incoming sunlight on the lower hemisphere. 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 the greater loss.

The upper portion of the preferred receiver is cylindrical for two important reasons. First, 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. Secondly, the ability to build and hang standard size heat exchanger panels offered technical and economic advantages with a cylindrical upper section. Structural assembly of a cylinder is much easier than a sphere, and the choice selected eliminated more than half of the complex curvature of a spherical shape.

Receiver dimensions are approximately 39 meters (128 feet) high and 39 meters (128 feet) in diameter. Aperture diameter is 16 meters (52.5 feet).

Figure 5 Alternative Configurations



CYLINDRICAL

SPHERICAL

COMPOSITE
(FINAL SHAPE)

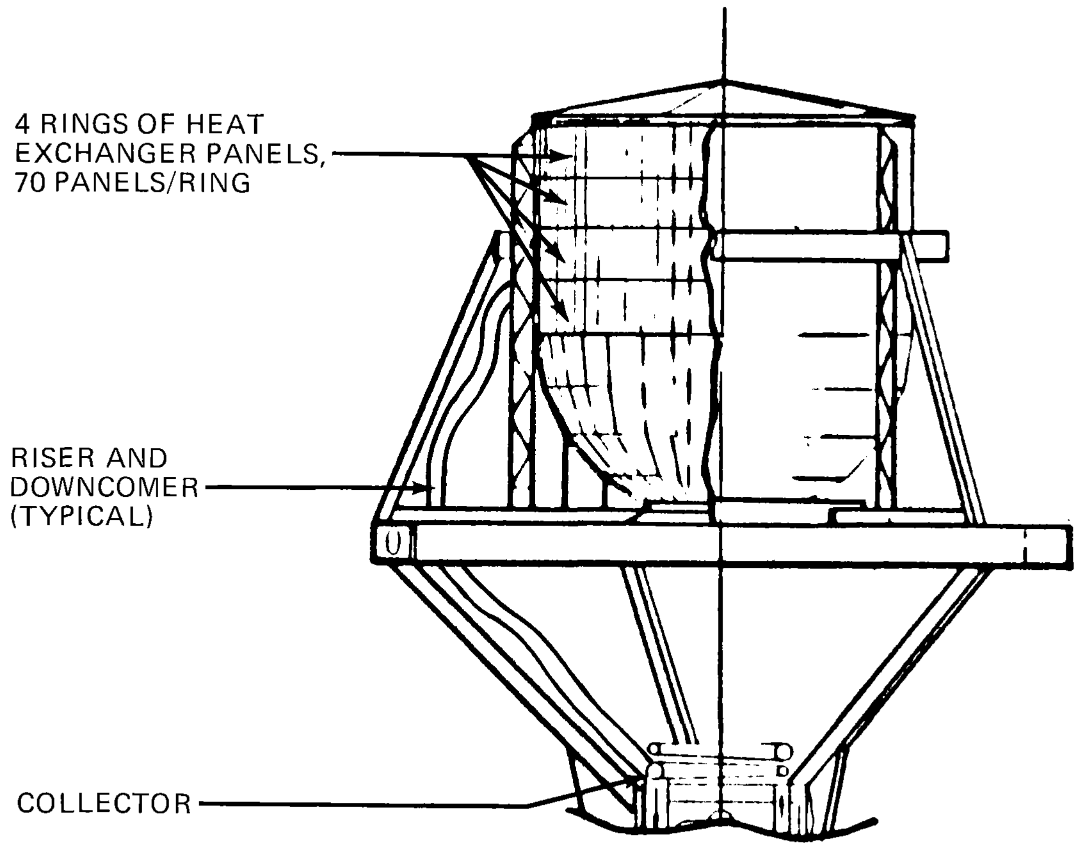
Structural Arrangement

The structural arrangement of the preferred receiver is shown in Figure 6. The cutaway view of the upper cylindrical section illustrates the mounting arrangement of the standard heat exchanger panels into 4 rows of 70 panels each. The lower hemispherical section closing to the bottom aperture has panels of insulation.

The receiver is mounted sufficiently above the tower to allow the field energy to enter the aperture. Five support legs extend up and out from the tower top to the main support ring. These supports are located away from the aperture to minimize heating by direct radiation from the field and to reduce field blockage. Vertical members extend in and up to the receiver stringer support ring. The vertical stringers support the roof section, receiver heat exchanger panels, and panel manifolds. Helium risers and downcomers are supported and guided from the main verticals down to the main collectors at the tower top. Each riser and downcomer set is partially shielded from the field radiation by the vertical supports.

A thorough analysis will be made of the field radiant flux heating during the study continuation. This will include the supports, risers, downcomers, the main support ring, and the receiver exterior. Areas where resultant temperatures are excessive will be protected by insulation or shields. Preliminary indications are that the support temperatures, in general, exceed the capability of mild steel. Coated stand-off shields show promise of keeping support temperatures within allowable limits. Another location of heating concern is near the aperture lip where the field radiation overflow will be most concentrated.

