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PHASE I OF THE FIRST SMALL POWER SYSTEM EXPERIMENT (ENGINEERING EXPERIMENT NO. 1)

Final Technical Report
Volume II - System Concept Selection

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY

MCDONNELL DOUGLAS CORPORATION

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PHASE I OF THE FIRST SMALL POWER SYSTEM EXPERIMENT (ENGINEERING EXPERIMENT NO. 1)

Final Technical Report
Volume II – System Concept Selection

MAY 1979

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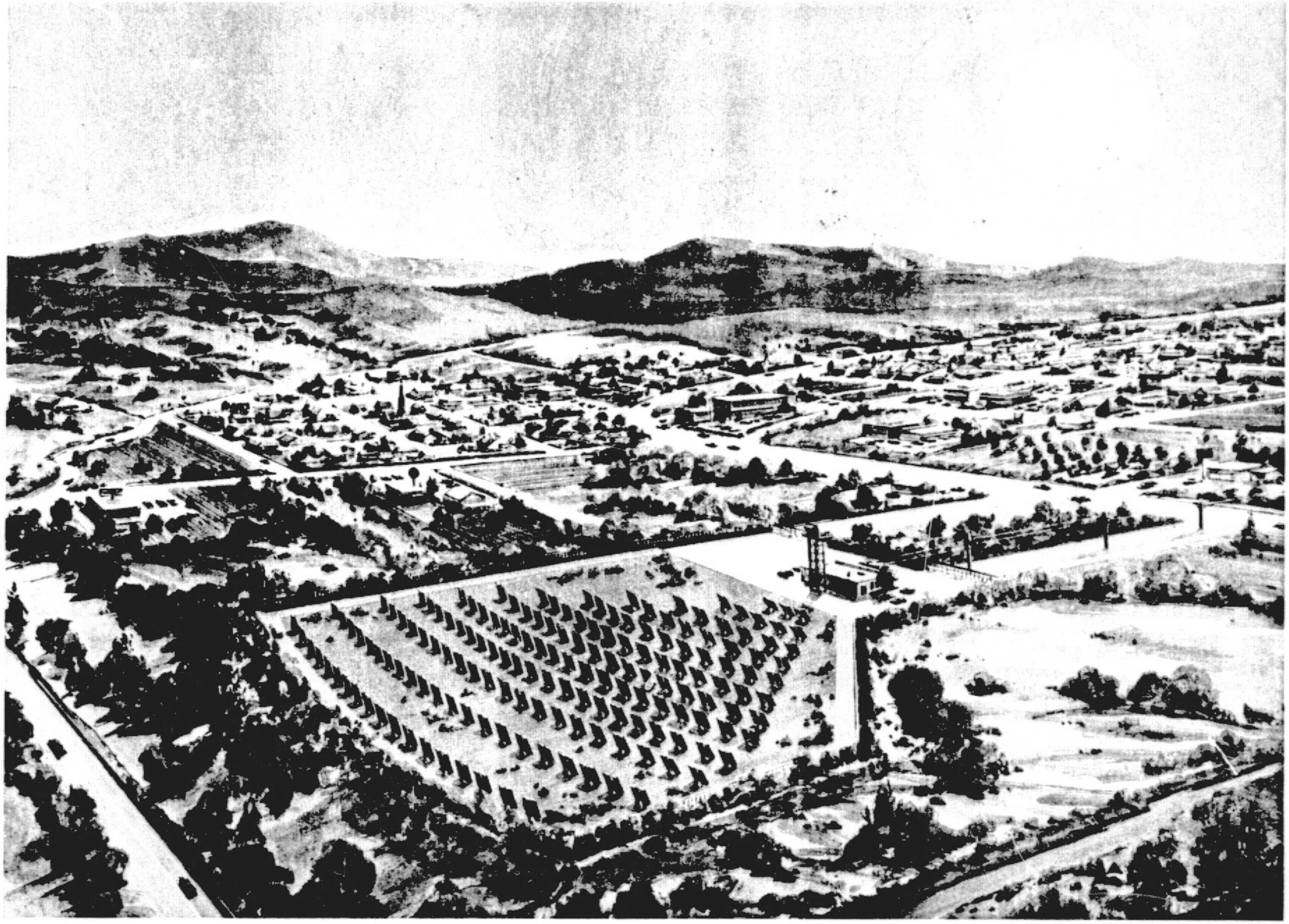

PREFACE

This document constitutes the McDonnell Douglas Astronautics Company (MDAC) final technical report for Phase I of the First Small Power System Experiment (Engineering Experiment No. 1). Phase I is an investigation of various system concepts that will allow the selection of the most appropriate system or systems for the first small solar power system application. This 10-month study is a part of the Small Power Systems Program that is being developed under the direction of the Department of Energy (DOE) and managed by the Jet Propulsion Laboratory (JPL). The final report is submitted to JPL under Contract No. 955117.

The final technical report consists of five volumes, as follows:

Volume I	Executive Summary
II	System Concept Selection
III	Experimental System Definitions (3.5, 4.5, and 6.5 Year Programs)
IV	Commercial System Definition
V	Supporting Analyses and Trade Studies

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CONTENTS

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Section 1
PHASE I PROGRAM INTRODUCTION

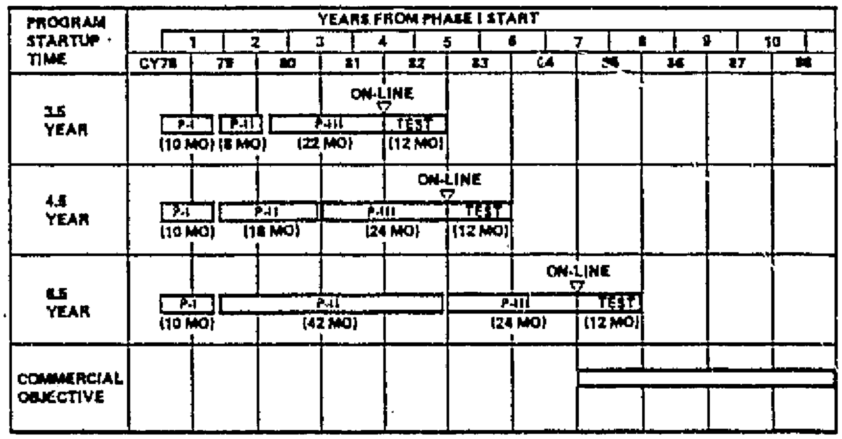
The Solar Thermal Power Systems Office of the Division of Solar Energy of DOE has initiated several application-oriented programs, one of which is the Small Power Systems Program. The overall objective of this program is to develop and foster the commercialization of modular solar thermal power systems for application in the 1 to 10 MWe range. Potential applications include power systems for remote utility applications, small communities, rural areas, and industrial users. Engineering Experiment No. 1 represents the first small power system to be developed under this program.

The primary goal of Engineering Experiment No. 1 (EE1) is to identify suitable technological approaches for small power systems applications and to design, fabricate, field install, test and evaluate a solar power facility based on an optimum use of near-term technologies. Investigation of the performance, functional, operational and institutional interface aspects of such a facility in a field test environment are additional objectives.

Engineering Experiment No. 1 will be conducted in three phases: Phase I - Concept Definition, Phase II - Design and Development Testing, and Phase III - Plant Construction and Testing. Three candidate programs for EE No. 1 are shown in Figure 1-1.

Phase I objectives were to investigate various system concepts and develop information which will allow selection of the most appropriate system for the first small power system application. System design and system optimization studies were conducted considering plant size, annual capacity factor, and startup time (the time from start of Phase I to the initiation of testing in Phase III) as variables. The primary output of Phase I was to be the definition of preferred system concepts for each startup time, design sensitivity and cost data for the systems studied, and Phase II Program Plans for each preferred system concept.

• THREE CANDIDATE PROGRAMS FOR EE NO. 1



• THREE PROJECT PHASES
  I CONCEPT DEFINITION
  II PRELIMINARY AND DETAILED DESIGN; COMPONENT/SUBSYSTEM DEVELOPMENT/TESTING
  III FABRICATION, INSTALLATION, TEST AND EVALUATION
• CATEGORY A CANDIDATE SYSTEMS - GENERAL, EXCLUDING DISH CONCENTRATORS

Figure 1-1. Overall Program Scope

Phase II involves the preliminary and detailed design of the preferred system, and component and/or subsystem development testing that are needed before proceeding with plant construction in Phase III. Phase II may be from 8 to 42 months depending on the program selected by JPL as a result of Phase I.

Phase III will consist of subsystem fabrication, plant construction, installation, testing, and evaluation of the solar power facility (Engineering Experiment No. 1). A 3-year schedule is anticipated for this phase, with testing conducted during the third year.

Late in the Phase I study period, DOE concluded that a better balance of the overall solar thermal electric program could be achieved by limiting the JPL Small Power Applications activities to point-focus distributed systems. Consequently, DOE directed that JPL take the necessary steps to constrain the JPL-managed First Engineering Experiment (EE No. 1) to point-focusing distributed receiver technology for all phases beyond Phase I. Accordingly, on 3 April 1979, all MDAC efforts on Phase II program planning were terminated by JPL directive.

1.1 STUDY TASK APPROACH

Phase I study objectives were: (1) select preferred system concepts for each of the three program durations, (2) complete conceptual designs for each of three system concepts, (3) provide sensitivity data over a range of; plant rating: 0.5-10 MWe; annual capacity factor: 0 storage to 0.7, (4) prepare detailed Phase II plans and cost proposal (3 versions of EE No. 1), (5) prepare Phase III program and cost estimates (3 versions of EE No. 1), and (6) recommend preferred EE No. 1 program. Three major tasks were planned for the 10-month Phase I effort. They were Task 1 - Development of Preferred System Concepts, Task 2 - Sensitivity Analyses, and Task 3 - Phase II Program Plans. The top-level study flow is indicated in Figure 1-2.

In Task I, three preferred concepts were defined to the conceptual design level. The concepts were consistent with the three specified program startup times of 3.5, 4.5, and 6.5 years. In Task I, power plants were considered

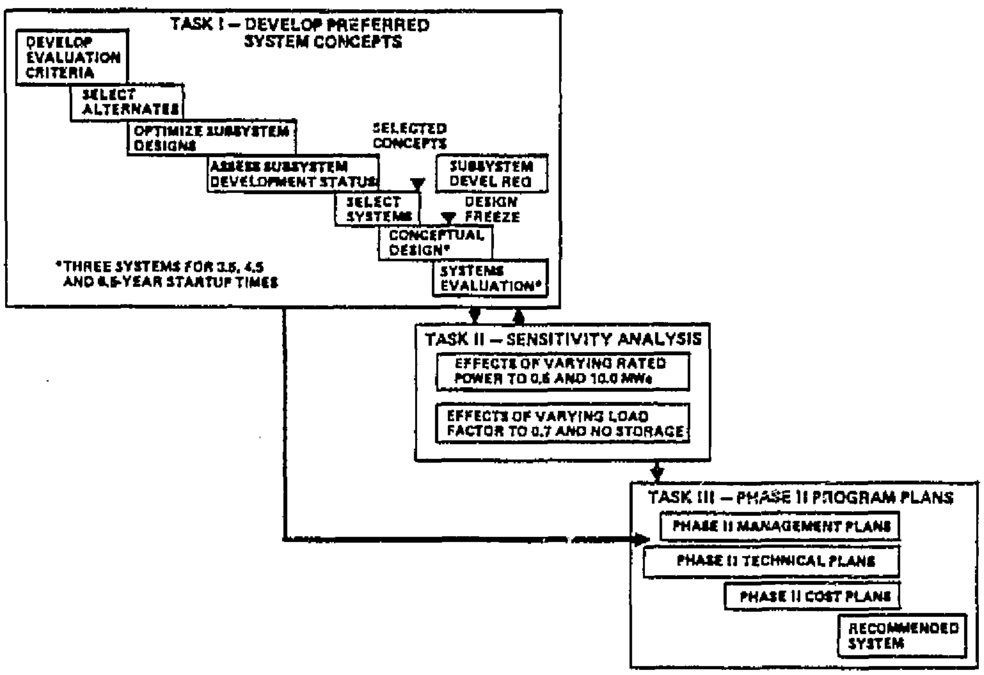


Figure 1-2. Top-Level Study Flow

for a nominal 1.0 MWe rated capacity and 0.4 capacity factor. Activities in Task I through the selection of the three preferred system concepts were primarily a systems engineering/evaluation conducted by MDAC. Subsystem characteristics, performance, and preliminary development requirements were supplied by the appropriate subcontractors. Following this concept selection, the conceptual design of subsystems was initiated in which descriptions, finalized development requirements, performance, reliability, and cost data for each of the three selected concepts were developed.

In Task II, the impact of varying rated power (0.5 and 10.0 MWe) and system capacity factor (zero storage case and 0.7) was investigated. Sensitivity analysis in Task II was performed by MDAC using subsystem data supplied by the subcontractors. This task featured system and subsystem reoptimization for each of the cases evaluated.

In Task III, the management, technical and cost plans for Phase II for each of the three selected concepts were to be prepared in accordance with JPL guidelines and MDAC system recommendations were to be provided. However, as reviewed above, during the latter period of the contract, JPL directed MDAC to terminate all Task III efforts. Accordingly, Task III efforts were discontinued and Phase II Program Plans are not reported.

1.2 ROLES AND RESPONSIBILITIES

A team of companies led by the McDonnell Douglas Astronautics Company (MDAC) was contracted to conduct the Phase I definition of Category A systems (general only, excluding dish concentrators). The team includes MDAC, Rocketdyne, Stearns-Roger, the University of Houston Energy Laboratory, and Energy Technology, Incorporated (ETI). MDAC was the prime contractor for the effort and was responsible for overall contract compliance. The four major subcontractors and their prime areas of responsibility were: (1) Rocketdyne Division of Rockwell International (receiver, dual-media energy storage), (2) Energy Technology, Inc. (radial turbine and gearbox), (3) Stearns-Roger (tower and plant layout/equipment), and (4) University of Houston Solar Energy Laboratory (collector field optimization).

1.3 SYSTEM SUMMARY

From the preliminary design analyses efforts to date, MDAC concludes that the proposed central receiver power system concept is a feasible, low-cost, and low-risk approach for a small solar power system experiment. It is particularly suitable for early deployment under the 3.5- and 4.5-year programs. The concentrator subsystem is currently under development and low-cost, high-production rate heliostats will be available for this program. The proposed receiver subsystem using Hitec is similar to existing fossil fired/Hitec heaters. The tower is a standard low-cost guyed steel tower. The energy transport system using Hitec is based on standard state-of-the art equipment and operating conditions. For the 3.5- and 4.5-year programs, a simple two-tank storage subsystem is proposed which requires no development. The power conversion system is based on existing axial steam turbines. All the balance of plant equipment involves state-of-the-art equipment and processes. The 6.5-year program contains development of a radial outflow turbine and qualification of a dual media thermocline storage subsystem. The technology employed in all programs is consistent with the development time available. Thus, the proposed MDAC concepts satisfy all of the important JPL selection criteria, namely, high operational reliability, minimum risk of failure, good commercialization potential, and low program costs.

1.4 CONTENTS OF VOLUME II

This volume includes all work conducted under Task II leading to selection of the three system concepts preferred for the three program durations. A brief summary of results are contained in Section 2, and more detailed information is given in Section 3. Additional supporting analyses are given in the appendixes. The results of this effort were the selection and definition of preferred system concepts for each of the three EE-1 programs. The preferred system concepts were then analyzed and evaluated in more detail in subsequent study efforts, as reported in Volumes III, IV, and V.

Section 2
SUMMARY OF SYSTEM EVALUATION AND SELECTION

A brief summary of the approach and results from the initial concept evaluation process leading to selection of the three preferred system concepts is given in this section.

2.1 SYSTEM SELECTION CRITERIA

System selection criteria to support subsystem and system selection for final design and analyses efforts, were developed. These selection criteria are summarized in their order of importance on Table 2-1. For each criteria, key issues and approaches to solution were assessed, and sensitivity factors were developed, where possible. These criteria were integrated into the selection and optimization of candidate subystem designs. Final concept comparisons and selection of the preferred candidates were made using these criteria.

2.2 SELECT CANDIDATES FOR EVALUATION

Candidate subsystems/components were identified and synthesized into complete systems for a 1-MWe solar electric power plant. The candidates considered for the collector, collector fluids, thermal storage and prime mover subsystems are listed in Table 2-2. To ensure that a complete range of options would be reviewed, subsystem components ranging from conceptual to off-the-shelf designs were considered. All candidate systems were then compared on the basis of technology readiness, projected system costs, and other factors and a first filtering of candidates was made. The candidates selected for further optimization are listed on Table 2-3 for the 3.5, 4.5, and 6.5 year programs. These candidate systems were subsequently studied further and reduced to one preferred system for each of the startup times. Concepts rejected by initial screening included all distributed collectors (due to high cost/low performance), the liquid sodium systems (system complexity and potential hazard),

Table 2-1
SELECTION CRITERIA (IN ORDER OF IMPORTANCE)

1) High Operational Reliability - Selected system concepts should lead to:
   - An experimental plant which will start up satisfactorily and operate with a high degree of reliability
   - Small power systems with an ultimate reliability which approaches that of a commercial power plant

2) Minimum Risk of Failure - Selected concepts should minimize development risk and thereby provide high confidence that the startup times will be met

3) Commercialization Potential - Selected concepts should use or contribute directly to the eventual systems that are likely to achieve commercial success in the late 1980s
   - Costs/performance
   - Flexibility
   - Institutional interface aspects

4) Low Program Costs - Concepts should be selected to minimize the estimated development and capital costs of Phases II and III

the saturated steam/steam engine system (high cost/low performance), the caloria/axial turbine (high cost), the Brayton cycle/thermal storage system (heavy tower mounted equipment), and thermochemical or latent heat storage subsystems (high cost, development requirements, and system impact).

2.3 OPTIMIZE SUBSYSTEM DESIGNS

The system and subsystem designs for the concepts selected for the 3.5, 4.5, and 6.5 year startup times (Table 2-3) were optimized prior to final comparison. Subsystems included the concentrator design, concentrator field,

Table 2-2
CANDIDATE SUBSYSTEMS

Collector

- Central receiver
- Distributed collectors
  - Parabolic trough
  - Segmented mirrors

Collector fluids

- Heat transfer salts
- Liquid sodium
- Syltherm
- Caloria HT-43
- Therminol 66
- Water/steam
- Air
- Helium

Energy storage

- Sensible heat
  - Dual media/thermocline
  - Single medium/thermocline
  - Two-tank
  - N-tank
  - Trickle charge
- Latent heat
- Thermochemical
- Battery

Prime Mover

- Radial steam turbine
- Axial steam turbine
- Organic turbines
  - Subcritical
  - Supercritical
- Reciprocating steam engine
- Gas turbines

receiver configurations, tower concepts, energy storage concepts, energy transport, power conversion and plant control. The performance potential of alternate cycles and fluids was also investigated as part of the system optimization.

2.4 SUBSYSTEM DEVELOPMENT STATUS

An assessment of the development status of each candidate subsystem was made and approaches to resolve technology issues were developed. Since a high startup reliability is desired for the first experimental unit, this evaluation became very important in the overall assessment and selection of preferred systems. The stipulated 3.5 year porgram only permits an 8-month Phase II

Table 2-3

CANDIDATES FOR THREE PROJECT DURATIONS - 1

Receiver fluid	3-1/2 years	4-1/2 years	6-1/2 years
HTS			
Temperature limit (°C)	430-510	510	510-580
Thermal storage	Two-tank	Two-tank Dual-media therm	Dual-media therm
Prime mover	Axial turbine	Axial turbine Radial turbine	Radial turbine
Syltherm			
Termperature limit (°C)	400-454	450-480	450-480
Thermal storage	Trickle charge	Trickle charge Dual-media therm	Trickle charge Dual-media therm
Prime mover	Axial turbine	Axial turbine Radial turbine Supercritical organic	Radial turbine Supercritical organic
Caloria			
Temperature limit (°C)	300-316	316	316
Thermal storage	Two-tank Dual-media therm	Two-tank Dual-media therm	Dual-media therm
Prime mover	Subcritical organic	Subcritical organic Radial turbine	Radial turbine

Table 2-3

CANDIDATES FOR THREE PROJECT DURATIONS - 2

Receiver fluid	3-1/2 years	4-1/2 years	6-1/2 years
Saturated steam	X		
Temperature limit (°C)		500-600	500-600
Thermal storage		Pressurized water	Pressurized water
Prime mover		Radial turbine	Radial turbine
Air	X		
Temperature limit (°C)		680-820	680-820
Storage		Battery	Battery
Prime mover		Gas turbine (open)	Gas turbine (open)
Helium	X		
Temperature limit (°C)		680-820	680-820
Storage		Battery	Battery
Prime mover		Gas turbine (closed)	Gas turbine (closed)

development and test period, and therefore, only existing components/subsystems with minor modifications were permissible. The 4.5 year program permits an 18-month Phase II period, in which some development testing can be accomplished. The 6.5 year program permits up to 42 months (3.5 years) of development testing in which advanced technology concepts can be pursued.

2.5 SELECTION OF PREFERRED SYSTEM CONCEPTS

System configurations were synthesized and optimized for the candidates shown in Table 2-3. The selection criteria from Table 2-1 were utilized to compare these systems and select the three preferred concepts for the three programs. Systems based on heat transfer salt as receiver coolant and thermal storage fluid were selected for all three programs.