Figure 6 Structural Arrangement



Interior Arrangement

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

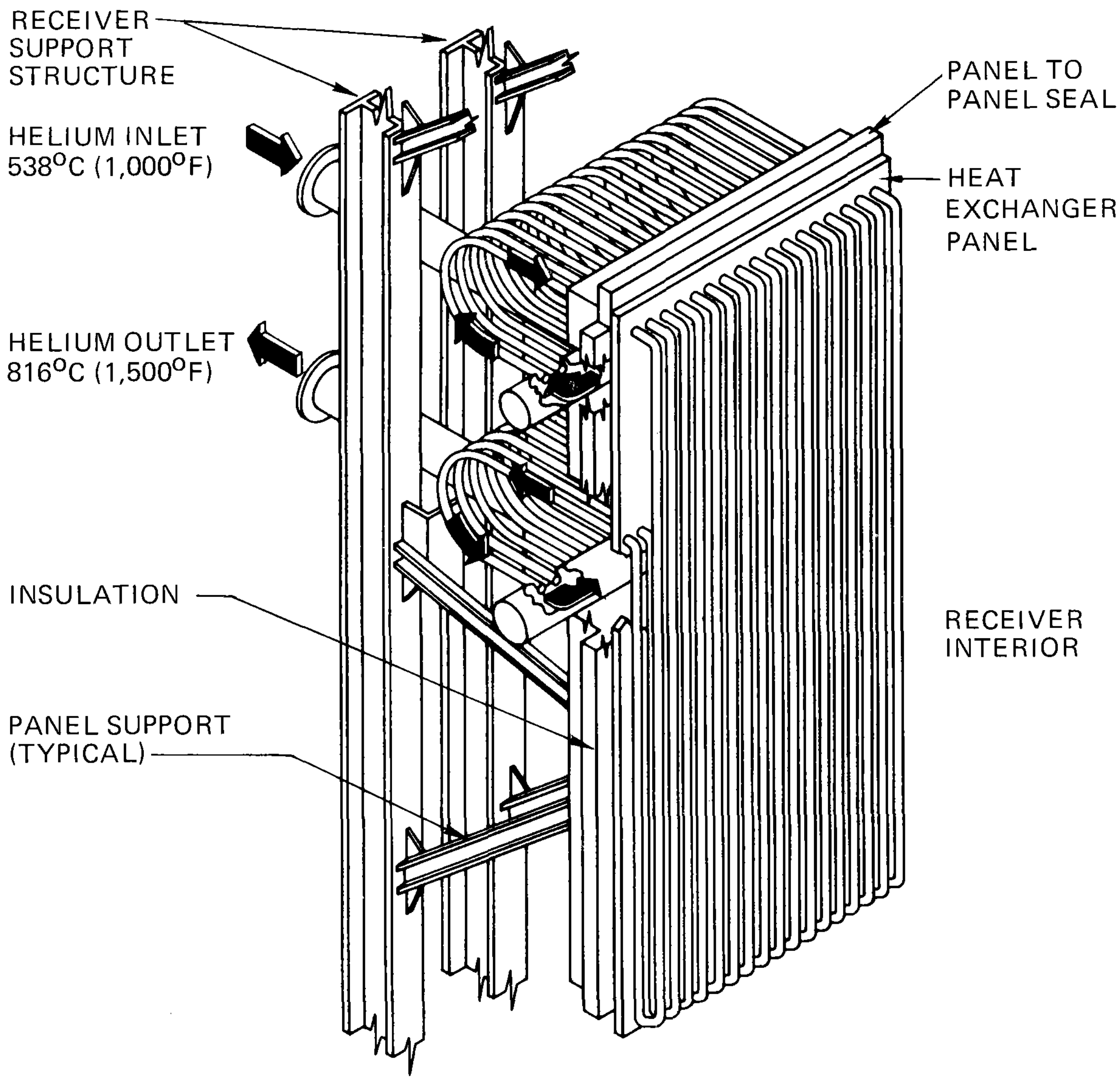
Figure 7 illustrates a standard heat exchanger panel and its structural supports. Progressing outward from the cavity, each panel consists of two offset columns of heat exchanger tubing, insulation, tubing loops to the helium manifolds, and the support structure to the outside wall. Each panel is designed to be removable as a unit, should replacement be necessary.

There are 20 tubes on a standard panel. Each tube makes two passes; one from the inlet down to the bend, and a second pass back close to the insulation up to the outlet. The difference in path length and the loops behind the insulation (shown exaggerated) are to provide proper tube expansion during thermal cycling to keep the interior tubes and tube pass-throughs in proper position. Material for the tubes was assumed to be Haynes 188 alloy because that material had been successfully thermal-cycled over a simulated 30-year lifetime. Tube dimensions are 2.84 centimeters (1.12 inch) outside diameter with a 0.32 centimeter (0.12 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 1820 kilograms (4000 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 concept and thickness.

The helium flow rate for each panel is established nominally at 0.58 kilograms/second (1.28 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 28°C (50°F) between the tube wall and the helium.

Figure 7 Interior Arrangement



Preferred Receiver Temperatures

A detailed temperature analysis was performed on the preferred receiver with heat exchanger panels mounted on the interior of the upper cylindrical section and insulation panels on the hemispherical section interior. Helium inlet and outlet temperatures of 538°C (1,000°F) and 816°C (1,500°F), respectively were assumed at each of the 4 rows of heat exchanger panels.