Operating temperature ranges were lower for the shorter-duration programs to accommodate the relatively short development periods available and to minimize program risks and costs. Similarly, two-tank thermal storage concept was selected to be relatively simple for the 3.5 and 4.5 year programs while dual-media thermocline storage was specified for the 6.5 year program. Existing axial steam turbines were selected for the shorter programs compared to the more advanced radial outflow steam turbine for the 6.5 year program. The results of this assessment task are given in Section 3.4.

Section 3
TECHNICAL REVIEW

This section contains the technical information used for the System Comparison and Selection of the three preferred system concepts. Related supporting data on costing and availability are contained in Appendices A and B, respectively.

3.1 SYSTEM SELECTION CRITERIA

For the Small Power Systems Program, which seeks to advance solar energy system technology, the risks are not limited to those normally associated with technological projects, such as performance, safety, and schedule. These program risks also encompass the inability to perform to specification, resulting in lower than expected efficiency, higher than expected production or operational costs, or unforeseen maintenance requirements. In addition, because devices for collecting solar energy are, of necessity, large and highly visible, the risks extend to relatively intangible areas such as institutional image, aesthetic appeal, and the effect on interfacing communities and their life styles.

For these reasons, and also in order to make meaningful comparisons between the power system concepts proposed by each contractor, JPL has identified several system selection criteria that are to be used in the assessment of each concept. The objective of this section is to further develop these selection criteria to the subsystem levels as required to support subsystem and system selection.

These system concept selection criteria in descending order of importance are:

1. High operational reliability
2. Minimum risk of failure

3. Commercialization potential

4. Low program cost

Each of these criteria are reviewed in more detail in the following subsections.

As shown on Table 3.1-1, the overall approach to meet the objective of this task is to define, extend and quantify each of these system selection criteria. Goals were established, where possible, and the contribution of each major subsystem to achieving these goals was identified. During this process, key issues were identified and possible approaches to solution were assessed. Sensitivity factors were also developed, where possible. Finally, a methodology for final concept comparison and selection was developed.

Table 3.1-1
DEVELOP SYSTEM SELECTION CRITERIA

<u>Objective</u>

Develop system selection criteria at levels appropriate for subsystem selection

<u>Application</u>

Selection of candidate concepts (Initial Screening)

Final selection of three preferred concepts

<u>Approach</u>

- Define, extend and quantify system selection criteria
- Establish goals to be pursued
- Identify the contributions of major subsystems to each target goal
- Assess key issues and approaches to solution
- Develop sensitivity factors, where possible
- Develop a methodology for final concept comparisons and selections

Some of these criteria were used to initially screen candidate systems and subsystems for further evaluation. All of the criteria were used in the final evaluation of candidate systems and the selection of the preferred system concepts for the three program start-up times.

3.1.1 High Operational Reliability

The initial definition of High Operational Reliability was defined by JPL in Reference 1 as follows: "The system concept should lead to a small power system with an ultimate reliability which approaches that of a commercial power plant." Further expansion of this criteria was furnished by JPL in Reference 2 as follows: "Engineering Experiment #1 is the first Small Power Systems Application in the Solar Thermal Program to be used in a utility and therefore will have a high visibility to persons in positions of responsibility for solar programs. It is important, then, that the Phase I concept selected for development during Phases II and III will lead to a highly reliable experiment; one which will start up satisfactorily and operate with a high degree of reliability." Additionally, "enhancement of reliability through modularity/redundancy should be considered," was added to this criterion. Our interpretations of these requirements are given below.

The initial definition requires that the system selected for Phase II and III should lead to a system that will eventually have an operational reliability or availability that is comparable to existing commercial power plants in the 1 MWe power range. As will be shown later, this implies an operational availability of approximately 0.95. The subsequent definition of operational reliability implies that the experiment itself must (1) start up satisfactorily and (2) operate with a high degree of reliability.

For the experimental unit to operate with a high degree of reliability implies that the experimental unit must either: (1) utilize highly reliable already-developed components with a history of good operational reliability or (2) undergo a long development/qualification period to ensure high operational system reliability after start-up. The 3.5-year program does not permit a long development/qualification test phase (8-month Phase II) and therefore, for this program, only existing components with minimum modifications should be used.

Table 3.1-2

HIGH OPERATIONAL RELIABILITY

OBJECTIVE: SELECTED SYSTEM CONCEPTS SHOULD LEAD TO SMALL POWER SYSTEMS WITH AN ULTIMATE RELIABILITY APPROACHING THAT OF A COMMERCIAL POWER PLANT (AVAILABILITY $\cong$ 95%)

RELIABILITY ALLOCATIONS:

	OUTAGE (HR/YR)		MAINTENANCE (HR/YR)	
	FORCED	PLANNED	CORRECTIVE	SCHEDULED
COLLECTOR	3	15	280	540
PWR CONVERSION	28	105	145	1380
ENERGY TRANSPORT	3	15	15	50
ENERGY STORAGE	1	15	5	70
CONTROL	0	0	5	10
TOTAL (HR/YR)	35	105*	450	2050
	140		2500	
OUTAGE RATE**(%)	1.0	3.0		
	4.0			
AVAILABILITY (%)	96.0			

*BASED ON SIMULTANEOUS MAINTENANCE FOR PLANNED OUTAGE

**BASED ON YEARLY OPERATION OF 3504 HOURS (LOAD FACTOR = 0.4)

The 6.5-year program allows a 3.5-year Phase II development test phase, and therefore, new components using advanced technology are permissible. The 4.5-year program permits an 18-month development test phase, which is adequate for limited development of advanced components depending upon the development/qualification testing required for each component.

The assessment of a system with an ultimate reliability approaching that of a commercial power plant can be more easily addressed. Appendix B of this report treats preliminary availability estimates for all of the candidate systems. From a review of commercial power plants covering a broad power range, an availability goal of 0.95 has been established for a representaitve power plant at the 1-MWe power level. A target reliability allocation is shown on Table 3.1-2. As reviewed in Appendix B, this goal can be met by most of the candidate systems.

There are several approaches to improving the reliability/availability of power systems. Components with high reliability can be selected or designed. Redundant components or backup systems can be added to the system. Scheduled maintenance can be increased (at some additional cost) to preclude failures. Historically, the most effective approach to enhance both start-up and ultimate reliability, is to maintain design and operational simplicity. This will be the principal guideline in the development and assessment of the candidate systems.

In Reference 2, JPL suggested that the contractors should consider the enhancement of reliability through redundancy associated with modular design. It was found that modularity has a small impact on total system availability but at substantial increase in capital and maintenance costs. This is discussed further in Appendix B.

3.1.2 Minimum Risk of Failure

The initial definition of Minimum Risk of Failure defined by JPL in Reference 1 is as follows: "The system concept should be selected in such a way that it lends itself to subsystem development which is achievable within the Phase II time (8 mo., 18 mo., or 42 mo.) and minimizes the risk of failure that the small power system can be brought on line at the selected start-up time (3.5 years, 4.5 years, or 6.5 years)." Further clarification of this criteria was furnished by JPL in Reference 2, as follows: "The thrust of this criterion is to assure a minimum development risk, and thereby provide a high degree of confidence that the start-up time will be met with the systems selected. Consideration should be given to selecting concepts that have hardware available with proven performance so that new hardware development within Phase II can be minimized." Moreover, as mentioned in Section 3.1.1, JPL has placed a high emphasis on the ability of the selected experimental system to start-up satisfactorily.

Our overall interpretation of this criteria is that the selected system concepts must achieve subsystem development and verification within specified Phase II times and achieve operational status with specified start-up times. The phase schedules for each of the three start-up programs are shown on Figure 3.1-1.

Phase I represents this current 10-month conceptual design effort for each of the three program start-up times. Phase II, which varies from 8 to 42 months, is to include system design, and the design, fabrication, test and evaluation of any subsystem or component that requires development testing. Phase III, which varies from 22 to 24 months, includes plant fabrication, installation and checkout. The actual test and evaluation of the experimental plant is to proceed for the subsequent 12-month period.

In order to have a high confidence that these schedule milestones can be met, it becomes clear that the key issue is the extent of development/verification testing required by subsystems or components. For the 3.5-year start-up program, which only permits an 8-month Phase II testing period, development testing of new hardware is virtually out of the question. Existing hardware and operating techniques must be used with minimal modifications. Proven technology

MINIMUM RISK OF FAILURE

OBJECTIVE: SELECTED SYSTEM CONCEPTS SHOULD ACHIEVE SUBSYSTEM DEVELOPMENT WITHIN SPECIFIED PHASE II TIMES (8, 18 AND 42 MONTHS) AND ON-LINE OPERATIONAL STATUS WITHIN SPECIFIED START-UP TIMES (3.5, 4.5 AND 6.5 YEARS)

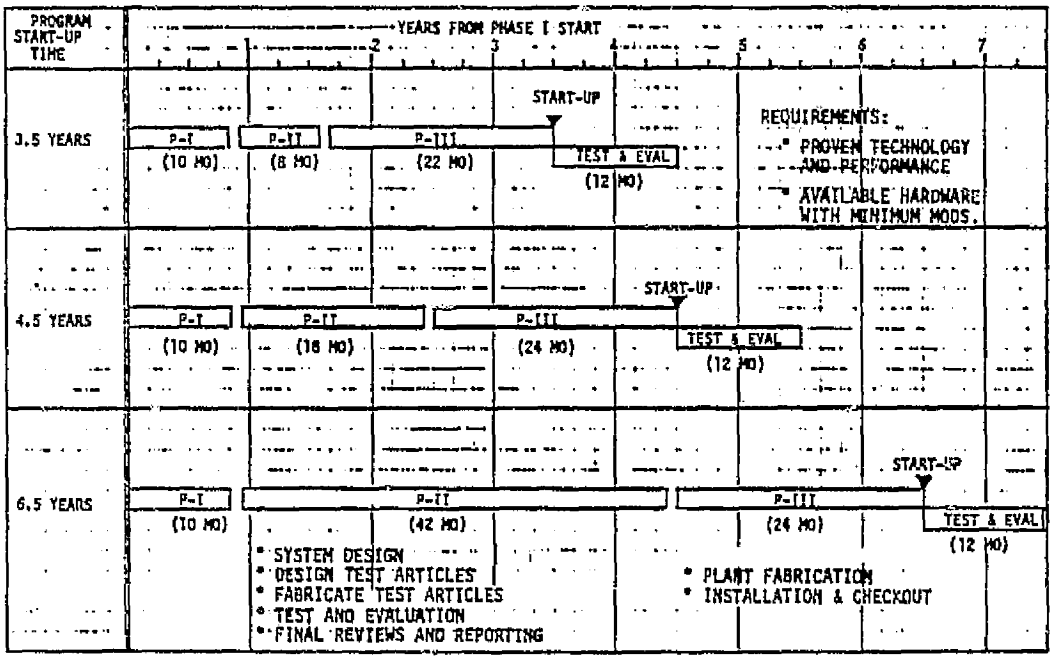


Figure 3.1-1. Minimum Risk of Failure

and performance techniques must be utilized and only verified during Phase II. For the 4.5-year startup program, an 18-month Phase II test period is allowed. This will permit some limited technology development and more extensive modifications of existing hardware and proven design/operational practices. The 6.5-year startup program permits a 3.5-year Phase II testing period. For this program, the use of advanced subsystems and design/operational techniques becomes quite feasible.

For each of the candidate system concepts, a qualitative assessment was made to determine the degree of confidence that the specified times will be met. Consideration was given to both the Phase II development period and Phase III operational startup times. Long-lead time items, testing facility and equipment availability, and critical testing durations were considered. Also, the system impact of test results that indicate substandard performance was considered. For those concepts that have appreciable risks, backup alternatives to minimize the total program risk were also identified.

3.1.3 Commercialization Potential

JPL has defined Commercialization Potential in Reference 1 as follows: "The system concept should use or contribute directly to the eventual concepts and systems that are likely to achieve commercial success in the late 1980's." Further clarification by JPL was given in Reference 2, as follows: "In the process of evaluating potential system concepts against this criterion, several important factors incorporated with this criterion should be considered. First, in order that eventual commercialization be realized, the system concept selected should be compatible with small community and utility applications requirements (e.g., utility interface, environmental and resource impacts, safety, aesthetics, etc.). Also, the concept selected should be adaptable to applications other than utility applications. In both cases, i.e., compatibility of the concept with the application and the adaptability of the concept to other applications, modularity of design should be one of the primary considerations. Modularity refers to the system approach being of such a design that the power plant can meet the power requirement in the 1-10 MWe range with a minimum change in design and minimum effect on overall performance. Finally, the selected concept, when fully upgraded (developed), should lead to both low capital costs and low energy costs for mass-produced plants. To achieve the low energy costs suggests that the selected concept should be of such a design that plant operation is relatively simple, thereby minimizing or eliminating the need for skilled plant operators, and minimizing operations and maintenance costs."

In view of these objective, we have elected to cover all commercialization potential requirements within three basic categories, which are: (1) Costs/Performance, (2) Flexibility, and (3) Institutional Interface Aspects. Each of these categories is reviewed below.

3.1.3.1 Costs/Performance

To meet this objective, selected concepts should lead to small power systems with ultimate energy costs that compete with commercial power plants in the late 1980's. In order to better assess the design and operational implications of this requirement, it was necessary to make some estimates of the operating costs of representative fossil-fueled and solar power plants in the late 1980's. Levelized busbar energy costs were estimated based on the technique described in Reference 3. Constants defined by JPL in Reference 4 for the life-cycle energy cost program were used together with other assumptions made by MDAC. The results indicated that levelized busbar energy costs ($/KWH) of solar energy plants and commercial oil-fueled plants will be competitive in the 1980's. The flow down of this criterion relative to concept selection is shown on Table 3.1-3 and are reviewed below.

Capital recovery and the costs of maintenance replacement parts for solar powered systems represent approximately 60 percent of the total energy costs. Therefore, it is prudent to initially select low-cost/high-performance subsystems and design for high reliability/long-life to reduce replacement costs.

Since fuel costs are zero for a solar plant, the operations and maintenance crew requirements represent the remaining 40 percent of the total energy costs. Thus, subsystems should be designed to reduce manned operation (particularly skilled operators). Also, design of high-reliability/long-life components to reduce maintenance requirements will help to reduce these costs. The necessary planned maintenance should be optimized to minimize crew manhours, and it may be possible to share skilled repairmen from nearby operating solar power plants. Operations costs may also be reduced by providing specialized servicing equipment to reduce crew manhours required for servicing. These factors were taken into account during the assessment and selection of the preferred system concept.

3.1.3.2 Flexibility

To meet this objective, the selected concepts should exhibit the flexibility to supply energy over a wide range of applications without major system impact. As indicated on Table 3.1-4, we have interpreted this requirement as the ability to provide incremental electrical energy output from 0.5 to 10.0 MWe and vary the load factor from no storage to 0.7. Furthermore, the scope of this requirement may imply stand-alone capability for applications other than utility support. Moreover, the applications may include shaft power for pumps, pulleys, milling, etc. Provision of thermal energy in addition to or in lieu of electricity would further enhance the system's flexibility.

Table 3.1-3

COMMERCIALIZATION POTENTIAL - LOW ENERGY COSTS

Objectives	Selected concepts should lead to small power systems with ultimate energy costs that compete with commercial power plants in the late 1980's
Approach	• Capital recovery and maintenance parts (59% of total costs) - Select low-cost/high-performance subsystems - Design for high-reliability/long-life to reduce replacement costs • Operations and maintenance crew requirements (41% of total costs) - Design subsystems to reduce manned operations (particularly skilled operators) - Design for high-reliability/long-life to reduce maintenance requirements - Optimize planned maintenance to minimize crew manhours (share skilled repairmen) - Provide specialized servicing equipment to reduce crew manhours

Table 3.1-4

COMMERCIALIZATION POTENTIAL - FLEXIBILITY

<u>Objective</u>	Select system concepts that exhibit the flexibility to supply energy over a wide range of applications without major system impact
<u>Scope</u>	• Incremental electrical energy output (from 0.5 to 10.0 MWe) • Varying load factor (from no storage to 0.7) • Other than utility applications (stand-alone capability) • Shaft power
<u>Issues</u>	• User requirements (type, amount, timing) • Costs to provide flexibility

3.1.3.3 Institutional Interface Aspects

To meet these requirements, the selected system concepts must satisfy basic interfaces with communities, utilities, and business concerns. As indicated on Table 3.1-5, these interfaces involve many factors. The environmental impact on a community must be considered in which noise, air pollution, water pollution, flood control, erosion and dust control and plant and animal ecology may be important. The hazards of certain systems must be considered which include explosion, fire, toxicity, radiation, leaks and glare problems. Aesthetics may also become important to some communities concerned with general appearance, landscaping, access and traffic impact.

The interfaces with existing utility grids can involve operating power profiles, dynamic interactions, control, emergency provisions and special interface equipment requirements. Nonoperating interfaces can involve start-up and shutdown requirements, planned outage for maintenance, forced outage and emergency procedures, and nonoperable periods.

The interfaces with local business concerns can involve ownership and control, power rates, profits, flexibility for future growth, local employment and taxes, emergency provisions, and stand-alone capabilities.

The major institutional interface factors to be considered with respect to selection of the preferred concepts are potential hazards and utility familiarity with the technology.

3.1.4 Low Program Cost

As defined by JPL in Reference 1, "The system concept should be selected to minimize the estimated costs of Phase II and Phase III." Further clarification by JPL was given in Reference 2, as follows: "The thrust of this criterion is to minimize Engineering Experiment #1 development and capital costs. To this end, consideration should be given to selecting concepts which have hardware available with proven performance so that development costs associated with

Table 3.1-5

COMMERCIALIZATION POTENTIAL - INSTITUTIONAL INTERFACE ASPECTS

OBJECTIVES - SELECT SYSTEM CONCEPTS THAT SATISFY BASIC INTERFACES WITH COMMUNITIES, UTILITIES AND BUSINESS CONCERNS

COMMUNITY INTERFACES

ENVIRONMENTAL IMPACT

- NOISE
- AIR POLLUTION
- WATER POLLUTION
- FLOOD CONTROL
- EROSION AND DUST CONTROL
- PLANT AND ANIMAL ECOLOGY

HAZARDS

- EXPLOSION
- FIRE
- TOXICITY
- RADIATION
- LEAKS
- GLARE (AIRCRAFT)

AESTHETICS

- GENERAL APPEARANCE
- LANDSCAPING REQUIREMENTS
- ACCESS ROUTES/TRAFFIC IMPACT

UTILITY INTERFACES

OPERATING INTERFACES

- POWER PROFILES
- DYNAMIC INTERACTIONS
- INTERFACE EQUIPMENT
- CONTROL
- EMERGENCY PROVISIONS
- FAMILIAR TECHNOLOGY

NON-OPERATING INTERFACES

- STARTUP AND SHUTDOWN
- PLANNED OUTAGE FOR MAINTENANCE
- FORCED OUTAGE/EMERGENCY PROCEDURES
- NON-OPERABLE PERIODS

BUSINESS CONCERNS

- OWNERSHIP AND CONTROL
- POWER RATES
- PROFITS
- FLEXIBILITY FOR FUTURE GROWTH
- EMPLOYMENT
- TAXES
- EMERGENCY PROVISIONS
- STAND ALONE CAPABILITY

Phase II can be minimized. In addition, the projected plant performance (i.e., overall efficiency and individual component costs) of the selected concepts should be such that the required capital investment for actual hardware for Engineering Experiment #1 can be minimized."

As indicated on Table 3.1-6, the approach to minimize program costs involves several steps. Technology requirements for the selected concepts should not be over-extended and must meet scheduling limitations. The technology developed from other programs should be used as much as possible. New or advanced technology requiring development testing in Phase II should be fully justified on the basis of performance and costs. The system design and operations should be kept as small as possible. Existing equipment should be used where possible with minimum modifications, and the facilities and tooling developed for similar programs should be used as much as possible.

Table 3.1-6
LOW PROGRAM COSTS

OBJECTIVES	SELECT SYSTEM CONCEPTS THAT MINIMIZE THE ESTIMATED COSTS OF PHASES II AND III
APPROACH	• DO NOT OVER-EXTEND TECHNOLOGY REQUIREMENTS • UTILIZE TECHNOLOGY FROM OTHER RELATED PROGRAMS • FULLY JUSTIFY AND PHASE II DEVELOPMENT • KEEP THE DESIGN AND OPERATIONS SIMPLE • USE EXISTING EQUIPMENT WITH MINIMUM MODIFICATIONS • MAKE MAXIMUM USE OF FACILITIES AND TOOL DEVELOPED FOR SIMILAR PROGRAMS

3.2 SELECTION OF CANDIDATES FOR EVALUATION

The objective of this effort was to identify those Category A system concepts which could be viable candidates for meeting the project's goals. This was accomplished through an initial screening of the very large number of potential candidates. The surviving candidates were then analyzed in greater depth prior to selecting the three preferred systems for the three project durations.

3.2.1 System Requirements

A set of system performance, design, and operational requirements were prepared to ensure that all candidate systems are compared on a common basis.

These requirements reflect information provided in the initial RFP as well as subsequent information provided by JPL. In areas where no information was available, the necessary requirements were based on MDAC design experience derived from other programs.

In all cases, these requirements reflect the level of understanding that was available early in the program. Subsequent clarifications to these requirements provided by JPL (Reference 2) were not fully incorporated into these initial requirements due to their arrival near the completion of this task. During subsequent tasks, however, these clarifications have been fully incorporated into the system requirements.