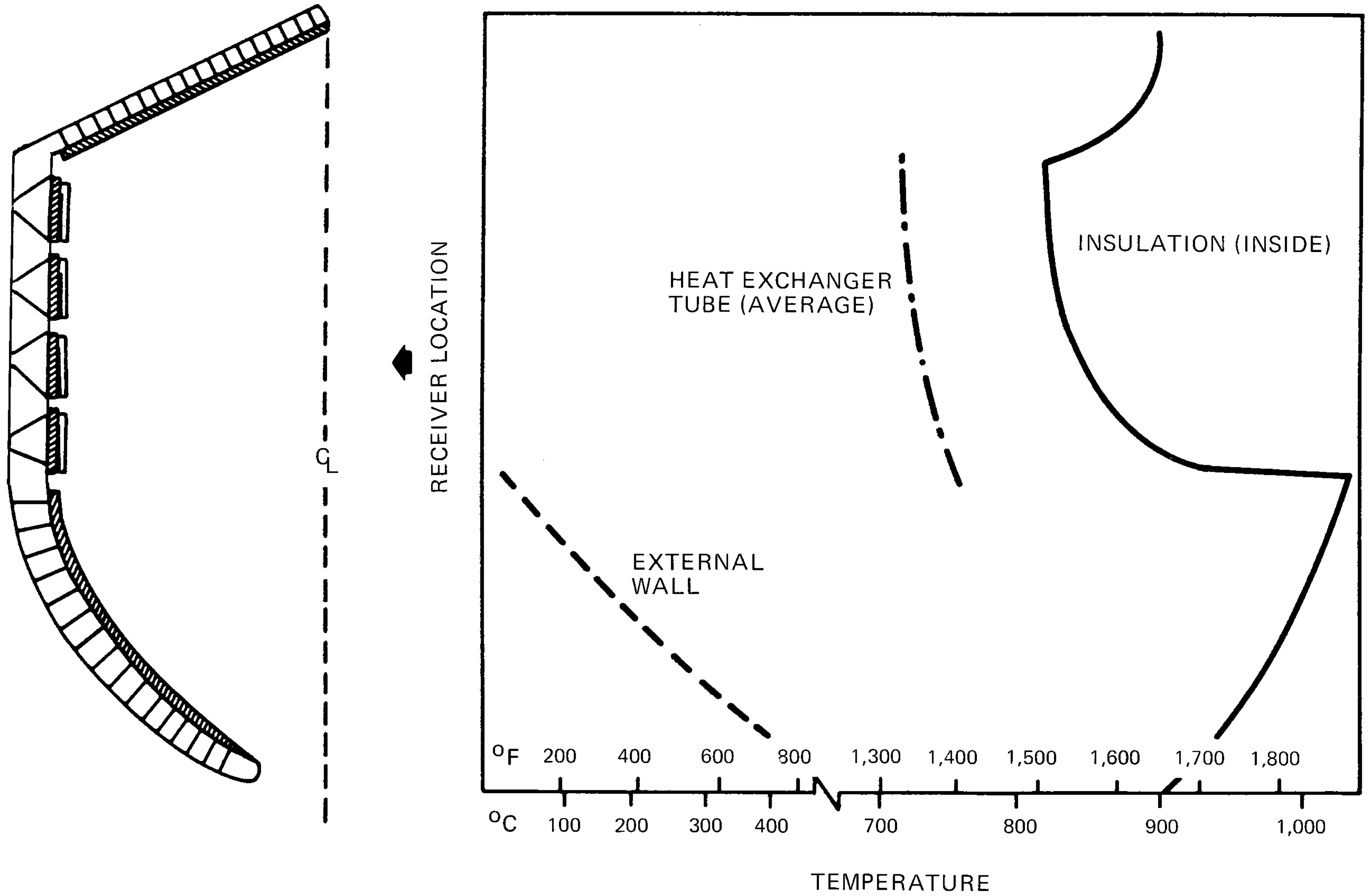
The results are shown in Figure 8 where temperatures of major components are plotted against location as shown in the receiver profile on the left-hand side. The inside temperatures on the insulated hemispherical wall rise steadily with increasing height, and then drop sharply back to a mean value of 870°C (1,600°F) on the insulation face of the heat exchanger panels.

A rise in temperature is noted across the top inner surface. All these temperatures are within the capability of available insulation materials.

The average heat exchanger tube temperature varies from about 760°C (1,400°F) at the lowest panel to 725°C (1,340°F) for the topmost panel. The upper two panel rows are essentially at a constant temperature, indicating that a constant mass flow rate of helium can be utilized. The temperatures at the lower two panel rows will require a higher helium mass flow to keep tube temperatures within reasonable limits. Maximum pressure loss through the tubes is 1-2% of system pressure.

The external receiver wall temperature starts high near the aperture and then drops rapidly away from that zone. The study continuation will examine the near-aperture heating problem in more detail.

Figure 8 Preferred Receiver Temperatures



Panel Insulation

Insulation panels entirely line the interior of the preferred receiver. The left side of Figure 9 shows wall heat losses for various inside face temperatures and insulation thicknesses. The highest conduction heat loss comes from the receiver hemispherical section whose bare insulation panels have an average hot-face temperature of about 980°C (1,800°F) when driven by the concentrated radiant flux (see Figure 8 for reference). The design point indicated in Figure 9 of 0.15 meter (6 inch) insulation thickness was a compromise choice to keep both wall conduction losses and insulation weights reasonably small. The heat loss for the chosen thickness is about 950 watts/meter^2 (300 BTU/hour-foot^2). The insulation concept selected consisted of the three successive layers of material arranged and dimensioned as shown by the drawing on the right-hand side of Figure 9. The same insulation concept is used on the heat exchanger panels behind the tubes. The face temperature for these panels is below 870°C (1,600°F). Total conduction loss for the receiver is about 6 megawatts, or slightly over 2% of the solar input into the receiver.

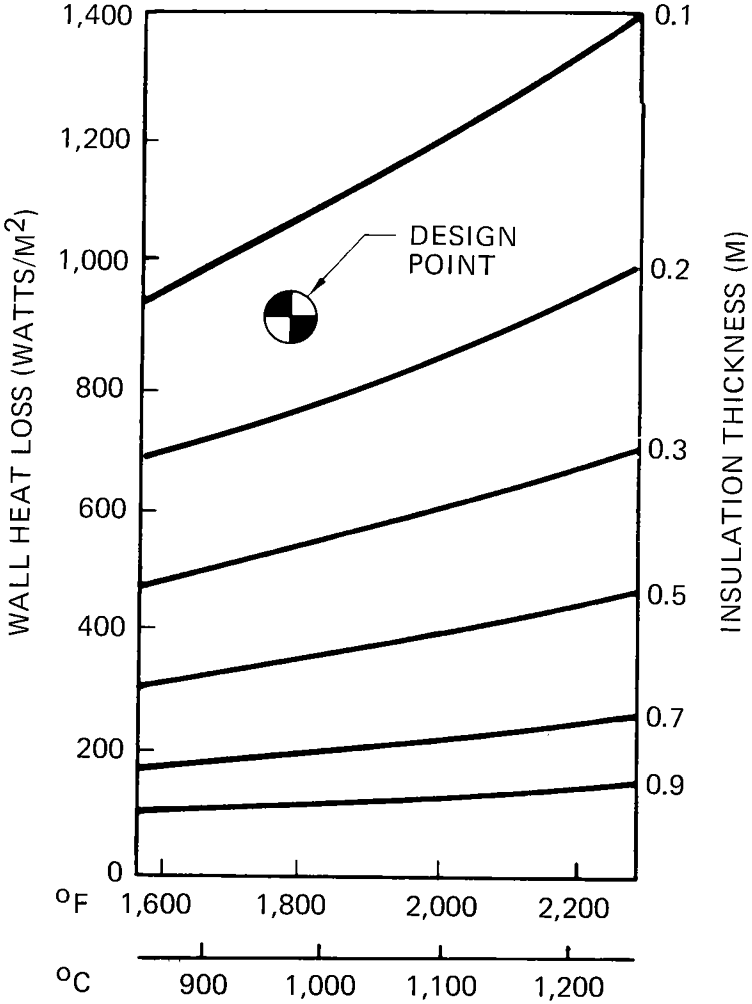
Structural Materials

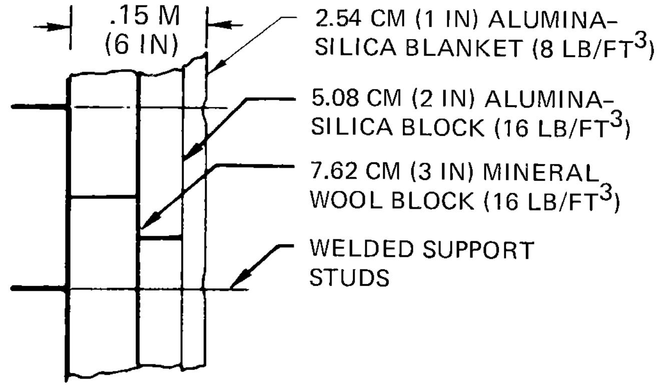
Materials specialists selected Haynes 188 and Inconel 617 as the primary material candidates for this high-temperature application. The Haynes 188 material was available for immediate testing so structural design for the interim study period was based on its use in the heat exchanger tubing, manifolds, and high-temperature piping runs to the turbine.

The initial material concerns were for the large number of thermal cycles to be experienced in 30 years of operational life and the large thermal swing of each daily cycle. To accommodate these factors, an initial concept had included receiver aperture doors and a "keep warm" circuit to retain receiver cavity heat. However, after examining in-depth property data for Haynes 188, it became evident that thermal cycling stress over the number of cycles was not the only problem. The time at high temperature (particularly temperatures in excess of 760°C or 1,400°F) was determined as a potential design constraint due to material creep properties. Accordingly, the provisions for aperture doors and "keep warm" circuitry were not only unnecessary, but actually undesirable. The deletions simplify receiver design.

Material selection for the receiver exterior and its external supports will await the results of further temperature analyses in the on-going study.

Figure 9 Insulation Concept





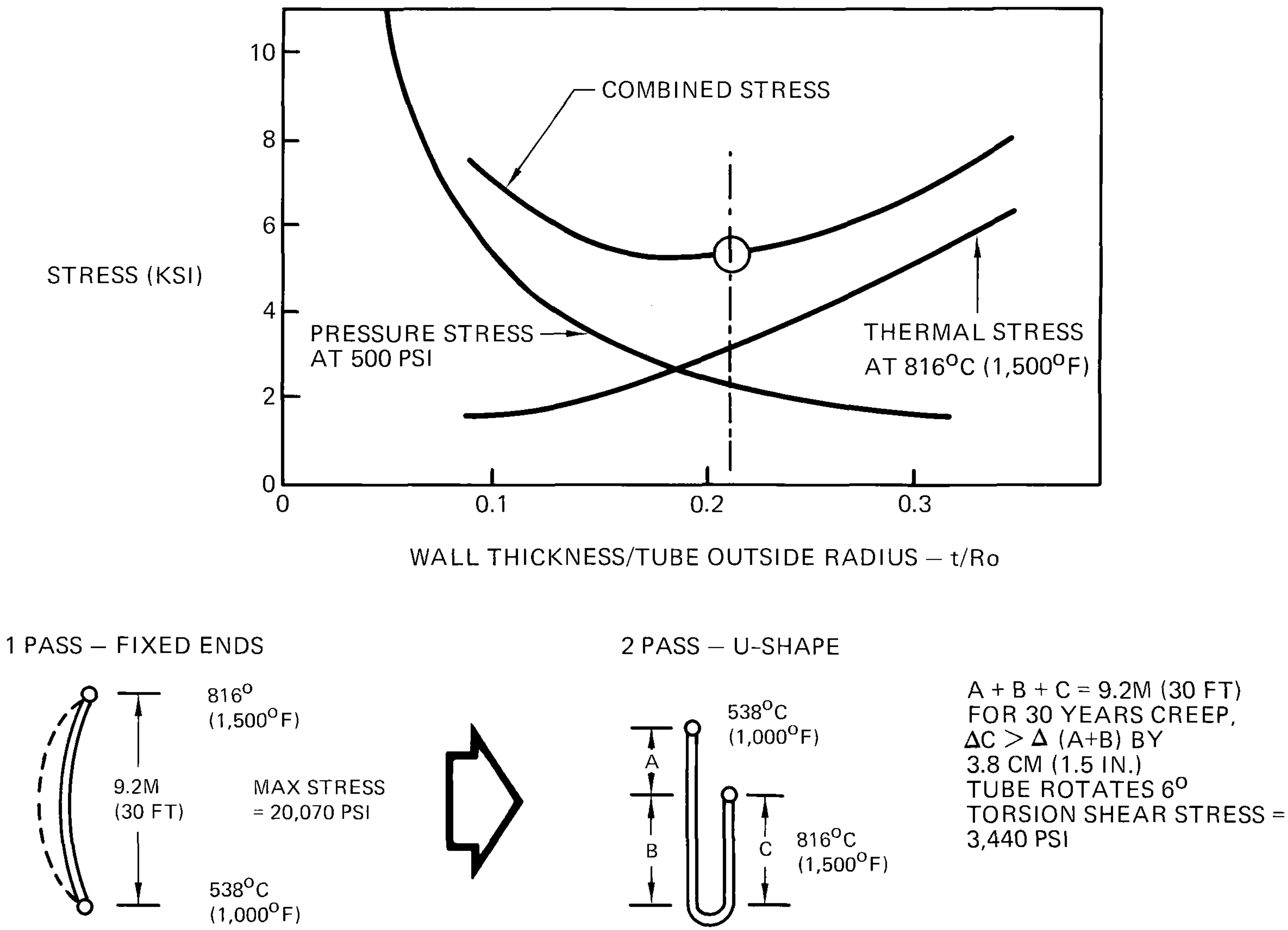
FACE TEMPERATURE	980°C (1,800°F)
BACK TEMPERATURE	72°C (162°F)
HEAT LOSS	950 WATTS/M^2)
WEIGHT	47.5 KG/M^2 (9.75 LB/FT^2)

Heat Exchanger Tube Geometry

A nominal tube radius of 1.27 centimeters (0.5 inch) was determined from thermal and fluid analyses which considered internal turbulent flow heat transfer and pressure drop. The spacing between tubes and the number of tubes on a standard heat exchanger panel were determined by required area for heat transfer and the incident flux on the panel. The remaining considerations were tube wall thickness and the configuration of the tube on the heat exchanger panel. Figure 10 displays these two characteristics. Thermal and pressure stresses, combined on the upper graph, show that the tubes will experience minimum stress levels at wall thickness-to-tube outside radius ratios (t/Ro) between 0.12 and 0.21. These levels are less than the maximum allowable for 30 years life at 816^{o}C (1,500^{o}F). The conservative value of t/Ro = .21 was chosen because the potential of tubes for internal scaling over 30 years lifetime was uncertain prior to testing the Haynes 188. Therefore, tube outside diameter was set at 2.84 centimeters (1.12 inches) with a 0.32 centimeter (.12 inch) wall thickness.