All systems were sized such that they are capable of producing 1 MWe net power on equinox noon with a direct insolation level of 800 W/m^2. This requirement produces somewhat smaller collector and energy transport subsystems than would occur if the system were sized to provide a 0.4 load factor throughout the year which is the preferred sizing approach as indicated in the JPL clarification (Reference 2). This requirement, however, is sufficient to make system level comparisons between generically similar systems; e.g., central receiver systems. In comparing distributed collector systems to central receiver systems, however, it was necessary to make the comparison on the basis of equivalent load factor due to differences in diurnal and seasonal energy collection performance. For this comparison, a load factor corresponding to the value derived for the central receiver system was used. Work carried out in later tasks conduct all comparative evaluations on the basis of a 0.4 load factor.

Maximum receiver coolant flowrates and energy storage sizing requirements for central receiver systems were defined on the basis of the insolation models shown in Figure 3.2-1. These data correspond to three "good" days which were taken from the Barstow data tape. The coolant flow was sized to handle the maximum energy collection capability as influenced by the concentrator and receiver performance. Energy storage was sized to handle the maximum daily energy surplus. The limiting day for the sizing of energy storage is equinox when concentrator field and receiver performance factors are included.

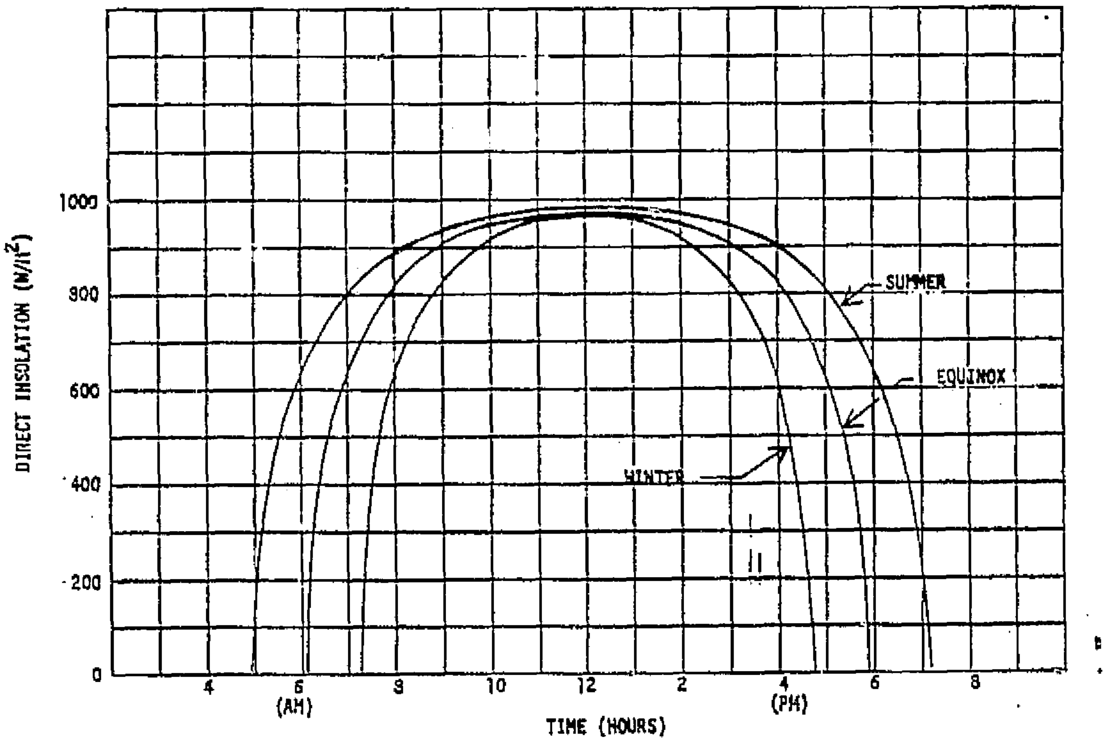


Figure 3.2-1. Reference Barstow Insolation Data

Receiver flowrates and energy storage capacity requirements for distributed collector systems were determined on the basis of conditions required to match the central receiver load factor. All load factor calculations were based on the insolation data presented in Figure 3.2-1.

Other design and performance requirements imposed on the systems include:

- 30-year lifetime with reasonable maintenance
- Availability approaching other commercial systems in the same size range
- Solar-only operation (excludes hybrid configurations)
- 13 m/sec (29 mph) operating wind limit
- 40 m/sec (90 mph) survival wind limit
- Ambient temperature per Table T-3 (Reference 1)
- Hail characteristics per Figure T-2 (Reference 1)
- 0.25 g horizontal acceleration seismic environment

Operationally, each system shall be configured to permit the following operating modes to be carried out:

- Collected energy used to charge storage only with no net electrical output to the grid
- Divert surplus collected energy to storage while providing electrical power to the grid
- Provide electrical power to the grid by simultaneously utilizing collected and stored energy
- Provide electrical power to the grid during non energy collection periods.

3.2.2 System/Subsystem Candidates

System candidates were formulated by combining compatible subsystem sets of: concentrators, collector fluids, energy storage concepts, working fluid and prime movers. The balance of plant equipment was not to be treated as there are significant variations due to choices of the elements specified above. Category A systems can include all concepts not utilizing a dish concentrator. However, as a result of a preliminary screening, concentrator candidates were limited to: (1) point focus central receivers and (2) linear concentrating distributed collectors with 1-axis tracking. Non-tracking collectors were excluded based upon their inherently low combined collection and conversion efficiencies. The linear focus central receiver was excluded based on the assessment that relief from the two-axis tracking requirement on the heliostat will not start to compensate for the greater tower and receiver costs and lower concentration factor when compared to the point focus central receiver at the 1 MWe size. Linear concentrating distributed collectors are represented by the parabolic trough and the segmented mirror concepts which appear to be the two preferred concepts in this class.

The collector fluids considered were:

(1) Heat Transfer Salts
(2) Liquid Sodium
(3) Syltherm
(4) Caloria HT43
(5) Therminol 66 (For distributed collectors only)
(6) Water/Steam
(7) Air
(8) Helium

Of course, the fluids that freeze at ambient temperature, heat transfer salts and liquid sodium, and the less effective heat transfer fluids, air and helium, are excluded for the distributed concentrator candidates.

Candidate thermal storage concepts considered included:

(1) Sensible Heat
  - Dual Media/Thermocline
  - Single Medium/Thermocline
  - Two-Tank
  - N-Tank
  - Trickle Charge

(2) Latent Heat

(3) Thermochemical

Battery storage was considered for concepts not able to utilize thermal storage.

The prime movers and corresponding working fluids in the power conversion subsystem included:

(1) Radial Outflow Steam Turbine

(2) Axial Steam Turbine

(3) Organic Vapor Turbines
  - Subcritical
  - Supercritical

(4) Reciprocating Steam Engine

(5) Gas Turbines (Air and Helium)

3.2.3 Methodology for Candidate Screening

Since the objective of this screening process was to exclude only those candidates which would have no chance of being selected as one of the three preferred systems following more detailed analysis and optimization, only coarse selection criteria was applied at this time. This comprised successive filters of technology readiness, commercialization potential, and low program cost.

Technology readiness was interpreted as the ability to assure meeting a project schedule in the 3 1/2 - to 6 1/2 - year range. Where advanced development is reqired, the existence of a commercially available backup was considered in meeting this criterion with the proviso that concept integrity is preserved. As an example, an existing steam turbine would be considered as an acceptable backup for a concept employing an advanced steam turbine but not for a concept employing a gas turbine.

Commercialization potential was considered in projected system costs in the last 1980's in a quantitative manner and system complexity and potential hazards qualitatively. Substantial differences in projected system costs was required to reject a candidate on this basis alone. This was interpreted as 25 percent greater cost using wholly consistent cost estimating relationships and 50 percent greater cost using different data sources. System simplicity and avoidance of hazardous materials was only used where significant differences exist to select between candidates having otherwise similar potential. Low program cost was applied to select systems with minimum development requirements unless the potential cost or performance advantage justifies the requisite development.

Following the screen of subsystems and components for technology readiness, candidate systems were synthesized from compatible subsystem sets. The first selection was the collector fluid which establishes the limiting fluid temperature. Selection of the working fluid and prime mover and the corresponding cycle efficiency allowed calculation of the collector area requirements and preliminary sizing of the energy transport loop. The collector fluid and temperatures together with the storage capacity requirement allowed selection of the preferred storage concept. Combination of these elements provided a first level system candidate. Other affected system elements, such as tower height or receiver weight, were estimated so that the fullest possible system impact was considered in comparing alternative candidates.

Brayton and Rankin cycle candidates were evaluated separately to select the preferred configurations for subsequent evaluation and comparison in the next project phase.

Alternative thermal storage candidates were evaluated to allow selection of the preferred concept for each system candidate.

Line-focussing distributed collector concepts were first optimized and then compared with the corresponding central receiver candidates.

Following the screening out of the non-viable approaches, the remaining system candidates were grouped according to their suitability for the 3 1/2-, 4 1/2-, and 6 1/2-year programs.

3.2.4 Brayton Cycle

The gas turbine offers several advantages that make it attractive for use in a solar power plant. The turbines are very compact relative to other prime movers and can be readily mounted on the receiver tower. Reliability is high, there are few auxiliaries, and operation is simple. Capital costs can vary widely depending on the efficiency and consequent equipment complexity required.

3.2.4.1 Candidate Cycles

The gas turbine can be configured into different cycle arrangements. Schematics of the three basic configurations are shown in Figure 3.2-2. The first cycle considered is the simple open cycle consisting of a compressor, heater, and the turbo-generator. In this cycle air is drawn from the atmosphere and compressed, heated, and then expanded with sufficient work being extracted by the turbine to run the compressor and the electrical generator.

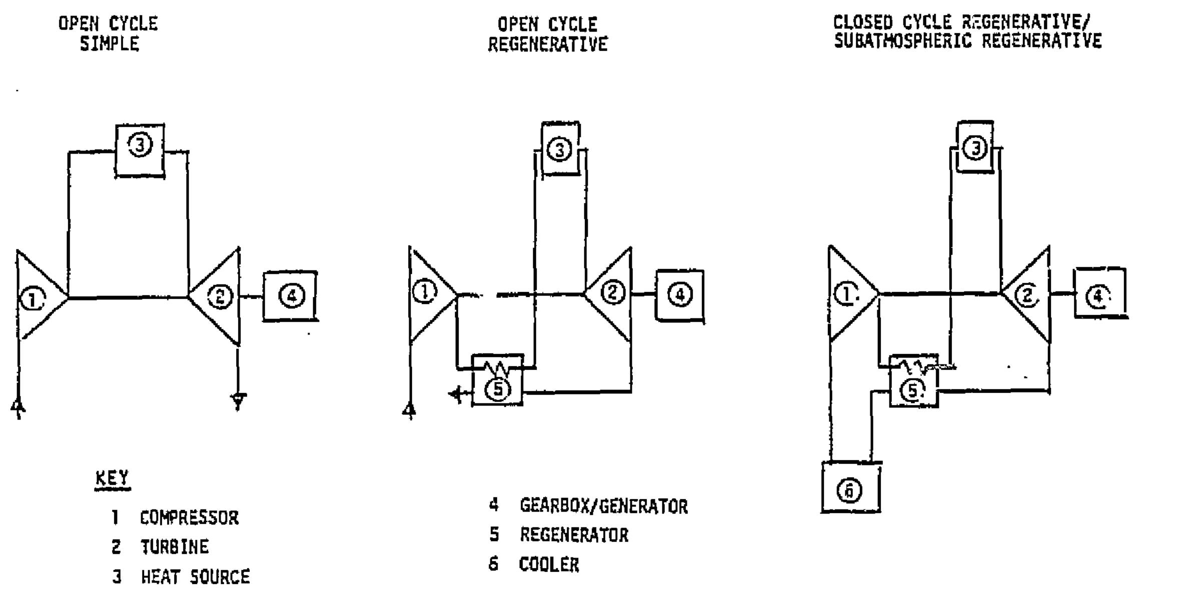


Figure 3.2-2. Brayton Cycle Schematics

This arrangement is the simplest of all gas turbine cycles, but suffers from lowest efficiency and poor part-load performance. The regenerative open cycle offers improved efficiency by utilizing a regenerator as shown in the schematic to recover some of the heat otherwise exhausted to the atmosphere. This cycle may be configured with a single shaft or with two shafts. As a single shaft type, a single gas turbine and its air compressor are mounted on the same shaft and the electrical generator is driven either directly or through gearing from that shaft. A two-shaft machine has two separate turbine units; one is directly coupled to the air compressor and the other is mounted on a separate shaft, which runs at a different speed and drives the electrical generator. The single shaft type is mechanically simpler, can be run up to speed very quickly, has fewer bearings and has better governing response to changes of load; but the two-shaft type gives better part-load efficiency. Since a solar plant may often be forced to run at part-load, this is a major consideration. The third configuration is the closed regenerative cycle. In this cycle, the fluid loop is similar to the open regenerative cycle except a cooler has been added in order to completely isolate the working fluid from the atmosphere. This allows a working fluid such as helium to be used which reduces the optimal pressure ratio with a consequent increase in turbine and compressor efficiency. It also allows the entire circuit to be pressurized resulting in greatly improved heat transfer, reduced component sizes and the ability to control the overall pressure level resulting in excellent part load performance. A modification of this cycle is the subatmospheric regenerative cycle in which the cooler operates at a subatmospheric pressure and the heater at atmospheric pressure. This permits the heater to consist of elements which the air passes over rather than through, hence the heater has no appreciable pressure to contain and can be constructed of any material capable of meeting the thermal requirements. Disadvantages are very large components and moderate part load performance.

3.2.4.2 Cycle Performance

A parametric performance analysis was conducted to compare alternative configurations. Parameters varied included turbine inlet temperature, the ratio of pressure drop at the turbine to the pressure drop at the compressor (β factor) and the regenerator effectiveness in regenerative cycles. Figure 3.2-3 shows the effect of turbine inlet temperatures and β factor on a simple open cycle with somewhat optimistic assumptions. Similar results are shown in Figures 3.2-4 and 3.2-5 for the open and subatmospheric regenerative cycles and the closed cycle.

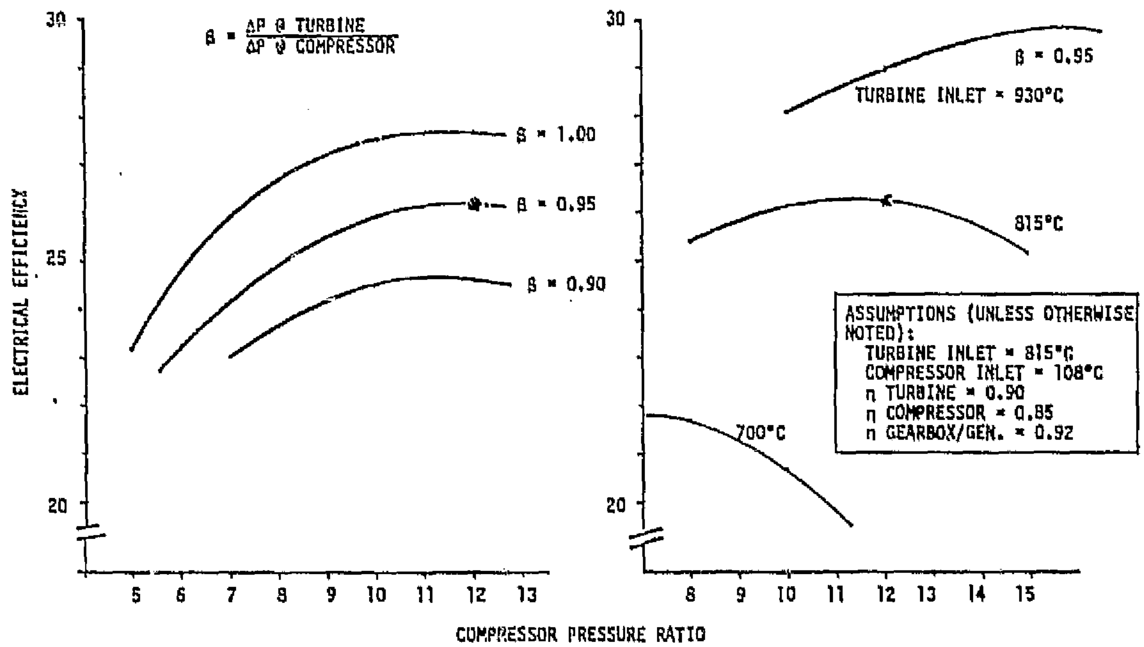


Figure 3.2-3. Parametric Performance Study Simple Open Cycle

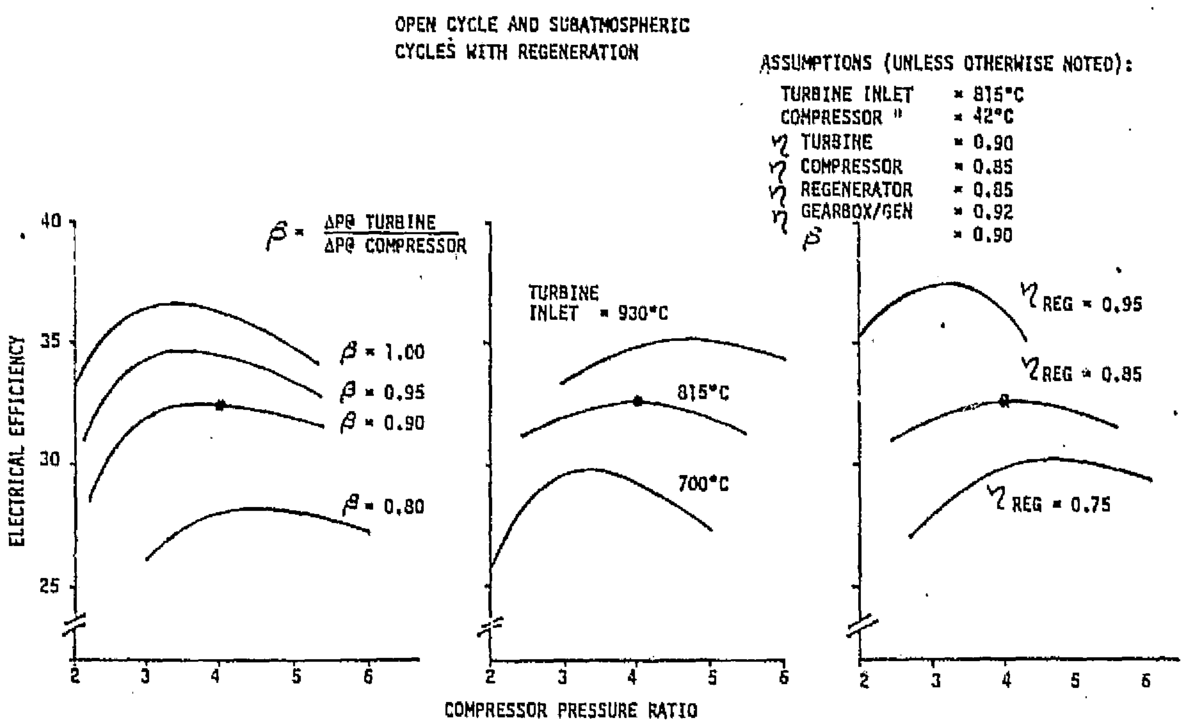


Figure 3.2-4. Parametric Performance Study

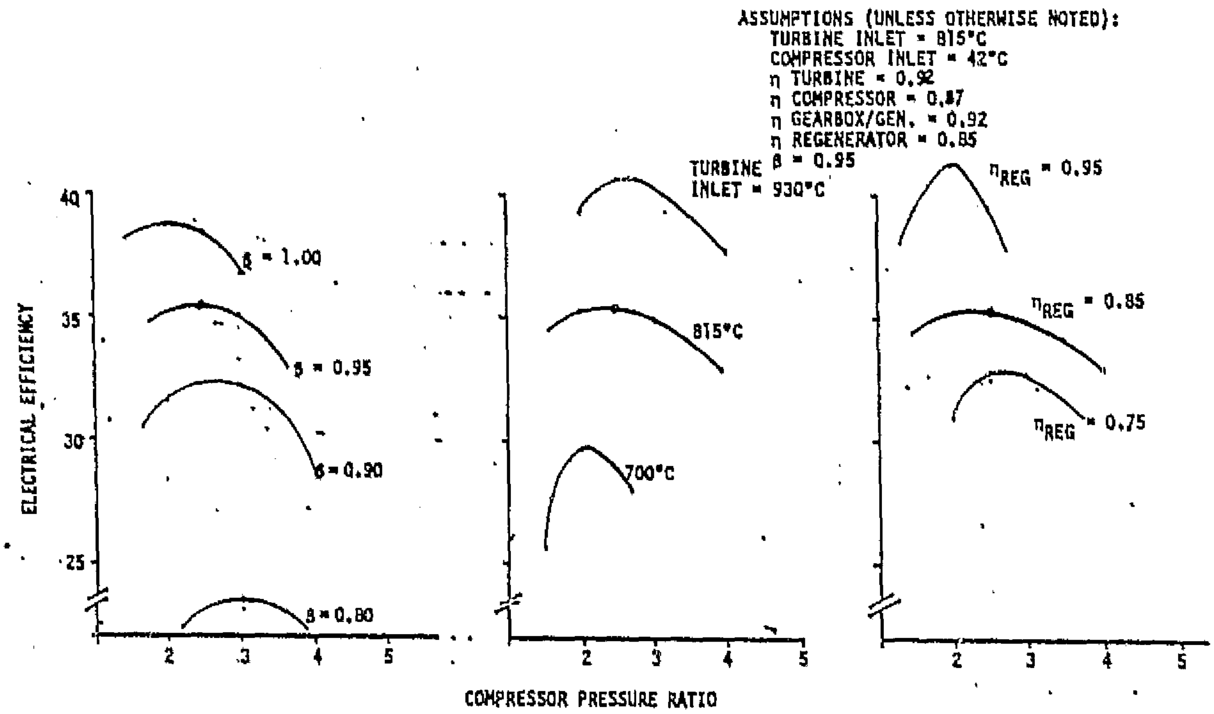


Figure 3.2-5 Parametric Performance Study (Closed Cycle Regenerative)