A U-shaped tube with two passes was selected as the basic configuration. The diagrams at the bottom of Figure 10 show two tube configurations. With ends fixed on a straight one-pass tube of the length desired, an intolerable maximum stress of over 20,000 psi would be obtained with a resultant large deflection. The U-shaped tube is designed to compensate the longer cooler leg (A+B) at its thermal expansion coefficient, with that of the shorter leg (C) at a higher thermal expansion coefficient. Considering thermal expansion and creep over 30 years lifetime, the change in the shorter leg C is only 3.8 centimeters (1.5 inches) more than the change in the longer leg. The result is a rotation of the tube (in a tube guide) of 6^{o}. The torsion shear stress is a tolerable 3,440 psi. The tubes are also looped after they pass through the insulation (see Figure 7 for reference) to allow the configuration to be stable through the insulation.

Figure 10 Heat Exchanger Tube Geometry



5. Material Selection and Test

Material Selection

Metal alloys were chosen as candidates for high temperature tubing applications based upon manufacturing, performance capability, and economic considerations. The high-side temperature limit was established at about 816°C (1,500°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 Test

A test specimen of Haynes 188 was fabricated and subjected to 10,560 thermal cycles under pressure to simulate a 30-year lifetime. The arrangement and dimensions of the specimen are shown on the left-hand side of Figure 11. The specimen was pressurized to 34 bar (500 psi) helium pressure and tube temperatures cycled between 482°C (900°F) and 830°C (1,525°F). The top figure on the right-hand side of Figure 11 shows test conditions for each weekly run. The bottom figure shows a schematic of the test setup featuring quartz lamp heaters and a regulated helium bottle supply.

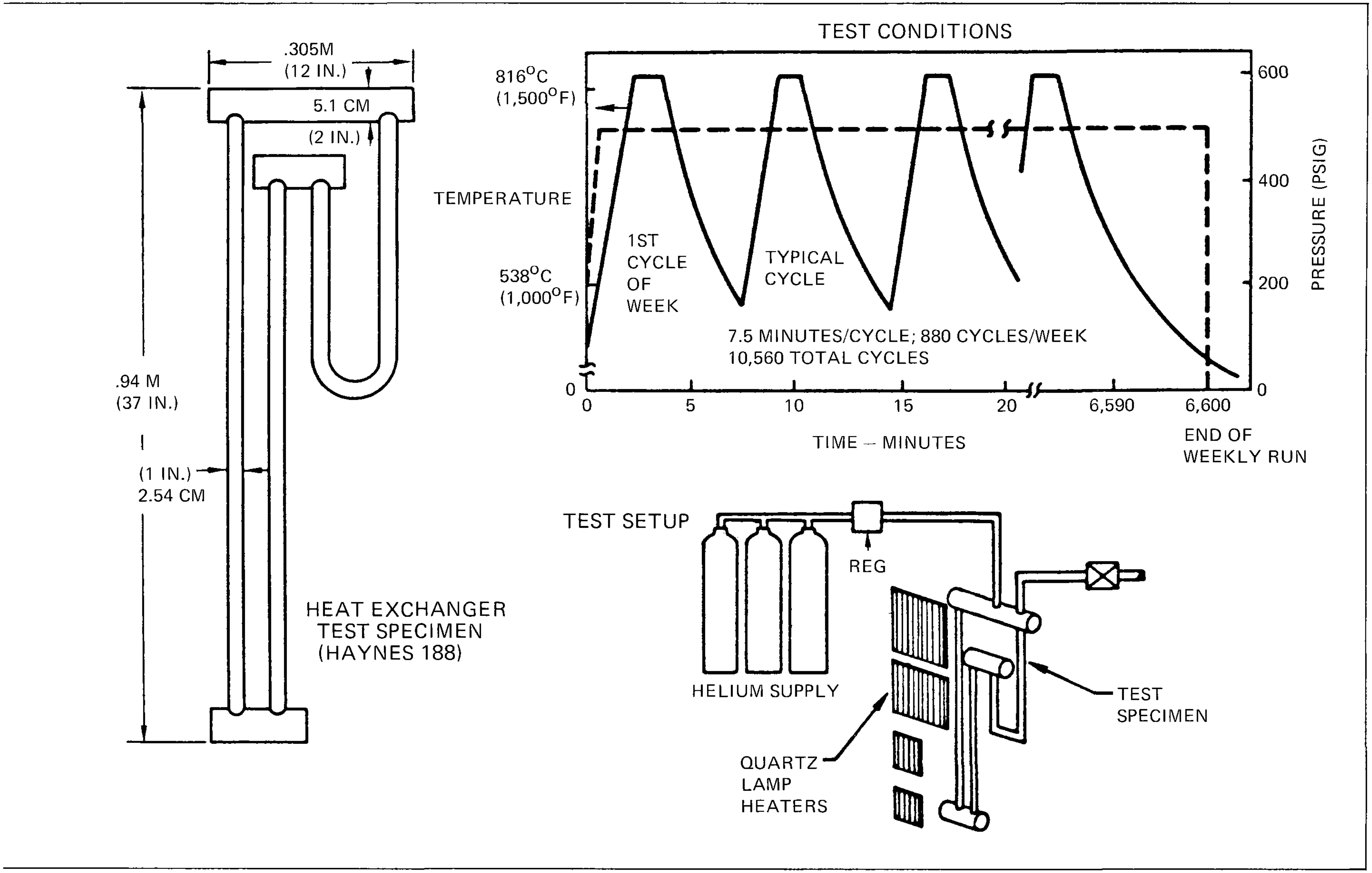
Test Results

Physical, mechanical, and metallurgical evaluations after the test gave the following results. External and internal surfaces of the Haynes 188 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 stress 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 microstructure comparisons. A sharp notch at a header weld root revealed no crack propagation under microscopic examination. Sections of the scaled tube surfaces showed little, if any, intergranular oxide penetration.

Test Summary

The thermal cycling test verified Haynes 188 as an excellent material for central receiver high temperature applications. Results of similar testing of Inconel 617 during the study continuation should provide sufficient data to make a final material selection based on performance and economics.

Figure 11 Thermal Cycling Test



6. System Implementation

The interim study not only considered the high temperature receiver, but included a conceptual analysis of a commercial size solar plant to implement the receiver and produce 100 megawatts of electrical power at the plant generator. This section highlights the system concept and those subsystems which, with the receiver subsystem, comprise a total power plant.

System Concept

The solar plant concept consists of a tower supporting a central receiver in the midst of a heliostat collector field, whose individual collectors track the sun in a manner to reflect solar energy continuously into the receiver. The focused radiant energy is converted to heat and absorbed by the transport fluid (in this case, helium) which transports the heat to the turbine-generator set within the tower for the production of electrical energy.

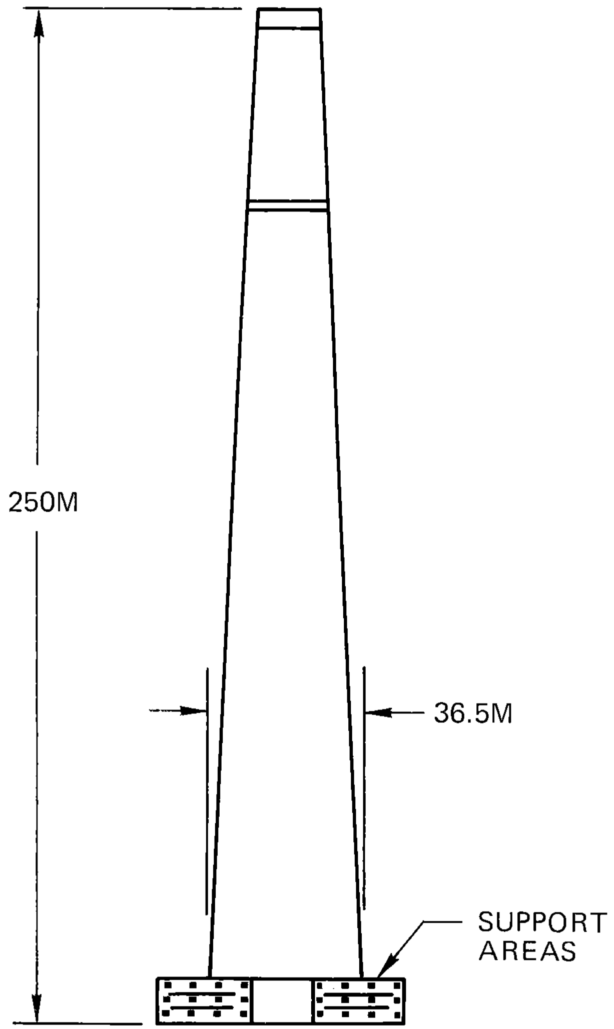
Collector Subsystem

The interim study initially used enclosed plastic mirrors as the collector field. For 100 megawatts of electrical power, this field required 32,000 collectors in a roughly circular field. Removing the enclosures reduces the number of collectors to 20,000 and field size to 0.50 square kilometers. Field width is constrained by practical tower height and the rapidly diminishing efficiency of mirrors at the outer rim of the field. For the study continuation, EPRI has provided Boeing with a plant model containing tower height, field size, and field performance so direct comparisons can be made of other receivers (and cycles). The collector subsystem is the subject of current ERDA studies.

Tower Subsystem

The height to the receiver aperture plane of 290 meters (951 feet), and the required space offset, resulted in a conceptual tower design of the size and characteristics shown in Figure 12. The tower top was made large enough to accommodate the turbine near the top if this concept were feasible.

Figure 12 Tower Subsystem



FEATURES

SLIP CAST REINFORCED CONCRETE CONSTRUCTION

CYLINDRICAL TAPERED SHELL (4–5% TAPER)

DESIGNED FOR SEISMIC RISK ZONE 3 (MAXIMUM)

DESIGNED FOR MAXIMUM WIND GUSTS OF 120 MPH

DESIGNED FOR 2,250,000 KG (5,000,000 LB) LOAD

CONTAINS APPROXIMATELY 35,000 CUBIC YARDS OF CONCRETE

SUPPORTS CENTRAL RECEIVER AND ASSOCIATED STRUCTURE

INTERNAL CONTAINMENT OF:

- TURBINE–GENERATOR
- COMPRESSOR(S)
- RECUPERATOR
- ENERGY STORAGE EQUIPMENT (AS REQUIRED)
- EQUIPMENT ROOM(S)
- ELEVATOR AND STAIRS

Thermal Engine Subsystem

The thermal engine cycle baselined for the study is a closed-loop recuperative Brayton cycle, using helium as a working fluid. The thermal engine subsystem consists of a turbine-generator design concept similar to that of an actual 50 MW_e helium turbine-generator in checkout at a public utility in Oberhausen, Germany. A picture of this installation is shown on the left-hand side of Figure 13. Visible in the center are the high pressure compressor, and high pressure turbine contained in a single unit. The high pressure turbine drives the low pressure turbine in the background, which drives the generator (located beyond). The laterals to the central unit are from the cylindrical intercoolers. In the foreground is the low pressure compressor being supplied low temperature helium from cylindrical precoolers (not shown). A recuperator is located below the installation.

The operational schematic of the helium cycle for the Oberhausen installation is sketched on the right-hand side of Figure 13. For the solar plant concept, the receiver replaces the heater as the heat supply. The high pressure side from the high pressure compressor would operate at 34 bar (500 psi). The recuperator is used to provide additional heat to the helium stream to reach the 538°C (1,000°F) receiver inlet temperature, this heat being supplied from the low pressure helium coming from the turbine. Helium leaves the receiver and enters the turbine at 816°C (1,500°F), where it is expanded and drives the generator.