3.2.4.3 Aperture Size Optimization

Due to the high temperatures at which the receiver must operate, thermal losses due to convection and radiation become a significant factor in the design of the plant. In order to reduce thermal losses a small aperture is needed. A small aperture, however, decreases the percentage of reflected energy intercepted by the receiver, thus lowering the collector field efficiency. The optimum aperture size will then be the aperture which results in the highest thermal efficiency which consists of the product of the receiver and collector field efficiencies. This optimum aperture size will be temperature dependent as thermal losses increase significantly with increasing temperature. Figure 3.2-6 shows the results of the optimization for a receiver operating at 815°C. The optimum aperture size is 3.7 m x 3.7 m with a corresponding overall collection efficiency of 0.42. The collector field efficiency is 0.63 and the receiver efficiency is 0.66. Repeating this process for other operating temperatures, a plot of receiver and collector field efficiency (with an optimum aperture size) as a function of turbine inlet

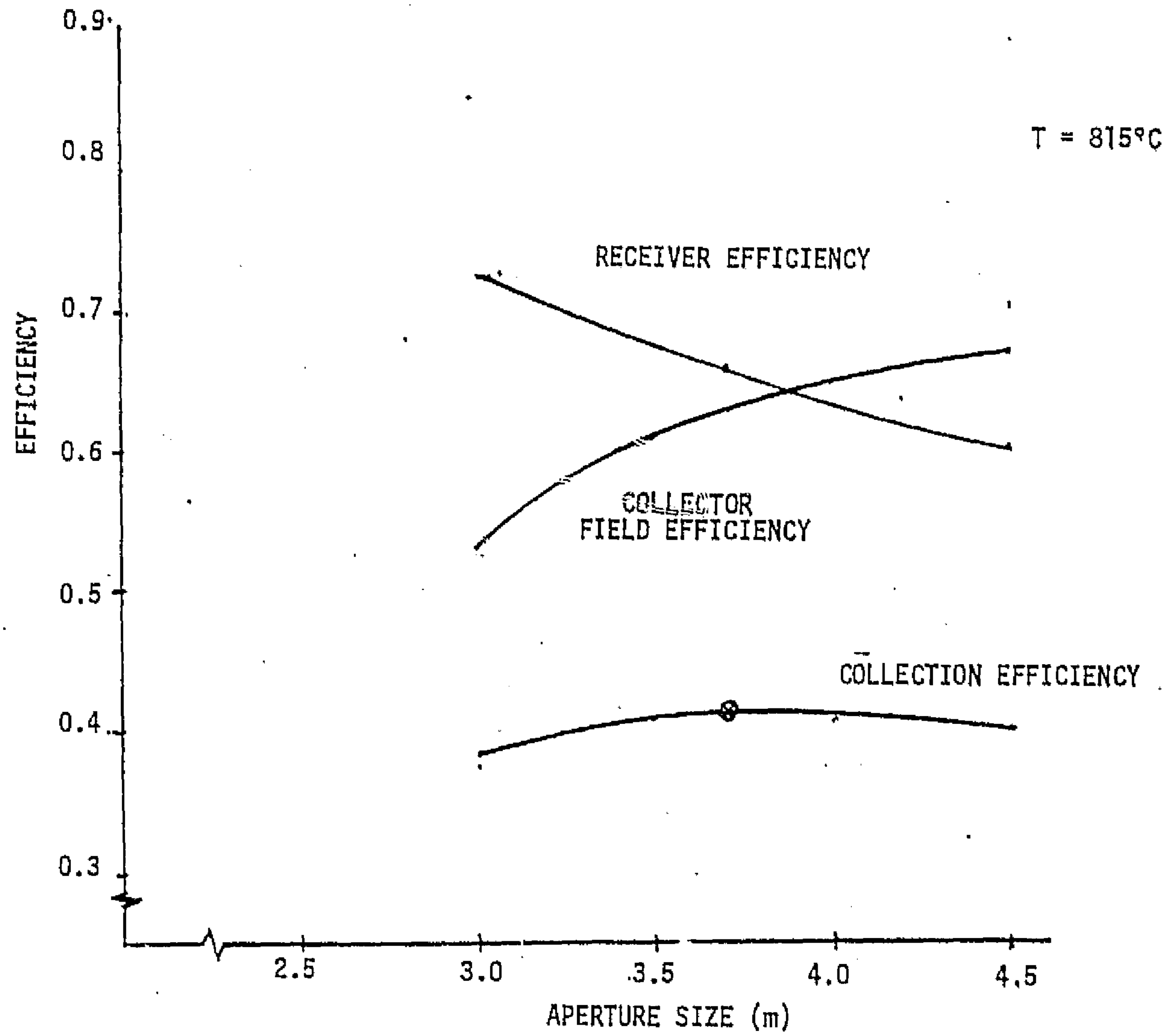


Figure 3.2-6. Aperature Size Optimization

temperature was developed as shown in Figure 3.2-7. Also shown in the cycle efficiency as a function of turbine inlet temperature. Multiplying these three efficiencies gives the net overall efficiency shown. The over efficiency has a peak at 750°C but appears to be relatively insensitive from 700°C to 800°C.

3.2.4.4 Gas Turbine Location

The effect of pressure losses through gas turbine components was analyzed in terms of the β factor in the parametric cycle analysis. The β factor, which

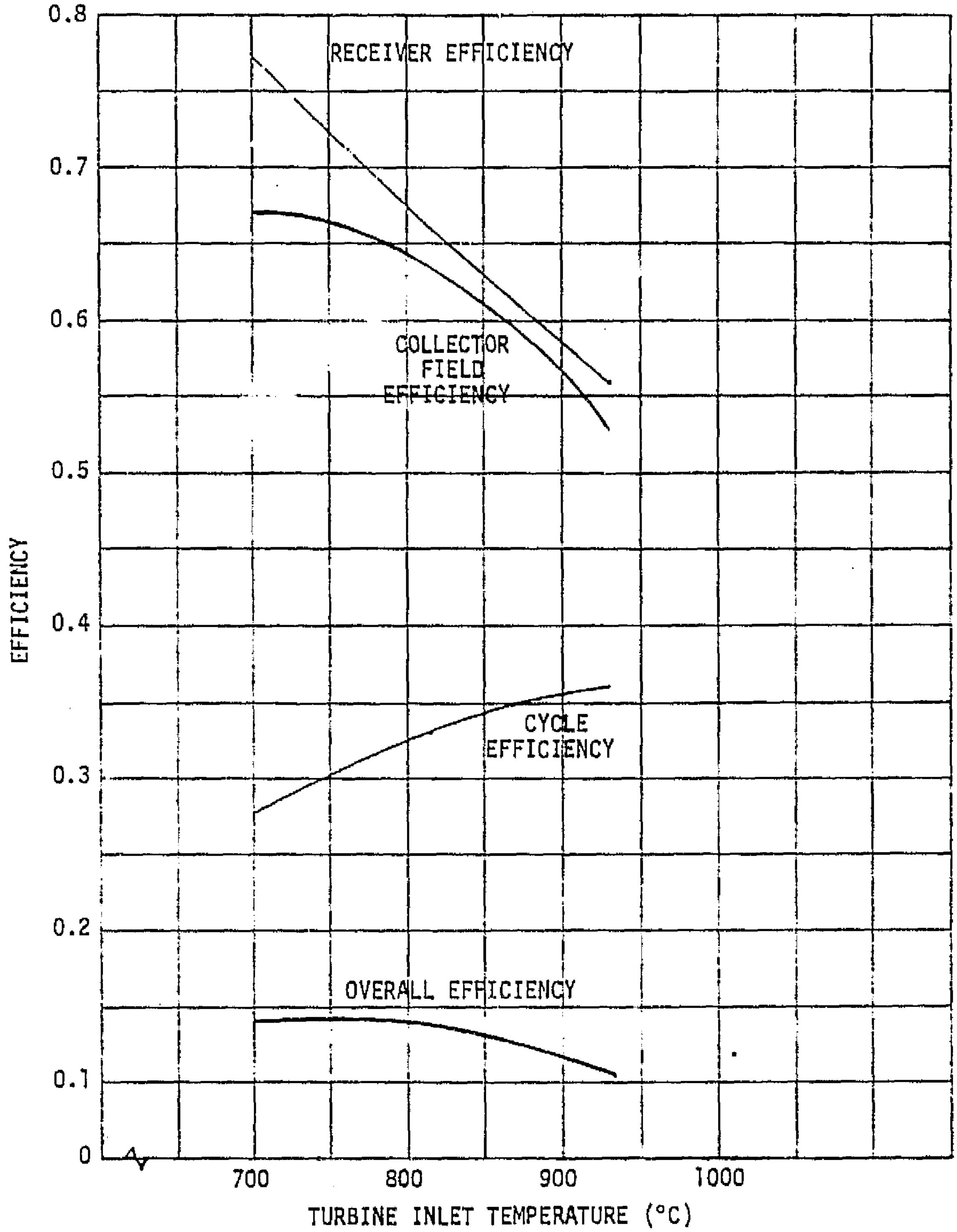


Figure 3.2-7. Turbine Inlet Temperature Optimization

represents the fraction of compressor power lost to friction in the piping, heater, regenerator and cooler, was shown to have a major impact on turbine performance with cycle efficiency falling rapidly as frictional losses increased. In order to determine the size of ducting needed to route gas up and down the tower to a ground-based receiver, some typical $\dot{\beta}$ factors which resulted in acceptable performance was selected. The subatmospheric cycle required a 0.9 m diameter riser and downcomer, the open regenerative cycle 0.6 m, the closed regenerative cycle 0.3 m, and the open cycle 0.3 m. The smallest of the ducts is 30 cm and operates at 12 and 15 bars pressure. The material to be used in these pipes must be similar to that used in the receiver, a material capable of withstanding repeated cycling to 800°C such as Hastelloy. In addition to this expensive material for the large diameter pipes, high temperature thermal insulation must be used on both riser and downcomer. Control valves of this size are a significant additional cost as are mounting provisions to allow for thermal expansion. The conclusion is that a ground based gas turbine creates performance and cost penalties that are unacceptable.

3.2.4.5 Storage Subsystem

Results of the previous discussion limit the location of the power conversion equipment to the tower. If a thermal storage subsystem is to be used then it too must be tower-mounted. This subsystem would most likely consist of a checkerwork refractory such as used in cowpers for blast furnace regeneration, or a pebble bed heater. Assuming an optimistic 25 percent storage capacity utilization, the required mass of brick (MgO) is 216,000 Kg for a 0.4 load factor. Such mass will require a tower that would be prohibitively expensive. For this reason, the use of gas turbines must utilize external (battery) energy storage.

3.2.4.6 Evaluation

The preferred Brayton cycle concept consists of a tower-mounted gas turbine with battery storage. This concept will require the gas turbine to closely follow the power absorbed by the receiver. This in turn requires a gas turbine capable of good part-load performance for acceptable daily efficiency. Since the opportunity does not exist to divert thermal power to thermal storage upstream of the gas turbine, the turbine generator equipment must be sized to accomodate the peak thermal power collected by the receiver. This requires the equipment to be significantly oversized relative to the 1-MWe requirement. The gas turbine configurations which meet this requirement are the closed regenerative cycle and open regenerative cycle with two shafts. These candidates are further developed in Section 3.3.8.

3.2.5 Rankine Power Conversion Cycles

The Rankine power conversion cycles considered during this preliminary evaluation task were:

STEAM RANKINE

- Radial flow
- Axial flow (single stage)
- Axial flow (multistage)

ORGANIC RANKINE

- Supercritical cycle
- subcritical cycle

RECIPROCATING STEAM ENGINE

The preliminary evaluation analyses carried out during this task were aimed at characterizing the equipment in terms of performance, design/operational features and limitations, cost, and hardware availability or development status. On the basis of these analyses, some of the options could be eliminated as being incompatible with the objectives of the overall program while the preferred characteristics for other cycles could serve as the basis for subsequent system synthesis.

3.2.5.1 Steam Rankine Cycles

The implementation characteristics of the three steam Rankine devices considered in this task are shown in Figure 3.2-8 along with an estimate of the corresponding turbine expansion and cycle efficiencies.

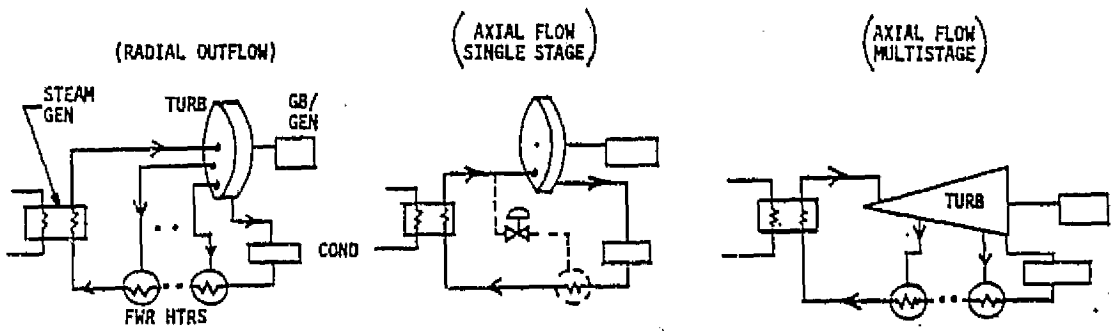
The radial outflow device, shown at the left of the figure, is currently under development. In this concept, high pressure, high temperature steam enters at the center and expands radially outward through a series of rotating and stationary blades. The nature of the device allows for the use of many expansion stages which permits a high turbine expansion efficiency to be realized. This efficiency is further enhanced by the "tight" tolerances and interstage seals that can be utilized in this device. As a result, turbine expansion efficiencies of 0.8 or greater are possible for 1 MWe turbines.

Because of the multistage nature of the device, extraction ports can be provided for regenerative feedwater heating which also leads to a high cycle efficiency. Since this device is under development, standard designs have not yet evolved. As a result, flexibility exists in the specification of the number of extraction ports which may be as high as five for high pressure machines.

The axial flow single stage turbine, shown schematically in the middle of Figure 3.2-8, represents an off-the-shelf device which is available from a variety of suppliers. The single expansion stage employed results in a low turbine expansion efficiency. Also, because it is a single stage device, no opportunity exists for the extraction of steam part way through the expansion process for regenerative heating. As a result, a low cycle efficiency is also realized.

When a subatmospheric condenser is used, as assumed in the schematic, a deaeration function must be provided to remove dissolved gases, particularly oxygen. Since this process must be carried out above atmospheric pressure to permit the venting of this gas, a feedwater heating station must be included to raise the saturation condition of the feedwater from the condenser state to that required for deaeration. Since no extraction steam is available, a controlled turbine bypass line must be included in the design to provide for the needed thermal energy.

The multistage axial flow turbine schematically shown at the right of Figure 3.2-8 also reflects the use of of-the-shelf turbine equipment. The multistage design produces a moderately good turbine expension efficiency while the extraction ports used for regenerative feedwater heating permit the realization of a moderate cycle efficiency.



TYPICAL 1 MWe DATA

N_{TURB}	= 0.80	0.43	0.63
N_{CYC}	= 0.367	0.14	0.25
EXTRACTION CAPABILITY	≤ 5	0	1-2

Figure 3.2-8. Steam Rankine Cycles

One of the factors limiting cycle efficiency is the low number of extraction ports which can be used for feedwater heating. Most domestic turbines are configured to permit a single uncontrolled extraction port. In some turbines, which have been designed for specialized applications, provisions for two extraction ports are included. Axial flow turbines with larger numbers of extraction ports are not in general commercially available based on an off-the-shelf design. To produce a turbine of this type, considerable engineering and design activities would be required with a corresponding increase in project cost.

Additional data related to the three candidate turbines are contained in Table 3.2-1. The steam conditions for the radial turbine reflect nominal

Table 3.2-1

STEAM RANKINE TURBINE/CYCLE CHARACTERISTICS

(1 MWe RATING)

	RADIAL FLOW	AXIAL FLOW	
		SINGLE STAGE	MULTISTAGE
NOM STEAM TEMP LIMIT	480°C (900°F)	400°C (750°F)	510°C (950°F)
NOM PRESS LIMIT	10.3 MPa (1500 PSIA)	4.83 MPa (700 PSIA)	6.3 MPa (910 PSIA)
TYPICAL TURBINE SPEED	12,000 RPM	5,000 RPM	5,000 RPM
GENERATOR TYPE	4-POLE (1800 RPM)	4-POLE (1800 RPM)	4-POLE (1800 RPM)
TYPICAL TURBINE/ GEN COST	∿ $150,000	$70,000	$285,000
LEAD TIME/STATUS	UNDER DEVELOPMENT	30 WEEKS	46-52 WEEKS
MANUFACTURERS	ENERGY TECHNOLOGY, INC STAL-LAVAL AIRESEARCH	COPPUS ELLIOTT TERRY	ELLIOTT STAL-LAVAL TERRY MURRAY TURBODYNE

conditions to which the equipment is currently being designed. The conditions for the axial turbines correspond to conditions for commercially available equipment. In all cases, the turbine speed exceeds the generator speed which necessitates the use of a gear box. Gear boxes designed for these speeds and power levels are commercially available from a variety of sources.

The cost estimates reflect vendor quotations for American built axial flow turbines while the cost for the radial turbine is based on ETI's estimate for future equipment. The list of manufacturers is intended to represent only a partial list of suppliers and reflects primarily American suppliers.

The performance characteristics for each of the three turbines are shown in Figure 3.2-9 as a function of power rating. It is seen that the efficiencies for both of the axial flow turbines decreases dramatically at lower power levels with 1 MWe equipment experiencing relatively poor efficiencies. By contrast, the radial turbine exhibits a high turbine expansion and cycle efficiency at the 1 MWe power level.

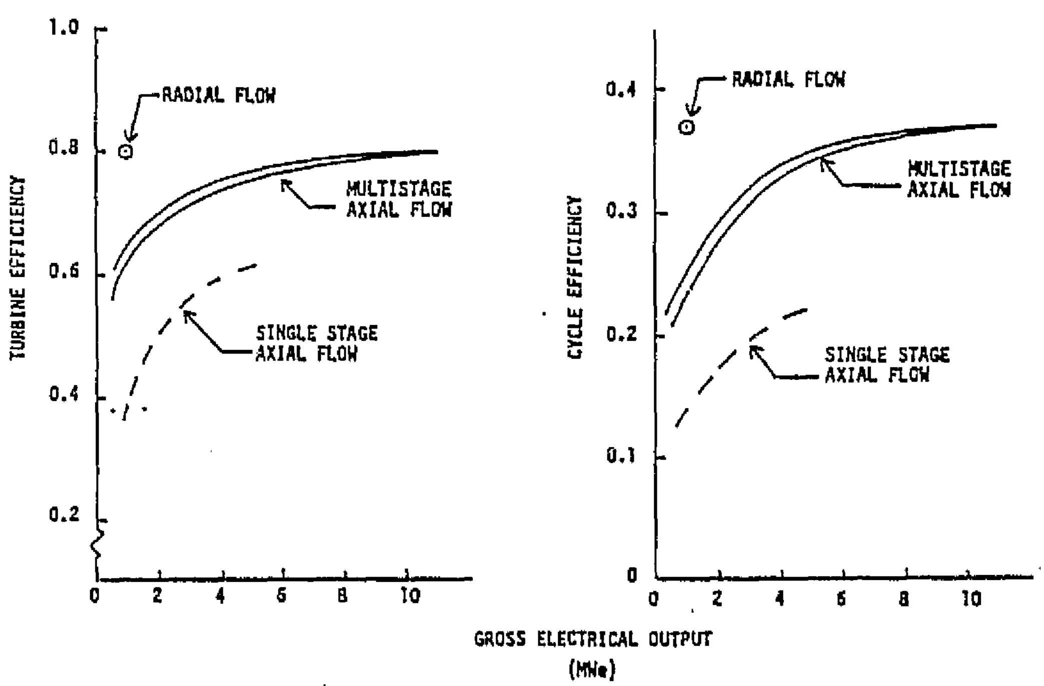
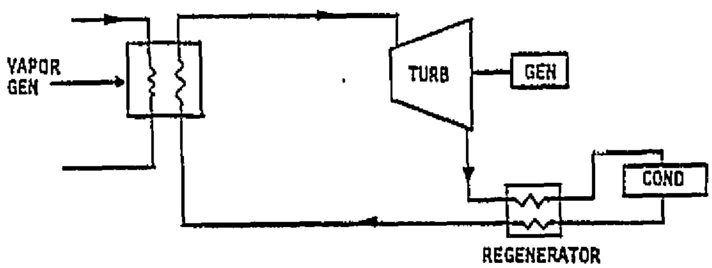


Figure 3.2-9. Impact of Turbine Capacity on Performance

Since the number of heliostats required to power the turbine is inversely related to the cycle efficiency, a high cycle efficiency is essential to minimize the cost of the heliostat field. For this reason, the single stage turbine was discarded from further consideration even though it was the lowest cost of the three turbine concepts. Conversely, the importance of continuing the development of the radial turbine is clearly apparent because of its high cycle efficiency and resulting ability to minimize the cost of the energy collection portion of the system.

3.2.5.2 Organic Rankine Cycles

Organic Rankine cycles were considered because of their high cycle efficiency which can be realized at low power levels. The principal features of an organic Rankine cycle are schematically shown in Figure 3.2-10 along with the characteristics of some of the more common working fluids.



WORKING FLUIDS	TEMP LIMIT	COMMENTS
TOLUENE	385°C (725°F)	• DEVELOPED TECHNOLOGY • OPERATE SUBCRITICAL OR SUPERCRITICAL • SAFETY CONCERN
R-11	200°C (400°F)	• RAPID DECOMPOSITION ABOVE 200°C • NON HAZARDOUS
R-113	215°C (420°F)	• RAPID DECOMPOSITION ABOVE 215°C • LIMITED TO SUBCRITICAL OPERATION • NON HAZARDOUS
FLUORINOL 85	315°C (600°F)	• NON HAZARDOUS (CF_4 EMITS F_2 ON COMBUSTION)

Figure 3.2-10. Organic Rankine Cycle

Because the organic fluids become superheated as they expand through the turbine, all regenerative heating can be accomplished with a regenerator located between the turbine exhaust and the condenser. In addition, the vapor generator need only produce saturated vapor at the turbine inlet thus reducing the complexity of the vapor generator. The low enthalpy change which occurs during the expansion process requires that a substantial flow rate pass through the turbine and the balance of the cycle. This results in substantially larger equipment than would be required for a water/steam system operating at an equivalent pressure. The cost of this equipment tends to offset part of the advantage of the organic Rankine cycle.

The working fluids identified in Figure 3.2-10 represent the most common fluids used for power cycle applications. Because of the desire for high cycle efficiency in order to reduce the costs of the energy collection hardware, the higher temperature fluids which permit high Carnot efficiencies are of primary interest. Toluene, which is currently used in both subcritical and supercritical turbine applications is preferred. The primary concern in using this fluid is safety because of its explosive and toxic nature. Fluorinol 85 was selected as a backup fluid due ot its non-hazardous nature. Cycle efficiency however would be compromised due to its lower peak temperature limit. The refrigerants (R-11 and R-113) were eliminated from consideration due to their low upper-temperature limit.

The Toluene cycle efficiency characteristics as a function of turbine inlet temperature are shown in Figure 3.2-11 based on a variety of data sources. The solid line represents assumed performance levels used for the current system evaluation analysis. The Sunstrand data point assumes a different condenser back pressure from the other data. When back pressure corrections are made, the Sunstrand point moves upward to the line presented by Barber-Nichols. The AiResearch data point assumes a permanent magnet generator which is lower in efficiency than conventional four pole generators but permits a hermetically sealed design which minimizes leakage problems. It is seen that for high temperature subcritical and supercritical applications, cycle efficiencies can be realized which exceed the values for the multistage axial flow steam turbine equipment (1 MWe).

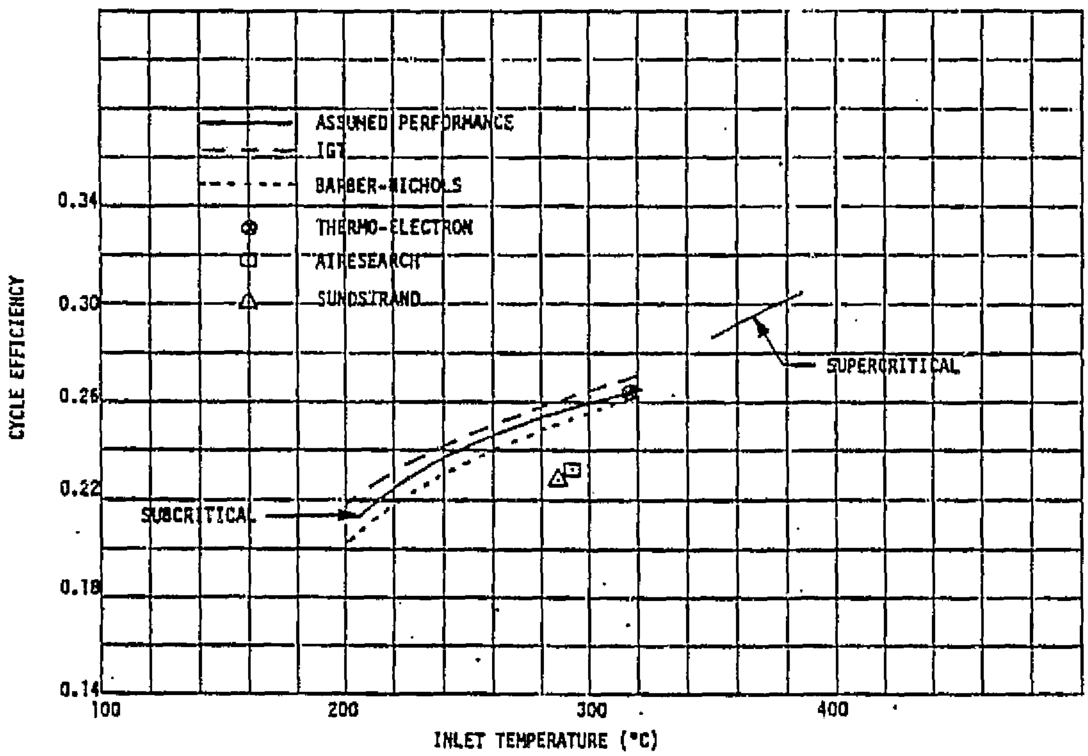


Figure 3.2-11. Performance of Toluene Cycles

3.2.5.3 Reciprocating Steam Engine

The reciprocating steam engine was also considered as a candidate power conversion device during this initial subsystem definition and evaluation task. A uniflow engine design was considered as representing the current state of the art in reciprocating steam engines. The uniflow design accepts steam from one end of the cylinder when the piston is near top dead center and exhausts the expanded steam at the other end of the cylinder when the piston is near bottom dead center. This uniform design prevents cool, partially condensed steam from being forced out of the cylinder through exhaust ports located at the inlet end of the cylinder which cools the cylinder metal surfaces and robs useful thermal energy from the next change of steam.

The principal characteristics of a uniflow engine cycle are shown in Table 3.2-2. As indicated, high expansion efficiencies can be realized with this equipment. In some applications, expansion efficiencies as high

Table 3.2-2

RECIPROCATING STEAM ENGINE CHARACTERISTICS

(1-MWe Rating)

TYPE	UNIFLOW
NOMINAL TEMP LIMIT	343°C (650°F)
NOMINAL PRESSURE LIMIT	3.21 MPa (465 PSIA)
EXHAUST PRESSURE	≥16.9 KPa (5 In Hg)
SPEED	327 RPM
GENERATOR	22-POLE (327 RPM)
NUMBER OF CYLINDERS	3
PISTON SPEED	3.08 M/SEC (14.5 FT/SEC)
EXPANSION EFFICIENCY	80-90%
APPROX ENGINE/GENERATOR COST	$280,000
MANUFACTURER	SKINNER ENGINE COMPANY

as 93 percent are reported. Due to limitations on expansion ratios of about 6.5:1, extreme pressure differentials between the inlet steam and condenser are not warranted. Also, because the engine is a low speed device, it can be coupled directly to a 22-pole engine type generator. This eliminates the need for any gear box.

The steam engine has other design features and limitations which influence its potential attractiveness to the overall system. For example, steam engines are designed for peak performance at 1/2 - 3/4 load as compared to turbine equipment which experiences peak performance at 100 percent load. The steam engine is also capable of continuous operation at 125 percent load as compared to the 110 percent overload capability for a steam turbine. In addition, the steam engine can utilize saturated inlet steam and imposes no limitations as to moisture levels contained in the exhaust steam.

On the other hand, no opportunity exists to extract partially expanded steam for regeneration purposes which is required to produce a high cycle efficiency. In addition, the oil required for piston and crank lubrication may be carried over in the exhaust steam to other system elements. This requires that a filtering station be included in the engine exhaust line, upstream of the condenser. The necessary filtration equipment is available in commercial packages.

3.2.6 Energy Storage

Energy storage can provide two basic functions in the generation of electricity from solar energy. Since there is a potential incompatibility between available solar insolation and electrical power demand, storage is utilized to retain a portion of the excess available energy so that power generation can be continued during periods of little or no insolation. This reserve energy allows the power demand to be synchronized with solar availability (Reference 5). This can be accomplished externally by storing electricity in batteries or internally by storing heat for the subsequent generation of electricity. In the latter case, thermal energy storage also can provide a second function by acting as a buffer to reduce the effects of solar transients on the power conversion subsystem.

The determination of the optimum internal energy storage technique is directly related to the energy transport fluid which is utilized. This is particularly true in the case of sensible heat storage where the transport fluid is used directly for thermal storage. The temperature limit, as well as the cost, physicochemical characteristics, and thermodynamic properties of the fluid have a direct bearing on the selection of the best storage technique. In systems employing latent or thermochemical storage, where energy from the receiver is exchanged across heat transfer surfaces to a secondary storage medium, the heat transport fluid operating temperature is the determining factor. The temperature limit of the transport fluid, for example, will restrict the selection of phase change materials to those with slightly lower melting points. The transport fluids being considered in this evaluation are as follows:

High Temperature Salt (Hitec, etc.)
Syltherm 800
Caloria HT 43
Sodium
Water

The purpose of this initial evaluation phase was to screen applicable energy storage techniques to identify preferred candidates for the various transport fluids being considered. The major techniques considered are as follows:

Thermal Energy Storage

- Sensible heat storage
- Latent heat storage
- Thermochemical storage

External Storage

- Battery storage

The preferred sensible heat storage systems were established for various fluids and compared to preferred latent heat and thermochemical storage systems. Optimum storage subsystems were then chosen for more detailed evaluation to facilitate the selection of preferred system concepts. External storage in batteries was evaluated as the only feasible concept for use with Bryton cycle concepts as described in Section 3.2.4.

3.2.6.1 Requirements

The optimum storage capacity of a system depends on the nature of solar availability and user demand. If the storage capacity is oversized based on peak insolation, it is economically prohibitive since the total amount of stored energy will only occasionally be used. The preliminary comparisons of storage concepts will be based on storing solar insoluation in excess of 800 w/m^2 at equinox noon. The storage capacity (MWH_t) is then a function of the power conversion efficiency.

To avoid losses of available energy, it is important to input and extract heat to and from storage at temperatures equal to or near the receiver outlet temperature. The energy storage operating temperature is therefore a function of the working fluid temperature limit. This limit is normally associated with the temperature at which the fluids undergo significant degradation or excessive vapor pressures occur. The limit for sodium is related more to material aspects concerning the liquid metal loop. These limits along with preliminary storage capacities used to size and compare various storage concepts are shown in Table 3.2-3.

3.2.6.2 Sensible Heat Storage

Sensible heat storage involves the thermal energy imparted to a substance (solid or liquid) as a result of increasing its temperature during charging or the extraction of heat by a similar temperature decrease. The primary advantage of this mode is the possibility of using the same fluid for both energy transport and energy storage. Furthermore, the utilization of an inexpensive solid in direct contact with the receiver fluid offers a desirable alternative. Of equal importance is the simplicity of basic conceptual designs which present little or no development risk. The concepts considered to be the most viable are shown in Figure 3.2-12 and discussed below.

Thermocline - A liquid Thermocline (temperature divide) is established when hot fluid from the receiver is pumped to the top of the storage tank and cold fluid is removed from the bottom. The higher density cool fluid has a tendency to remain below the lower density hot fluid. As the hot fluid is extracted at the top of the tank and pumped to the steam generator during discharge, the thermocline moves up the tank. The opposite effect occurs during energy input. A dual media thermocline can be established by utilizing a rock bed (or other solid material) along with the receiver fluid which fills the void between the solid particals. The solid lends stability to the thermocline and replaces the more expensive liquid. The disadvantage of the system is the possible incompatibility of the solid and the liquid which results in fluid degradation. Since only one tank is required and low cost solids are utilized, this system is the least expensive of the sensible heat concepts.

Table 3.2-3
ENERGY STORAGE REQUIREMENTS

Working Fluid	Temperature Range (°C)	Prime Mover	Temperature Range (°C)	Storage Capacity (MWH_t)
HTS (HITEC)	510	Radial-Steam	288-510	4.48
		Axial-Steam	288-510	5.57
Syltherm	400	Axial-Steam	232-400	7.21
		Organic Rankine Supercritical	232-400	5.08
		Radial-Steam	232-400	5.53
Caloria	316	Axial-Steam	218-316	7.35
		Organic-Rankine Subcritical	218-316	5.94
		Radial-Steam	218-316	6.42
Sodium	650	Radial-Steam	288-510	4.48
		Axial-Steam	288-510	5.57
Water/Steam	315	Radial-Steam		7.12
		Steam Engine		9.12

Two Tank - In the two tank system low temperature fluid from a cold tank flows through the receiver to a hot tank where it is stored until required. During heat extraction the hot fluid flows through the steam generator and is returned to the cold tank. This system is inherently simple and will require almost no development. Both tanks must be sized to contain the entire fluid inventory.

n-Tank - In an attempt to reduce the cost of the two tank system, an alternative system is proposed consisting of n hot tanks plus one cold tank. The n tanks are sized to contain the required liquid inventory with a cold tank large enough to hold the fluid from any of the hot tanks. Based on current tank cost estimates, this system does not appear to be cost effective for small storage volumes, especially when the added cost of piping and valves are considered. The concept is further hampered by the decreased reliability associated with multi-tanks that see dual use service. Rejection of the concept for these reasons is justified.

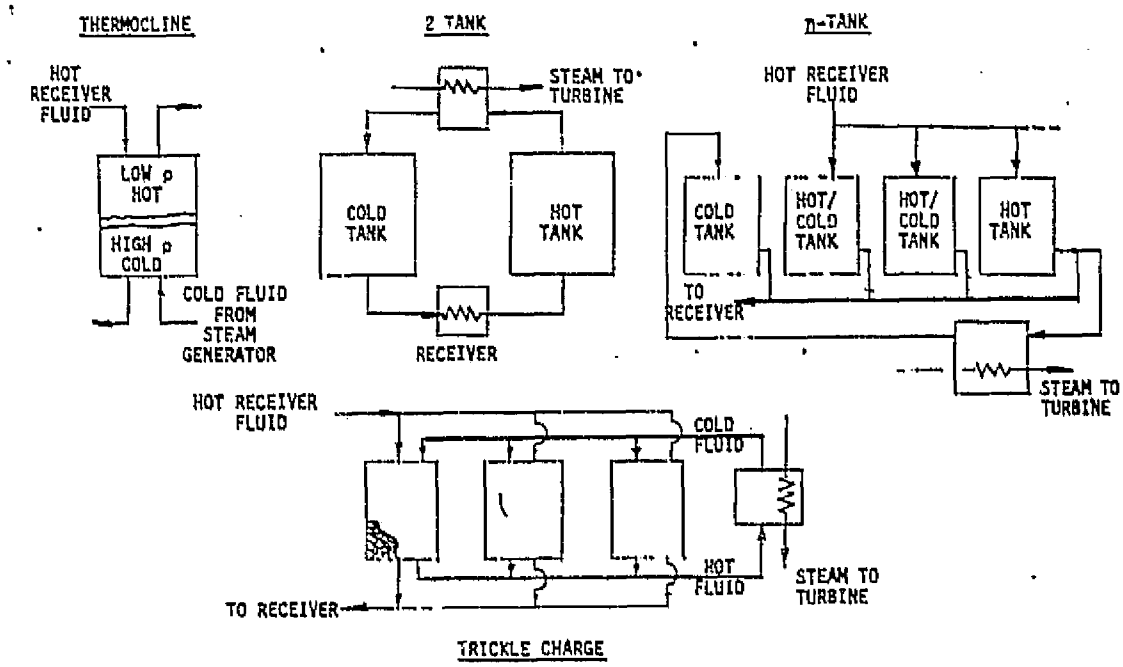


Figure 3.2-12. Sensible Heat Storage Subsystem Concepts

Trickle-Charge - When the use of Syltherm is being considered as a receiver fluid, energy storage subsystems which require a considerable fluid inventory are quite expensive due to the high cost of the fluid. Even dual media system costs are high with void volumes in the range of 25-45% which must be filled with liquid. A proposed solution to this problem involves the trickle charge concept which is schematically represented in Figure 3.2-12 with three tanks. Depending on storage requirements, more tanks may be needed. When excess energy is available, hot receiver fluid is pumped to the first tank, which is filled with a bed of solid particles, and allowed to trickle through the void. The liquid transfers its heat to the solid and when the solid inventory has heated up to the fluid input temperature, the flow is switched to the second tank, and so on. As energy is required, cold fluid from the steam generator is pumped to the top of the first tank and is heated to the required temperature while trickling through the solid particles. Again, flow is transferred to the next tank after the energy is extracted from Tank one. The major problem with this system is the difficulty associated with extracting heat from partially charged tanks. Cool fluid is heated as it flows through the upper region of the bed but cools down again as it flows through the lower uncharged portion of the tank. The disadvantage is partly overcome by the use of multiple tanks, but this becomes expensive and complex as piping and valve requirements increase. The problem of channeling is also anticipated as fluid may establish paths of least resistance and fail to cover the available surface area of the solid particles. This will result in the inefficient extraction of heat.

Pressurized Water - The use of pressurized water storage is only practical for a system in which saturated steam is utilized as an energy transport fluid. Steam from the receiver enters a high pressure storage tank and is condensed with spray water and stored as high temperature saturated water. As energy is required, the hot water is flashed to saturated steam and delivered directly to the power conversion subsystem. The major disadvantage of this system is the high cost of pressurized storage vessels.

The storage concepts are compared in Table 3.2-4. Capital costs shown include preliminary estimates for tanks, media, insolation, site preparation, valves, and installation. In general, the dual media thermocline is the most cost effective but will require some development. The two tank system, although more expensive could be built today with little risk. The trickle charge system has several drawbacks and is only applicable for expensive storage fluids. It is, however, being developed for DOE's total energy system (large scale experiment) at Shenandoah, Georgia.

A summary of the sensible heat concepts is given below:

Preferred Concept: Dual Media Thermocline

- Least expensive
- Materials compatibility require verification
- Storage duration limited

Table 3.2-4

LATENT HEAT STORAGE CONCEPT COMPARISONS (RADIAL STEAM TURBINE)

Fluid (Temperature Range)	Storage Concept	Total Capital Cost	Remarks
Hitec (288-510°C)	Dual Media	173,000	Requires compatibility verification
	2-Tank	241,500	No technology development required
	N-Tank	306,500	Excessive complexity, expensive
	Trickle Charge	238,500	Complex, partially charged tank problem
Syltherm (232-400°C)	Dual media	301,500	Compatibility problems
	Trickle Charge	305,000	Complex, partially charged tank problem
	2-Tank	432,700	Too expensive
Caloria (218-316°C)	Dual media	175,000	Being developed for Barstow 10 MW Plant
	2-Tank	219,800	Too expensive
Sodium (288-510°C)	2-Tank	388,500	Only practical system for sodium
Water/Steam	Pressurized Water	480,000	Excessive costs of pressurized tanks

Backup Concept: Two-Tank:

- No technology development required
- Higher cost than dual media

Rejected Concept: N-Tank

- Too complex and expensive

Trickle charge is only suitable for high cost fluids

- Complex plumbing
- Requires multiple storage tanks
- Difficult to utilize partially charged tanks
- Must provide separate buffer tank
- Channeling may occur

3.2.6.3 Latent Heat Thermal Storage

Latent heat storage involves the change in internal energy of a material as a result of changing phase. The process occurs under essentially isothermal conditions. Because of relatively high heats of fusion, as compared to specific heats, latent heat storage concepts generally exhibit high energy densities resulting in smaller volume requirements. The major difficulty in the development of storage devices has been the low thermal conductivity of the solid phase and the accumulation of solids on heat exchange surfaces.

The selection of preferred latent heat storage subsystems is divided into two areas: (1) identification of optimum phase change materials and (2) identification of optimum heat exchanger concepts.

Phase Change Material Selection

Potential phase change materials are available from several sources (References 6-10). The selection criteria used to reduce candidates to those applicable to the system candidates being evaluated are listed below:

(1) Melting Point between 315°C and 593°C,
(2) Energy Density greater than 525 MJ/M^3,
(3) Relative Cost less than $\$3.50/kWH_t$, and
(4) Secondary Considerations were: Containment, Stability, Toxicity.

It should be noted that the thermal conductivity of the material is also a critical parameter which will dictate the cost of a majority of heat exchanger concepts. However, the relative cost and energy density limits were chosen to eliminate materials which would not be economically competitive with sensible heat concepts on the basis of material and tankage costs alone. With one exception, the resulting list of candidate materials are all eutectic chloride salts as shown in Table 3.2-5. The containment of molten chloride salts in mild steel containers is feasible if all traces of water are removed (Reference 12). This was not considered in the material cost although stainless steel tanks were assumed for all but the $KC\ell/NaC\ell/MgC\ell_2$ salt melting at 385°C. The relative cost shown is the ratio of the latent heat salt plus container costs to the cost of a Hitec/rock thermocline sensible heat storage system (Media + tank) based on 9.5 MWH_t storage capacity.