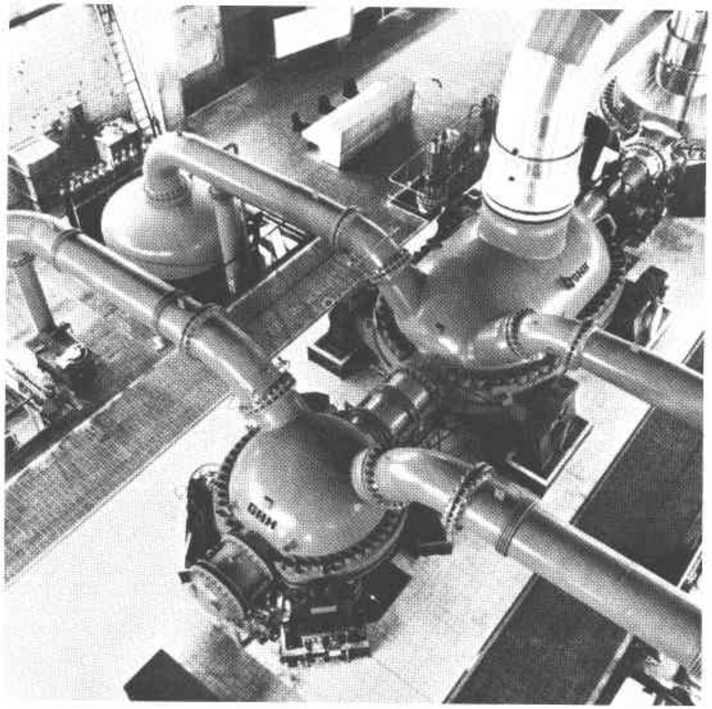
Turbine installation will be in the tower. The size is such that it could be mounted vertically at the tower top, shortening helium runs (and decreasing heat losses) to the receiver. This advantage will be weighed against the dynamics problems associated with vertical installation, and longer runs to the storage equipment. The most likely installation will be horizontal and at a location lower in the tower.

The 816°C (1,500°F) turbine inlet temperature of helium is consistent with turbine blade material capability, and provides the high cycle efficiency of 0.47, considerably above that of more conventional turbines.

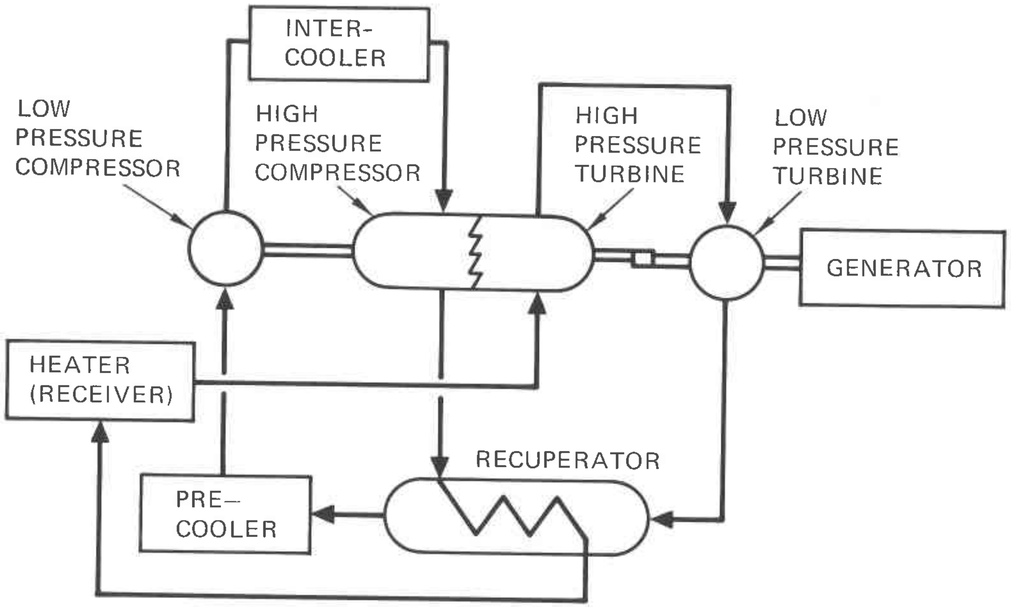
Energy Storage Subsystem

This subsystem is indicated here to complete the system definition. Energy storage was not considered in the interim study, but is included in the study continuation.