Table 3.2-5
CANDIDATE PHASE CHANGE MATERIALS

Material	Melting Point (°C)	Cost $/KWH	Capacity(1) MJ/M^3	Comparative Cost(3)
*$NaC\ell$-$KC\ell$-$MgC\ell_2$	385	1.66	525	0.40(2)
$NaC\ell$-$MgC\ell_2$	450	1.47	958	0.73
$KC\ell$-$MgC\ell_2$	470	1.84	854	0.79
$KC\ell$-$CaC\ell_2$-$MgC\ell_2$	487	1.86	864	0.79
*$NaC\ell$-$CaC\ell_2$	500	0.85	607	0.72
$KC\ell$-$NaC\ell$-$BaC\ell_2$	542	3.23	669	0.98
*Na_2CO_3-$NaC\ell$-NaF	575	2.18	1295	0.74

(1) Based on ρ @ 25°C
(2) Latent heat cost based on carbon steel tank
(3) Latent heat media and tank costs divided by sensible HTS dual media and tank costs (neglecting heat exchanger costs)
*Preferred for specified temperature

These ratios show little cost differences among the latent heat storage systems and illustrate a potential cost advantage over sensible heat storage if heat exchanger costs can be kept to a minimum in spite of low thermal conductivities associated with chlorides. The preferred salts in three temperature ranges for applicable working fluids are given below:

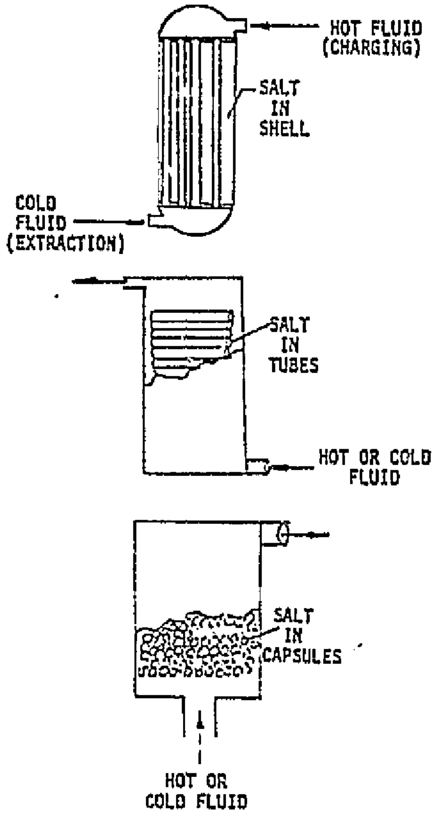
Working Fluid	Salt	Melting Point
Sodium	26 Na_2CO_3-39 $NaC\ell$-35 Naf	576°C
Hitec	33 $NaC\ell$-67 $CaC\ell_2$	500°C
Syltherm	24.5 $NaC\ell$-20.5 $KC\ell$-55 $MgC\ell_2$	385°C

The $NaC\ell$-$CaC\ell_2$ eutectic is preferred over other choices because of its low cost, high temperature capability, and relatively lower hazard rating.

Heat Exchanger Selection

The means by which heat can be input to or extracted from a phase change material can be divided into passive and active heat exchangers, some of which are discussed in References 13 and 14. The passive types are shown in Figure 3.2-13 and are basically represented by the tube/shell configuration (Reference 7). During charging, hot fluid is pumped to the top of the module and flows downward through the tubes. Energy is input to the salt as it melts and expands into the upper expansion volume. To extract heat, cool fluid enters the bottom and flows upward, allowing solidification to proceed in the same direction while molten salt fills the voids that form. This design can be adapted to utilize separate tubing for energy input fluid and energy extraction fluid. The major advantage of this concept, which helps outweigh the high cost of tubing, is the relative state of development. A similar system is currently being developed using a $NaOH/NaNO_3$ mixture (Reference 15).

Alternative concepts considered include macro and microencapsulation (Reference 16). The heat transport fluid flows over the salt filled containers and solidification proceeds inward from the outer radius. These systems appear to offer little advantage over tube/shell designs and will be more expensive unless large scale fabrication is possible.

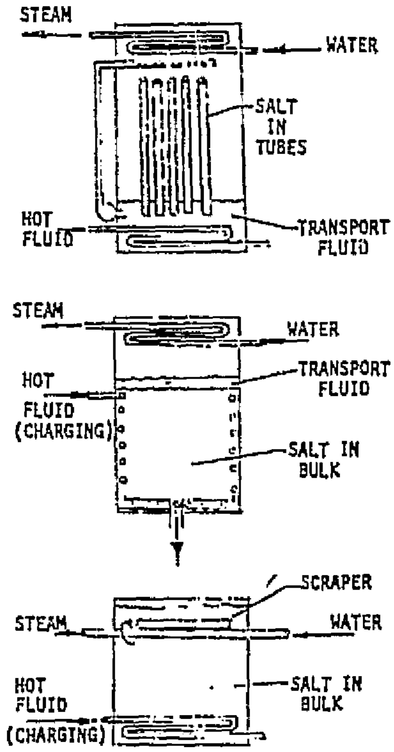


TYPE	REMARKS
TUBE/SHELL	• REQUIRES LEAST DEVELOPMENT • POSSIBILITY OF TWO FLUIDS • EXPENSIVE TUBING
MACRO-ENCAPSULATION	• LITTLE ADVANTAGE OVER TUBE/SHELL • MORE EXPENSIVE
MICRO-ENCAPSULATION	• SHOULD BE EXTREMELY EXPENSIVE

Figure 3.2-13. Passive Latent Heat Storage Configurations

The more advanced class of heat exchange configurations, shown in Figure 3.2-14, are designed for the primary purpose of restricting solidification on heat extraction surfaces. The first is a system proposed by the Naval Research Lab (Reference 17). During charging, the working fluid is pumped through the bottom coils and delivers heat to a (secondary) heat transport fluid which vaporizes and condenses on the containerized phase change salt, causing the salt to melt. To extract heat, water is pumped through the upper coils while transport fluid is sprayed on the hot salt containers. The fluid vaporizes and in turn condenses on the coils to produce steam.

A second concept which has been investigated at McDonnell Douglas is similar to the above configuration with the primary exception that the secondary heat transport fluid is in direct contact with the bulk phase change material. Heat is input to the pipe network along the container sides and bottom causing the salt to melt. The more dense solid phase will remain in the



TYPE	REMARKS
PHASE CHANGE TRANSPORT ENCAPSULATED	DEVELOPMENT RISK EXPENSIVE
PHASE CHANGE TRANSPORT DIRECT CONTACT	COST SAVING POTENTIAL PERFORMANCE INCREASE POTENTIAL DEVELOPMENT RISK
MECHANICAL SCRAPERS	DEVELOPMENT RISK SHAFT POWER PENALTY

Figure 3.2-14. Advanced Latent Heat Storage Configurations

proximity of the bottom coils by forcing the molten salt upward along the sidewalls. To extract energy, the working fluid can be pumped through the upper coils. In the manner of a heat pipe, the transport fluid will continuously deliver heat by vaporizing out of the molten salt and condensing on the upper pipe network while the salt solidifies and proceeds toward the tank bottom. The concept has the potential of reducing costs from the above configuration by eliminating the salt containers and pumping system. Performance will increase as well with direct contact heat transfer. Because of compatibility constraints at high temperatures, the system must probably be limited to sodium (transport fluid) and sodium salts (phase change material).

The final configuration involves the use of rotating scrapers to remove the solid phase from heat extraction pipes (Reference 18). The shaft power penalty and problems with scraper clogging at high heat extraction rates, in addition to the development risk associated with all three advanced concepts, justify the rejection of this concept.

Preferred Latent Heat Storage Subsystems - Based on the above discussion the following subsystems are considered to be the most attractive for latent heat thermal storage:

Operating Temperature, °C	575	500
Working Fluid	Sodium	Hitec
Phase Change Material	Na_2CO_3-NaCℓ-NaF	NaCℓ-$CaCℓ_2$
Heat Exchanger	Phase Change Transport (Direct Contact)	Tube/Shell
Transport Fluid	Na	None required

The tube/shell design is considered to require the least development while the 3 phase heat exchanger, employing a secondary fluid for heat extraction, has the greatest potential for cost reduction and performance increase.

3.2.6.4 Thermochemical Energy Storage

This concept is based on the utilization of reversible chemical reactions and the associated heats of reaction. In the endothermic process, decomposition of a compound yields products which can be separated and stored. The products can then be recombined to regenerate heat at a slightly lower temperature. The major advantages of this concept are the high heats of reaction associated with the breaking of a chemical bond and the reduced heat losses resulting from storage at ambient temperatures.

Reaction Selection and Evaluation

Investigations conducted at the University of Houston (Reference 19) and Rocket Research Corp. (Reference 20), led to the selection of several reactions appropriate for solar thermal storage. The following evaluation criteria were applied to further reduce the candidate reactions for thermochemical energy storage in small power systems:

(1) Heat of reaction greater than 110 KCAL/KG,
(2) Equilibrium Temperature between 315°C and 593°C,
(3) Reversible reaction,

(4) Fast Reaction (Low activation energy),
(5) Products can be readily separated,
(6) Handling of compounds should be simple, and
(7) Compounds commercially available at low cost

The resulting candidate reactions are shown in Table 3.2-6. Two apparently attractive reactions do not involve solids, but the NH_4HSO_4 decomposition requires 900°C for the product separation scheme and the C_6H_{12} reaction involves the expense of pressurized hydrogen storage. Of the remaining candidates, the decomposition of $Ca(OH)_2$ and NH_4Cl are the most desirable. The dehydration of $Ca(OH)_2$ has been studied by Atomics International (Reference 21) and appears suited to the storage subsystem shown in Figure 3.2-15 (Reference 22). Solid $Ca(OH)_2/CaO$ is contained in tubes of a fixed bed reactor. During charging hot fluid flows through the reactor shell generating water vapor which is condensed and stored. Heat is regenerated by vaporizing the stored water with heat diverted from the power cycle for the hydration of CaO in the reactor tubes. The heat of reaction is transferred to the working fluid flowing through the shell.

Close examination of the alternative NH_4Cl decomposition reaction reveals several disadvantages relative to the above concept. This reaction would require a fluidized bed reactor which could be quite small, but the need for storage tanks, heaters, and condensers for each product, HCl and NH_3, is apparent. Containment and a conveyer system for the solid NH_4Cl in addition to product separation equipment would also be required. The $Ca(OH)_2$ subsystem was therefore chosen as the preferred thermochemical concept.

3.2.6.5 Thermal Storage Evaluation

It is important to realize the relative performance of the three modes of thermal energy storage relative to the generation of steam (or organic vapor) to operate a turbine used to produce electricity. This is illustrated in Figure 3.2-16, in which the enthalpy change of a turbine working fluid using sensible heat storage is compared to the relative change using latent or thermochemical heat storage. Assuming that the receiver fluids have equal temperature limits (T_1), it is easily seen that the turbine inlet temperature

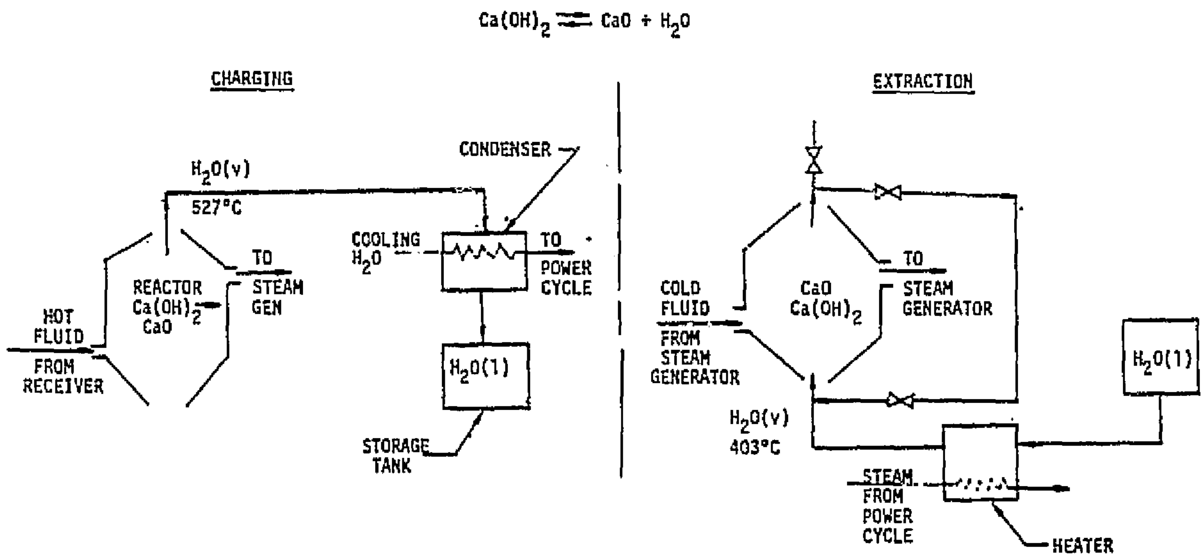
Table 3.2-6

CANDIDATE REACTIONS

		T* (°C)	$\Delta H°_{25°C}/V$ ② (MJ/M³)	REMARKS
1.	$Ca(OH)_2 \rightleftharpoons CaO + H_2O$	479	1970	
2. ①	$NH_4HSO_4 \rightleftharpoons NH_3 + H_2O + SO_3$	467	3100	900° C Required for Separation
3.	$NH_4Br \rightleftharpoons NH_3 + HBr$	404	2680	Less Desirable than NH_4Cl
4.	$MgCO_3 \rightleftharpoons MgO + CO_2$	400	2343	CO_2 Storage less Attractive than H_2O
5.	$NH_4Cl \rightleftharpoons NH_3 + HCl$	345	2390	Complex Storage System
6. ①	$C_6H_{12} \rightleftharpoons C_6H_6 + 3H_2$	295	253	Hazard and Expense for H_2 Storage Low Temperature Capability

① Reactions which do not involve solids

② $\Delta H°$ Reduced by heat of vaporization of liquid product



- FIXED BED REACTOR
- RECEIVER FLUID IN SHELL
- $Ca(OH)_2/CaO$ IN TUBE ANNULUS

Figure 3.2-15. Preferred Thermochemical Storage Subsystem Schematic

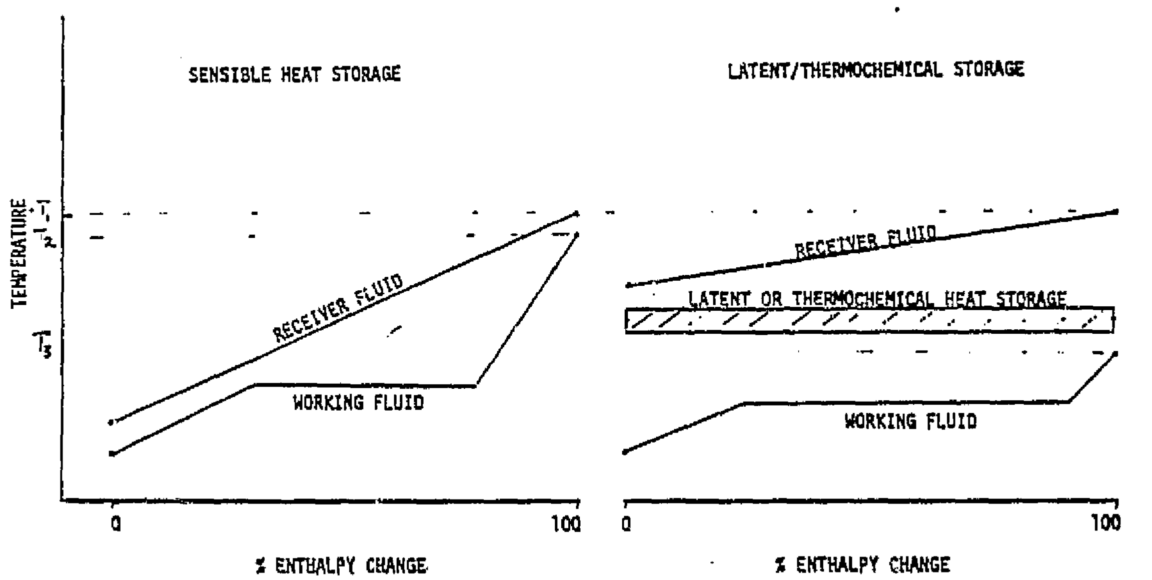


Figure 3.2-16. System Impact of Thermal Storage Concepts

(T_2) in the sensible heat system (and the associated efficiency) is higher than the corresponding temperature (T_3) in the latent/thermochemical heat storage case. This is a result of the isothermal process associated with the phase change of a substance. The constant temperature is shown as a finite band since the temperature at which heat is extracted from a reversible chemical reaction (exothermic) must realistically be lower than the temperature at which heat is input (endothermic). There is also a temperature difference across the metal wall and the solidified layer of salt in most latent heat storage devices. If the latent heat or thermochemical operating temperature is chosen closer to the receiver fluid temperature limit, the receiver fluid would be exercised over a smaller temperature range requiring higher flow rates and a penalty must be paid in pumping power. In essence, the utilization of latent or thermochemical storage will result in:

(1) increased pumping power requirements,
(2) decreased thermal collection efficiency, and
(3) decreased power conversion cycle efficiency.

Based on the storage capacity of 4.4.8 MWH_t for a steam Rankine turbine (radial outflow), major system storage costs were compared for the preferred sensible, latent and thermochemical energy storage system in Table 3.2-7. On the basis of higher costs, development risk, and performance penalties associated with latent and thermochemical energy storage, both concepts have been rejected in favor of sensible heat storage. The results of this preliminary evaluation are summarized below:

Preferred Concept

Sensible Heat Storage

- Little Development Risk
- Low Cost
- Good Performance

Rejected Concepts

Latent Heat Storage

Tube/Shell Configuration

- Extremely High Tubing Costs
- Degradation in Cycle Efficiency

Direct Contact (Transport Fluid) Configuration

- Cost is Competitive
- Requires Undesirable Transport Fluid (Sodium)
- Problems with Subambient Pressure Operation
- Phase II Development Costs Unjustified
- Degradation in Cycle Efficiency

Thermochemical Energy Storage

- Complex and Costly
- Degradation in Cycle Efficiency
- Phase II Development Costs Unjustified

Table 3.2-7

PRIMARY THERMAL STORAGE COST COMPARISONS

Concepts	Cost Elements				
	Tank(s)	Media	Heat Exchanger Tubing	Reactor	Total
SENSIBLE (288-510°C)					
Hitec Dual Media Thermocline	39,500	11,700	---	---	51,200
Hitec Two Tank	51,600	36,400	---	---	88,000
LATENT (500°C)					
$NaC\ell$-$CaC\ell_2$ Tube/Shell	41,000	3,830	76,300	---	125,140
Na_2CO_3-$NaC\ell$-NaF Phase Change Direct Contact	36,000	10,840	37,800	---	84,640
THERMOCHEMICAL (527°C)					
$Ca(OH)_2$ Fixed Bed	7,500	450	---	305,250	313,200

3.2.6.6 External Storage

As explained in Section 3.2.4, the only practical means of storage for the Brayton cycle is external storage. Electrochemical storage involves the use of batteries which convert dc electrical energy to chemical energy during charging and back to electricity during discharge (Reference 23). The battery basically consists of cells which contain a positive and negative electrode separated by an electrolyte. The cells can be connected in series and parallel arrangement to produce a storage system of a desired capacity. The types of batteries being considered are listed below:

State of the Art
- Lead Acid

Advanced Batteries
- Lithium/Metal Sulfide
- Sodium/Sulfur
- Zinc/Chlorine
- Redox System

The Redox System utilizes inorganic salts in aqueous solution stored in large tanks at ambient temperatures (Reference 24). The cathodic and anodic fluids are pumped through their respective electrolyte chambers, which contain inert electrodes separated by a membrane. The system offers the potential for high efficiency but is considered to be in an exploratory stage and development within the 1980's is not expected. This concept was therefore rejected. The remaining battery systems will be treated in detail in Section 3.3.6.7 with respect to cost and performance comparisons.

3.2.7 Candidate System Evaluation

The purpose of this evaluation was to carry out a preliminary screening of the candidate Rankine cycle systems prior to a more detailed subsystem optimization analyses required for final concept selection. The candidate systems for this screening process were developed by synthesizing the previously defined subsystems and appropriate working fluids into meaningful systems which had a state of technology readiness consistent with the program requirement (3-1/2 to 6-1/2 years).

In addition to the issue of technology readiness, the principal screening criteria was commercialization potential as expressed in terms of system cost. The methodology used in this process was aimed at assessing all candidates on a total system basis. In order to ensure favorable treatment of all candidates, preliminary optimization analyses were carried out for each system prior to the comparison. Identical load factors were also established for all systems so that candidates were compared on the basis of equal annual electrical output.

The reference load factor was determined for the central receiver system on the basis of three representative Barstow insolation days while simultaneously satisfying the requirement to produce 1 MWe net on equinox noon at an insolation level of 800 W/m^2. The corresponding central receiver load factor for this condition was calculated to be 0.346 assuming 35 annual cloudy days. The distributed collector systems were sized to match this load factor based on the same insolation model.

In all cases, the energy storage subsystems were sized to accommodate all of the excess energy above that required to produce 1 MWe for the greatest energy collection day based on the reference insolation days considered.

For the central receiver system, the day of greatest surplus energy occurred on equinox. For the energy transport subsystem, that portion between the collector subsystem and the energy storage was sized to accommodate the peak thermal power capability of the collector subsystem while the portion downstream of the energy storage subsystem was sized to be compatible with the 1 MWe plant output requirement. Clearly, these initial approaches to sizing the energy storage and energy transport subsystems must be expanded upon during subsequent design activities. However, for this task, it ensures a sufficient level of consistency necessary to carry out the preliminary screening process.

3.2.7.1 Central Receiver Concepts

The central receiver concepts considered for evaluation were developed by synthesizing candidate subsystems and fluid types into complete systems subject to the sizing constraints defined in the previous section. In addition, consideration was given to the thermodynamic compatibility between collector and power conversion subsystem fluids.

The candidate fluids and subsystems considered in the analysis include:

- Receiver/Storage Fluids
  - Heat transfer salt
  - Sodium
  - Syltherm
  - Caloria HT-43
  - Water/steam
- Power Conversion Subsystem
  - Radial steam turbine
  - Axial multistage steam turbine
  - Organic Rankine turbine
  - Reciprocating steam engine
- Energy Storage Subsystem
  - Two-tank
  - Dual media thermocline
  - Dual media trickle charge
  - Pressurized water
- Collector Subsystem
  - (Concentrator)
    - Heliostats
  - (Receiver)
    - Planer absorber/guyed tower
    - Two-zone absorber/guyed tower

The temperature limits for the receiver/storage fluids impose restrictions on the selection of the power conversion subsystem equipment and their corresponding operating efficiencies. Table 3.2-8 lists the combinations of fluid and power conversion equipment considered along with operating temperature and cycle efficiency data. The energy storage and receiver approaches used for each of these combinations were selected to minimize cost subject to performance restrictions imposed on the designs. For example, two-tank storage approaches were ignored for Syltherm systems because of the high cost

Table 3.2-8

CANDIDATE RANKINE CYCLE CENTRAL RECEIVER SYSTEMS

POWER CONVERSION SUBSYSTEM				RECEIVER			ENERGY STORAGE
FLUID	TEMP/PRESS (°C/KPa)	PRIME MOVER	CYCLE EFF.	FLUID	T_{OUT} (°C)	T_{IN} (°C)	TYPE
STEAM	482/10343	RAD. TURB	34.6	HTS*	510	288	• DUAL MEDIA THERMOCLINE
STEAM	482/6205	AXIAL TURB	27.8	HTS*	510	288	• TWO-TANK (LIQUID)
TOLUENE	385/4827	ORG. RANKINE	30.5	SYLTHERM	400	232	• TRICKLE CHARGE
STEAM	371/4137	AXIAL TURB	21.5	SYLTHERM	400	232	• DUAL MEDIA THERMOCLINE
STEAM	371/3172	RAD. TURB	28.0	SYLTHERM	400	232	
TOLUENE	302/1379	ORG. RANKINE	26.1	CALORIA	315	218	• DUAL MEDIA THERMOCLINE
STEAM	288/4137	AXIAL TURB	21.1	CALORIA	315	218	• TWO-TANK (LIQUID)
STEAM	288/1379	RAD. TURB	24.2	CALORIA	315	218	• TRICKLE CHARGE
STEAM	482/10343	RAD. TURB	34.6	SODIUM	510	288	• TWO-TANK (LIQUID)
STEAM	232/2916	RAD. TURB	21.8	WATER/STEAM	260	110	• PRESSURIZED WATER
STEAM	232/2916	RECIPROCATING ENGINE	17.0	WATER/STEAM	260	110	

*HTS - HEAT TRANSFER SALT (HITEC PROPERTIES USED FOR THE CURRENT ANALYSIS) associated with the fluid inventory in comparison to the thermocline and trickle charge designs. Planar absorbers were assumed for the receiver in systems where the receiver fluid could accommodate the incident heat flux without exceeding the film temperature limit. In other cases where the heat transfer potential of the fluid limited the receiver design, a two-zone receiver design was selected.

The collector subsystem performance characteristics for a system utilizing a heat transfer salt are shown in Figure 3.2-17 for the three reference insolation days. The subsystem performance is expressed on a per m^2 basis which is a measure of heliostat performance as influenced by receiver efficiency. For reference, the turbine demand line required to produce 1 MWe is also shown. The calculated load factor based on these energy collection curves and an assumed 35 cloudy days is 0.346. All other system comparisons were made on a common load factor basis.

The collector field definition used in conjunction with each of these systems was arrived at by scaling a reference collector field in direct proportion to the peak power requirement. The reference collector field was defined as

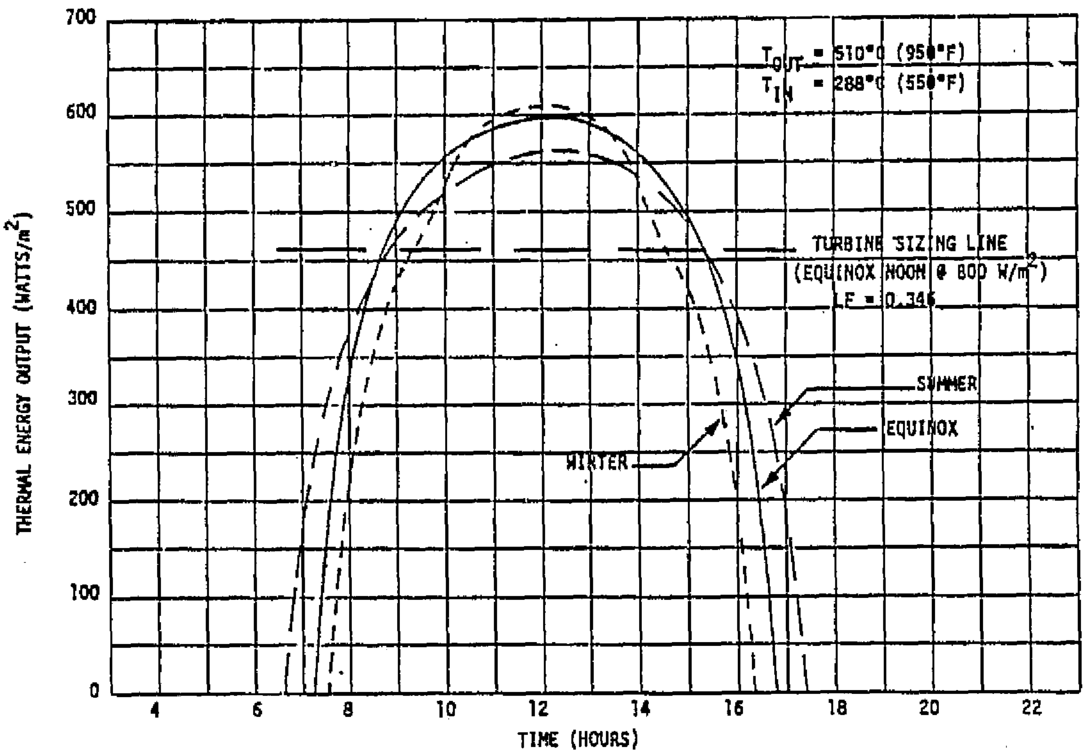


Figure 3.2-17. Central Receiver Performance for Reference Barstow Data

a result of an optimization procedure which included optical, thermal, and economic considerations. Specific factors included in each were:

OPTICAL

- Heliostat reflectivity
- Field cosine
- Blocking and shadowing
- Beam interception
- Atmospheric attenuation
- Tower shadow

THERMAL

- Receiver thermal performance

COST

- Heliostats
- Wiring
- Land
- Receiver
- Tower

The resulting number of heliostats required for the various candidate systems are shown in Figure 3.2-18 as a function of the product of receiver and conversion cycle efficiency. The dramatic increase in heliostat requirement for low efficiency systems results in a high cost collector subsystem which can dominate other system cost elements.

The characteristics of the receiver in terms of required aperture area and thermal efficiency are shown in Figure 3.2-19. The increase in area for lower cycle efficiencies reflect the larger quantity of thermal power that must be collected to power a 1 MWe conversion cycle. The increase in area in moving to Syltherm and Caloria from Hitec or sodium also reflects the heat transfer limitations for each of these fluids.

The thermal efficiency data includes both the surface area and temperature effects of the receiver. As indicated, the larger low-temperature receivers also have the lowest thermal efficiency. The thermal efficiency however is

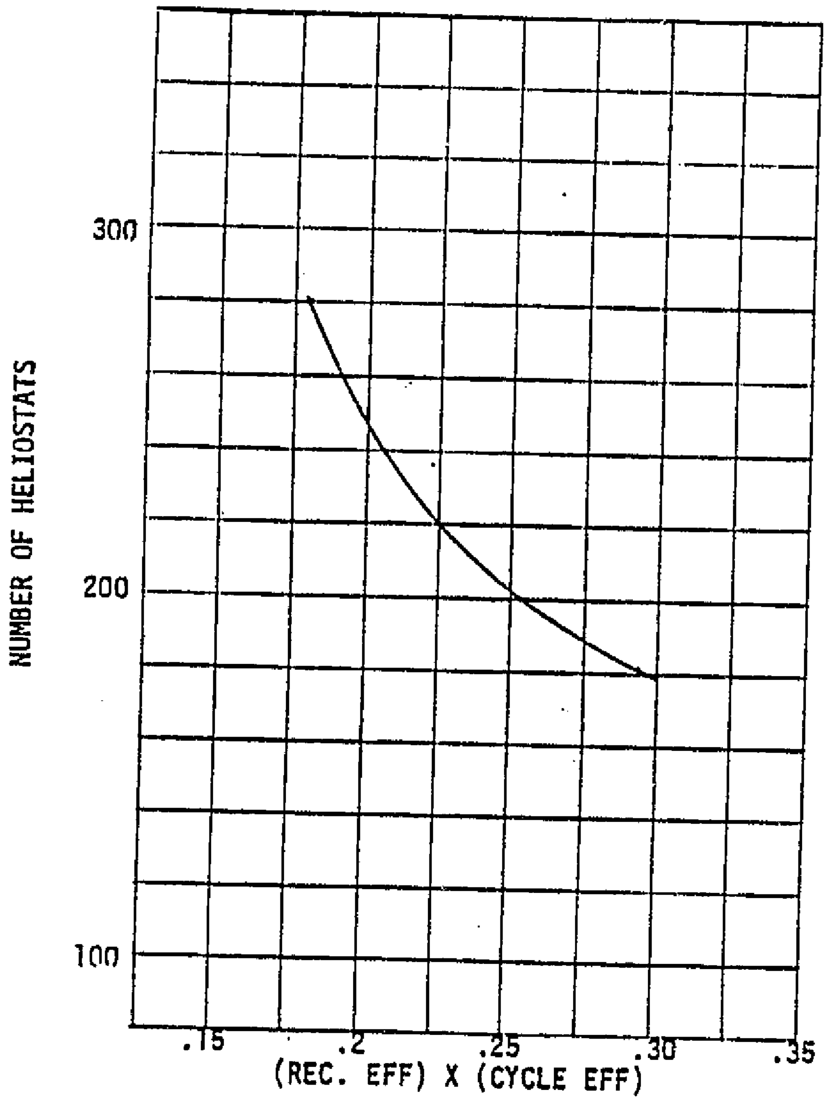


Figure 3.2-18. Heliostat Requirements

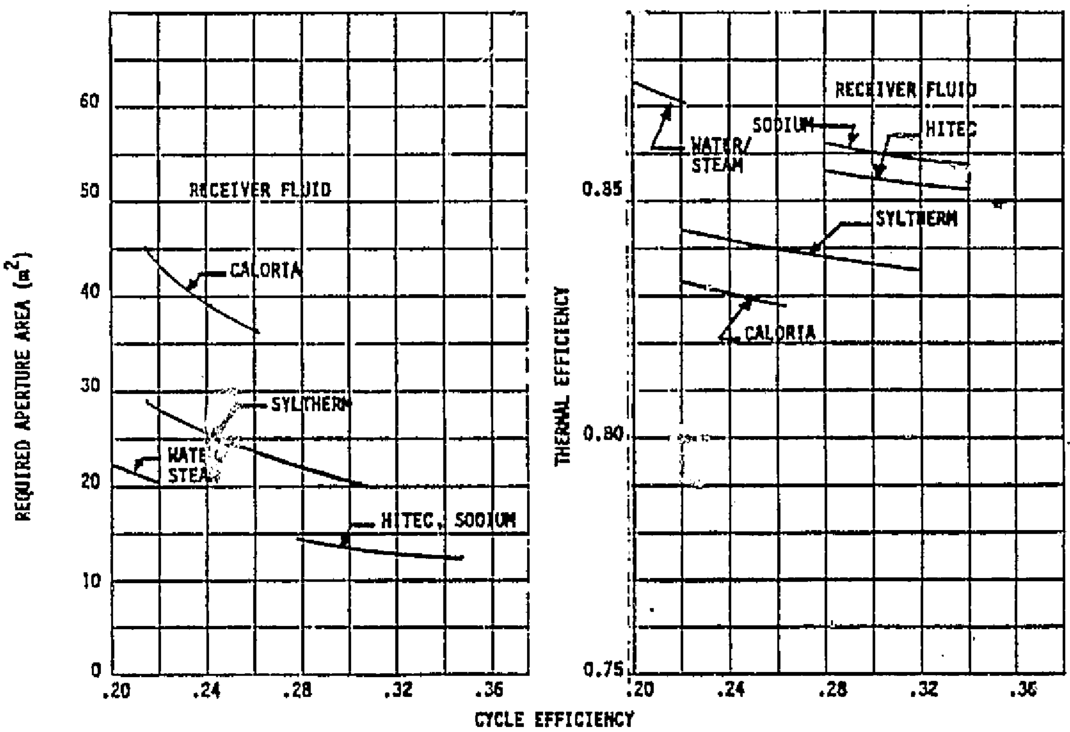


Figure 3.2-19. Aperture Receiver Characteristics

not significantly better for the smaller receivers due to their higher surface operating temperatures. Although the thermal efficiency values indicated in the figure were used in the analysis, it is seen that a value of 0.85 would be reasonable approximation for any of the candidate receivers.

Receiver tower design and cost implications must also be included in any comparative system analysis. The two principal factors influencing tower cost are height and supported weight. The influence of cycle efficiency on each of these parameters is shown in Figure 3.2-20. The weight advantage for sodium and Hitec receivers is clearly apparent, relative to the larger Syltherm and Caloria designs and the high pressure water/steam receiver.

The impact of the design and performance characteristics defined above on the system evaluation is contained in their influence on overall system cost. The heliostat cost estimates, which represent the most significant cost portion of the system, are summarized in Table 3.2-9. The entries for the "Barstow" heliostat correspond to work that was carried out prior to August

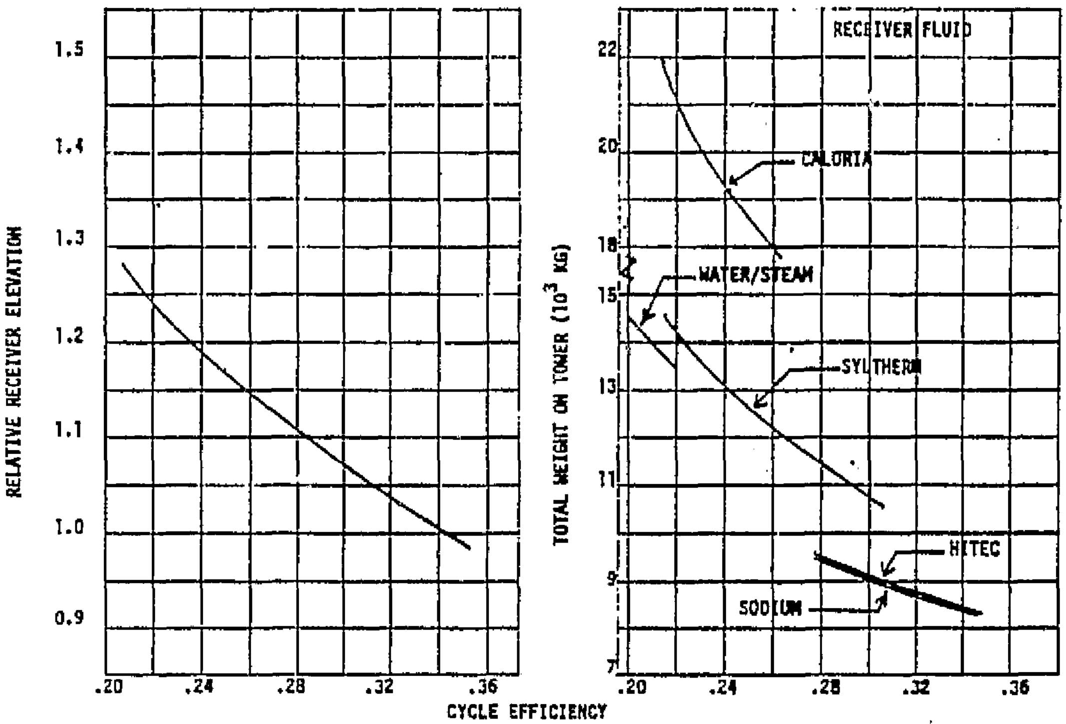


Figure 3.2-20. Tower Requirements

Table 3.2-9
HELIOSTAT COST ESTIMATES

Configuration	Cost	Comments
Barstow (38 m^2)		
[Aug '77 Design]		
1st	$250/$m^2$	Use for Phase III estimates
Nth	$101.5/$m^2$	Assuming no design change
Advanced Design (49 m^2)		
[DOE Cost Reduction Program]	$73-84/$m^2$	Middle 1980's (produced at 25,000 per year)
DOE Goal	$65.5/$m^2$ ($72/$m^2$ - Reflect)	Could be achieved in late 1980's

1977 as part of the DOE 10 MWe Pilot Plant Design Contract (Reference 25). The two cost values shown correspond to a first unit and a Nth unit assuming no changes to the original design. The numbers shown for the "advanced design" heliostat are for heliostats which would be available during the late 1980's assuming an annual production rate of 25,000 units per year. The lower cost numbers relative to the "Barstow" design reflect a series of cost reduction design changes implemented into the basic heliostat design. This work was also carried out under DOE Contract (Reference 26). For comparison purposes, the DOE goal of $\$72/m^2$ - reflectivity is also shown. This target value corresponds to a cost of $\$65.5/m^2$ assuming an average reflectivity value of 0.91.

The heliostat cost estimates presented in this table reflect the results of extensive design evolution carried out by MDAC since 1973. It includes the previously referenced DOE contracts as augmented by in-house funding plus work currently underway in support of A. D. Little under the sponsorship of EPRI (Reference 27). In all cases, the cost estimates were based on a detailed "bottoms-up" approach which considered individual parts as well as the impact of production and site assembly facilities. These analyses were supplemented by detailed transportation and installation studies. In addition, the estimates were based on vendor quotations from a variety of sources.

The costs associated with the other central receiver system components were arrived at by scaling a reference design to the appropriate power level or energy capacity for the specific system design under consideration. From the receiver standpoint, the lowest cost configuration was selected which is compatible with heat transfer limitations. The tower costs were based on a guyed reference tower design which was adjusted to reflect height and receiver weight.

The energy transport subsystem was assumed to bring the collection fluid to a central point in the plant. The cost estimates reflected consideration of flow control and circulation equipment, pipe materials, supports, and insulation. The energy (thermal) storage subsystem was sized to accommodate thermal energy in excess of the turbine demand required to produce the 1 MWe design output. For each case, a minimum cost approach was selected for the system comparison. Details regarding the energy storage characteristics for each approach along with their relative costs were given in Section 3.2.6. In all cases, the energy storage costs reflect any required ancillary equipment.

The resulting cost estimates for each of the candidate systems are shown in a "layer cake" format in Figures 3.2-21 and 3.2-22. The results are for two heliostat cost models: (1) Nth Barstow heliostat, and (2) the average value for the "late 1980's" costs given in Table 3.2-9. These results ignore costs associated with plant control, site improvement, architect and engineering services, distributables, initial plant spares, and plant startup. These values are assumed to be essentially identical for each of the systems considered.

These results show the superiority of the radial turbine system utilizing a high temperature heat transfer salt (Hitec) as the receiver and storage fluid. A cost superiority exists in the receiver/tower and heliostat entries. The energy transport and energy storage entries are equal to or larger than some of the lower temperature alternatives because of the use of stainless steel which is required to accommodate the higher temperatures. The axial turbine case utilizing Hitec as a receiver fluid is more expensive than the radial turbine alternative because of the lower power conversion cycle efficiency which increases the thermal requirement for each of the subsystems. In addition, the axial turbine is significantly more expensive than the estimated costs for the radial turbine.

The sodium receiver case with a radial turbine shown in Figure 3.2-21 is significantly more costly than the Hitec receiver/radial turbine. The principal differences occur in the energy storage subsystem which requires multiple tanks of pure sodium and in the energy transport subsystem which requires additional sodium-specific hardware.

In the cases of a Syltherm or Caloria receiver, the radial turbine and the organic Rankine cycle result in similar costs, while the axial turbine case results in a somewhat higher cost. These values most directly reflect the differences in cycle efficiency.