Figure 13 Thermal Engine Subsystem



HELIUM TURBINE INSTALLATION
AT OBERHAUSEN, GERMANY



HELIUM CYCLE SCHEMATIC

7. Model Receiver

1 MW_{th} Model Receiver Design

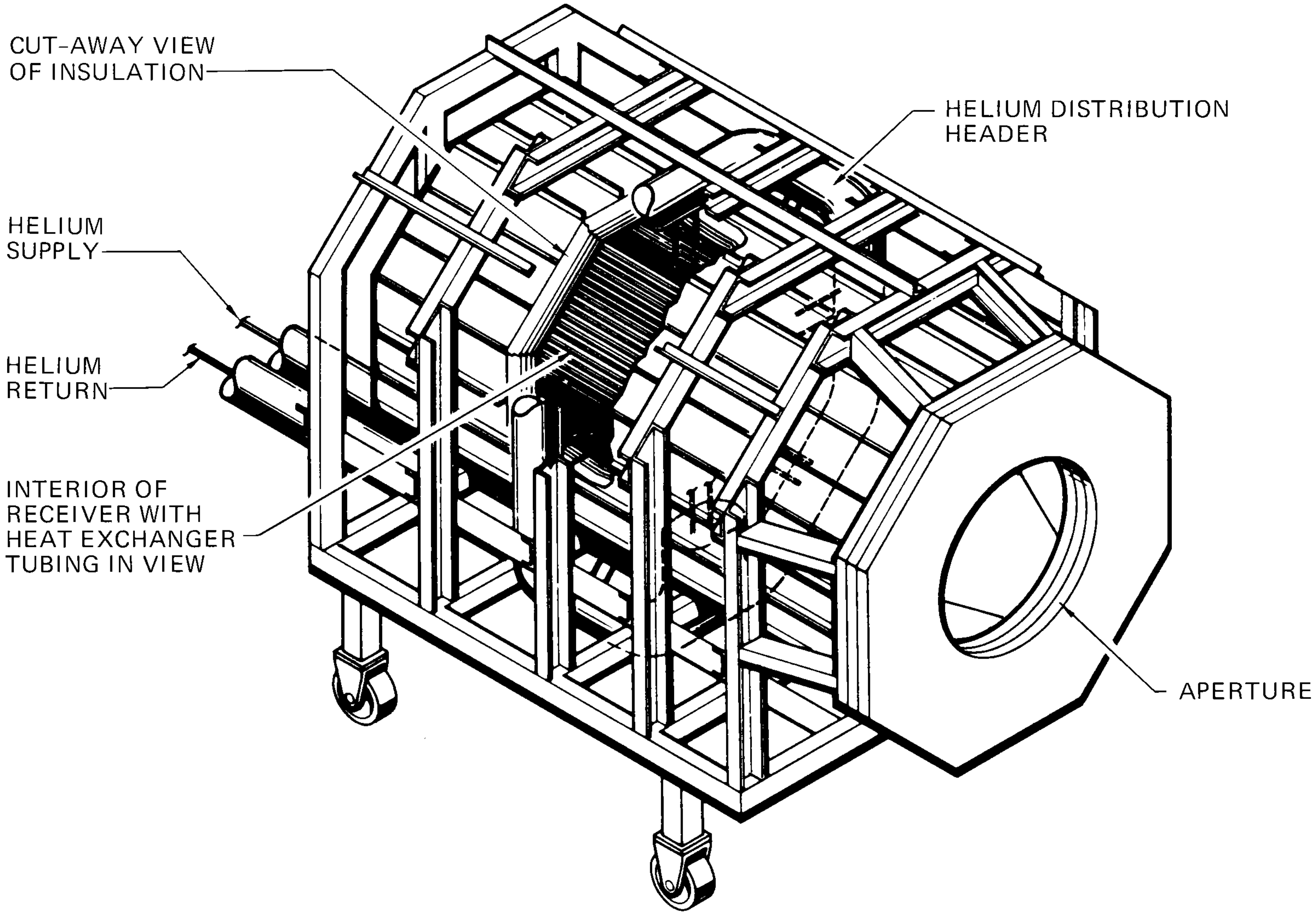
Definition of the 100 MW_e receiver concept provided the performance and configuration guidelines for preliminary design of a test model receiver. The other primary guideline was the consideration of the Centre de la Recherche Scientifique (CNRS) Solar Energy Laboratory at Odeillo, France, as the test facility. The CNRS reflector field provides one megawatt of radiant energy at its focal zone (model aperture). The model receiver was scaled to preserve temperature and heat flux distributions of the 100 MW_e receiver interior within the size constraints of the three meters (ten feet) cubed working space in the CNRS focal building. The resultant 1 MW_{th} model (Figure 14) approximates the structural arrangement of the 100 MW_e receiver by an octagonal shape. Dimensions of the design are 2.74 meters (9 feet) high, 2.05 meters (6.67 feet) diameter, and 3 meters (9.86 feet) long. Aperture diameter is 0.91 meters (2.98 feet). Weight is approximately 1,000 kilograms (2,200 pounds).

Eight heat exchanger panels are positioned around the inner periphery, away from the aperture end. Each panel removes the same incident heat flux per unit area as the full-scale concept. Thirty (30) tubes on each panel, with two passes per tube, preserve the radiant exchange and turbulent flow conditions as in the full-scale receiver. All panels and walls are backed by 0.15 meters (6 inches) of insulation similar to the 100 MW_e receiver. External inlet and outlet manifolds for the working fluid are positioned to facilitate connections at CNRS.

Model Receiver Development Plan

A development plan for the 1 MW_{th} test model has been formulated based upon testing at CNRS in the third quarter of CY 1978, with summary wrap-up early in CY 1979. The potential availability of a 5 MW_{th} ERDA test facility for concept testing in the United States in late 1977 would permit an accelerated development program. The model receiver shown would require only minor design modifications to adapt to the facility.

Figure 14 Receiver Layout 1 MW_{th} Model



8. Future Effort

At the conclusion of the interim study, EPRI defined reflector field characteristics, tower size, required storage capability, megawatt output, and associated costs for an individual solar plant module. These will be incorporated into the continuing study, such that the study will concentrate on the receiver, the helium heat transport subsystem, storage subsystem, turbomachinery, and the model receiver. Costs and performance of these elements will be added to (or replace) those defined by EPRI to permit an overall system comparative evaluation with other central receiver concepts.

Materials testing will continue with Inconel 617 to be thermally cycled. Receiver operational aspects for year-round performance will be examined in the context of total plant operation. Receiver thermal analysis will be extended to defining temperatures of support structures. Where problems are indicated, methods of alleviation will be indicated. The preliminary cost estimates of the preferred receiver were greater than expected. A thorough review will be made of ways to reduce this cost by refined construction estimates, material substitution, design practice, and/or selective component replacement prior to the end of 30-year life.

Energy storage candidate concepts will be examined and a preferred concept selected for detailed plant operations and cost analyses. Bus bar energy costs will be developed for the helium turbine solar plant and compared to that of steam turbine solar plants.

Operational availability of helium turbines of the sizes required for the full-scale receiver and the model receiver should be pursued aggressively. If development lead times for new turbines or availability of existing turbines (such as at Oberhausen) prove too long, adaptation of existing air turbines should be examined as an alternative. The continuation study will address potential problems associated with the turbine location and orientation (horizontal or vertical) as well as the practical attainment of the turbine efficiency (0.47) used in the interim study.

A thorough review will be made of test facilities and methods to be used as alternatives to model receiver testing at the CNRS facility in France. One area of examination will cover the adaptability of radiant lamps (IR testing) for the model receiver or its panels. The other area to be reviewed is testing in the ERDA 5 MW_{th} facility to be located in Albuquerque, New Mexico. Figure 15 shows an accelerated schedule for testing in that facility in the third quarter of CY 1977.

Figure 15 Accelerated Model Receiver Development Plan