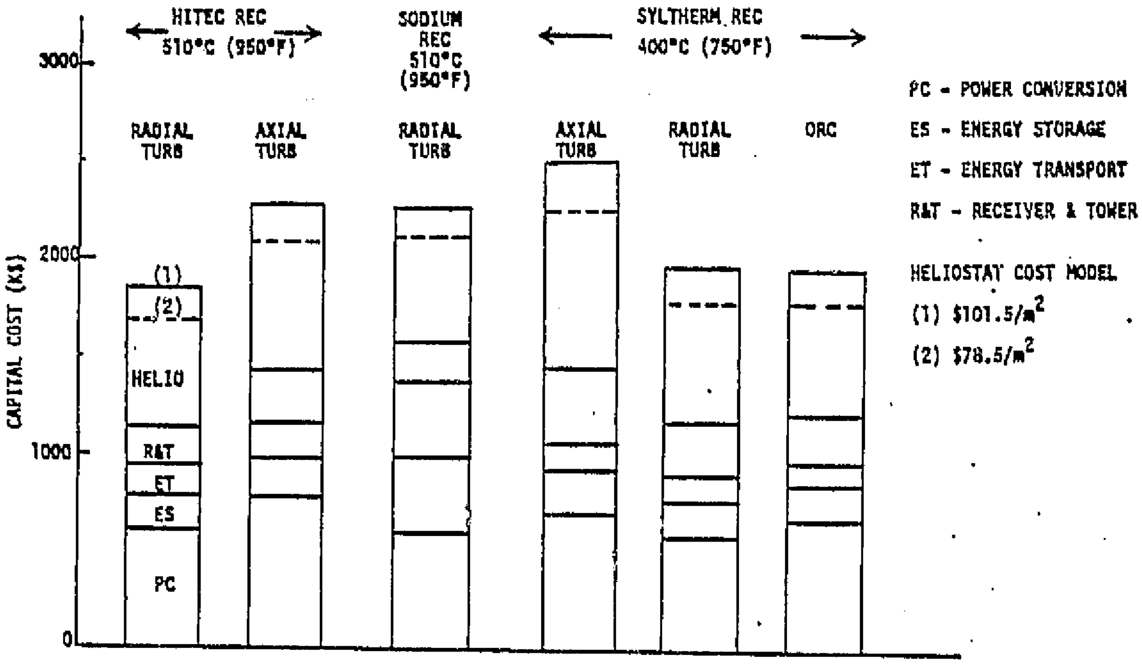


Figure 3.2-21. Comparison of Central Receiver System Concepts — 1

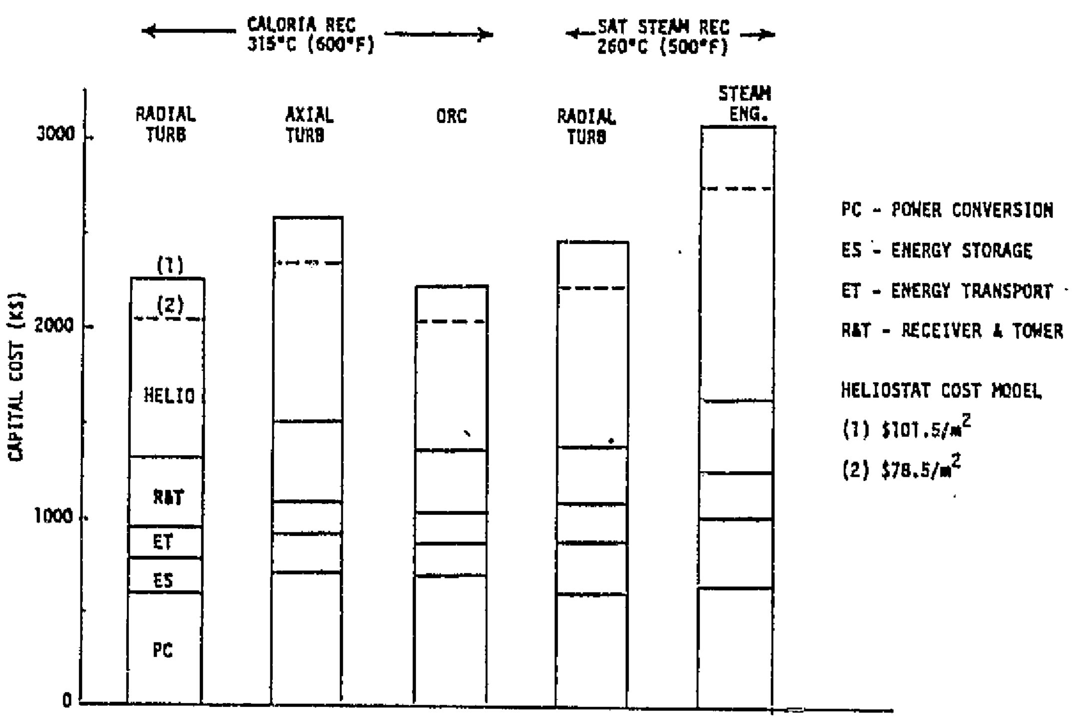


Figure 3.2-22. Comparison of Central Receiver System Concepts — 2

The saturated steam receiver cases show a somewhat higher cost for the radial turbine and a significantly higher cost for the reciprocating steam engine. These results are dictated by the corresponding cycle efficiencies.

In all cases, the collector subsystem including the heliostats, receiver, and tower represents approximately half the indicated cost. Since these elements vary inversely with the efficiency of the downstream elements, it is important to design systems with high subsystem efficiencies unless the subsystem costs become prohibitively high.

3.2.7.2 Line Focusing Distributed Collector Concepts

This section of the report describes the evaluations performed on various line focusing distributed collector designs to determine their performance, system costs, and suitability for the small power system application. Two generic types of single-axis tracking collector designs were evaluated: (1) parabolic trough and (2) segmented mirror. Detailed characteristics of each collector type considered are given in Table 3.2-10.

Parabolic Trough

Parabolic trough concentrators which redirect the incident sunlight onto a linear absorber are the most highly developed of all solar concentrator concepts and are commercially manufactured and marketed by a variety of companies. Performance tests carried out by Sandia Laboratories - Albuquerque in the Collector Module Test Facility have indicated that the Hexcel collector exhibits superior performance over other parabolic trough collectors previously tested (Reference 28). As a result, the characteristics of the Hexcel collector will be used as the basis of evaluation for the parabolic trough collector concept in conjunction with the 1 MWe system power requirement.

This collector is fabricated from treated aluminum honeycomb with aluminum skins. The reflecting surface was an aluminized second-surface acrylic film, FEK 163. Reflectivity was estimated to be 0.86 (clean, unweathered condition). The outer surface of the steel absorber was plated with a selective black chrome to enhance solar radiation absorption and reduce thermal

Table 3.2-10

CANDIDATE DISTRIBUTED COLLECTOR DESCRIPTION

COLLECTOR	PARABOLIC TROUGH	PARABOLIC TROUGH	SEGMENTED MIRROR	SEGMENTED MIRROR
STATUS	PRODUCTION	IDEALIZED	PRODUCTION	BREADBOARD DEVELOPMENT
COMPANY	HEXCEL	-	SUNTEC (SHELDAHL)	ITEK
ORIENTATION	E-W	E-W	E-W, POLAR TILT	E-W, POLAR TILT
TRACKING	1-AXIS, N-S	1-AXIS, N-S	1-AXIS, N-S	1-AXIS, N-S
SIZE	2.8 m BY 6.0 m	2.8 m BY 6.0 m	3.0 m BY 6.1 m 10 MIRROR SEGMENTS	2.3 m BY 6.3 m 7 MIRROR SEGMENTS
REFLECTOR	SECOND SURFACE ALUMINIZED ACRYLIC FEK 163 (0.86)	SILVERED GLASS (0.88)	SILVERED GLASS (0.88)	SILVERED GLASS (0.88)
RECEIVER TYPE	GLASS JACKET	GLASS JACKET	PLANAR	INSULATED TUBE
ABSORBER COATING (ABSORPTIVITY)	BLACK CHROME (0.89)	BLACK CHROME (0.95)	BLACK CHROME (0.89)	PYROMARK PAINT (0.95)
GLAZING (TRANS-MISSION)	PYREX TUBE (0.92)	PYREX TUBE (0.92)	LUSTERGLASS (0.92)	PYREX TUBE (0.92)
INTERCEPT FACTOR	0.96	0.96	0.91	0.92
ABSORBER BLOCKING	0.97	0.97	0.93	0.93
OPTICAL EFFICIENCY	0.6[illegible]	0.72	0.61	0.66

radiation losses. Measurements were made prior to thermal testing of the Hexcel collector to determine the solar spectrum absorptance and emittance of the black chrome absorber tube. The average value of the absorptance was 0.89 which was less than the normal as plated absorptance of 0.95. After thermal testing, the absorptance had degraded to an average value of 0.86.

To further reduce thermal losses from the absorber tube, a half-cylinder of pyrex glass was fitted over the tube on the radiation absorbing side. The back half of the absorber tube was covered with a double layer metal shield. Glazing transmissivity, absorber blocking, and receiver intercept factors were estimated to be 0.92, 0.97, and 0.96, respectively. Thus, the optical efficiency for the Hexcel test collector was 0.86 x 0.89 x 0.92 x 0.97 x 0.96 = 0.66.

The collector was tested at receiver outlet temperatures in excess of 300°C. The instantaneous peak noon efficiency (adjusted to an insolation of 1,000 watts/m^2) obtained in the tests is shown in Figure 3.2-23.

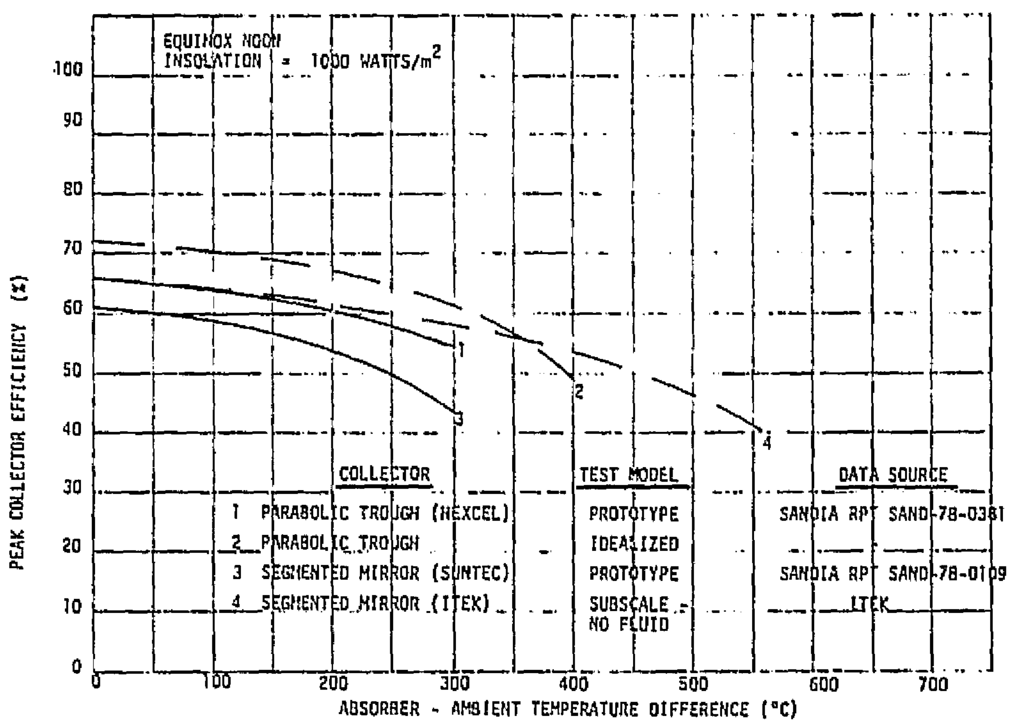


Figure 3.2-23. Distributed Collector Efficiencies

Material improvements have the potential of increasing the reflectivity to 0.88 (weathered including dust buildup degradation) and absorptivity to 0.95. Small specimens of thin- and sagged-silvered glass have demonstrated superior reflectivity and environmental protection. Further research and development of black chrome coatings can be expected to improve the receiver thermal aging characteristics. With these projected improvements, an idealized parabolic trough can be defined (Table 3.2-10) which will have an optical factor of 0.72. Assuming the identical receiver thermal characteristics as the Hexcel collector, the instantaneous peak noon efficiency of the idealized parabolic collector is also shown in Figure 3.2-23.

An important test of a one-axis tracking collector's efficiency is not only the instantaneous efficiency at solar noon, but also the all day efficiency curve. In addition to the conventional incidence (cosine) loss, a further reduction in efficiency of the one-axis tracking parabolic trough occurs during the day as a result of shadowing obstructions due to structural elements and end losses due to reflected light rays that either impact the trough end or miss the absorber tube. Figure 3-2.24 illustrates the measured Hexcel collector performance throughout the day. Based on experimental data for the University of Minnesota/Honeywell parabolic trough (Reference 29), the reduction in optical factor during the day was accounted for by the correction factor: $F(\alpha) = (1 - 0.23 \tan \alpha) \cos\alpha$, where α is the sun-trough angle. This correlation was used in previous MDAC solar energy studies (References 30 and 31) and provides a good fit to the Hexcel data except in the late afternoon when the actual efficiencies decrease at a faster rate. Less end loss would occur when similar collector modules are placed in long rows in a typical collector field, so the all day correlation previously used seems adequate to describe the daily efficiency.

Segmented Mirror

In this collector concept, segmented linear reflectors having a curved surface track the sun to concentrate the sun's rays on a fixed linear absorber. A design of this type developed by Suntec has been installed and tested in the Sandia test facility. The Suntec SLATS collector consists of 10 reflectors per segment with each reflector 0.3m wide and 6.1m long. The silvered glass reflectors are concave arcs with a radius of 6.6m. Receiver design consists

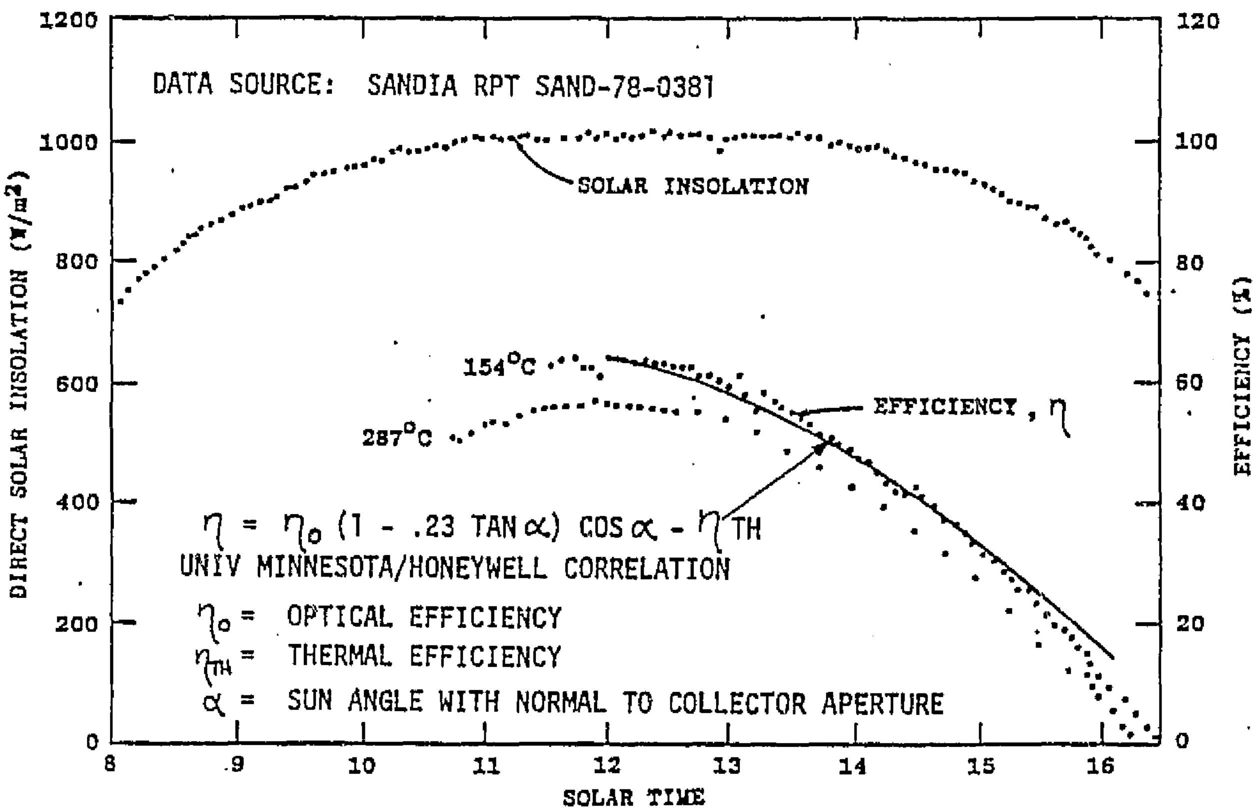


Figure 3.2-24. Hexcell All Day Efficiency

of two parallel lengths of 0.038 meter diameter carbon steel tubing with a black chrome selective surface.

Receiver intercept losses for the segmented mirror concept are greater than a parabolic trough of equivalent aperture due to the increased distance from the rim to the receiver and the greater angle of incidence to the collector aperture. Test results (Reference 32) of the SLATS collector module are shown in Figure 3.2-23 as a function of the absorber-ambient temperature difference. The optical efficiency derived from the Sandia tests is 0.61.

Another version of the segmented mirror design is the Itek solar collector. This design has seven reflector segments 2.3m wide by 6.3m long. The Itek receiver design is conventional except for a deep mirror-lined cavity for directing the energy to the absorber tube and glass louvers to minimize convection. This receiver design has the potential of achieving a high receiver efficiency. Efficiency data calculated from measurements made on Itek's one square meter breadboard demonstration collector are shown in Figure 3.2-25. To be consistent with an assumed weathered reflectivity of 0.88 for silvered glass, the Itek efficiency is characterized by the lower curve in Figure 3.2-25. The optical efficiency of 0.656 consists of the elements shown in Table 3.2-10. For purposes of comparison, the assumed Itek segmented mirror instantaneous performance is also shown in Figure 3.2-23. The segmented mirror daily losses is presented in Figure 3.2-26 in terms of an effective collector area (Reference 33). Daily losses include: (1) cosine, (2) absorber shadowing, (3) mutual shadowing of mirrors, and (4) end effects.

Performance Summary

Experimental data on the candidate distributed collectors have been primarily obtained with an East-West horizontal orientation. A North-South oriented

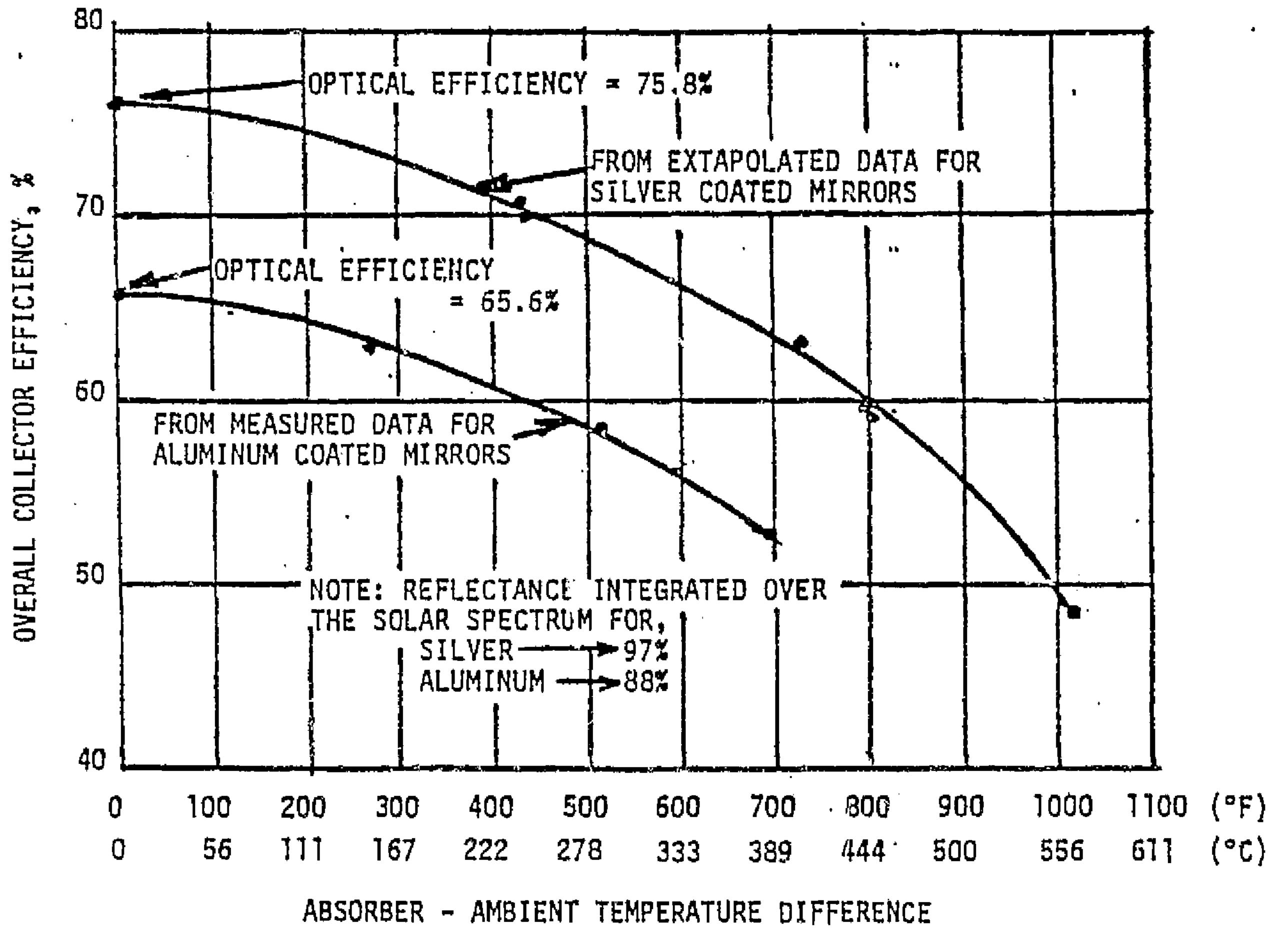


Figure 3.2-25. ITEK Breadboard Collector Performance

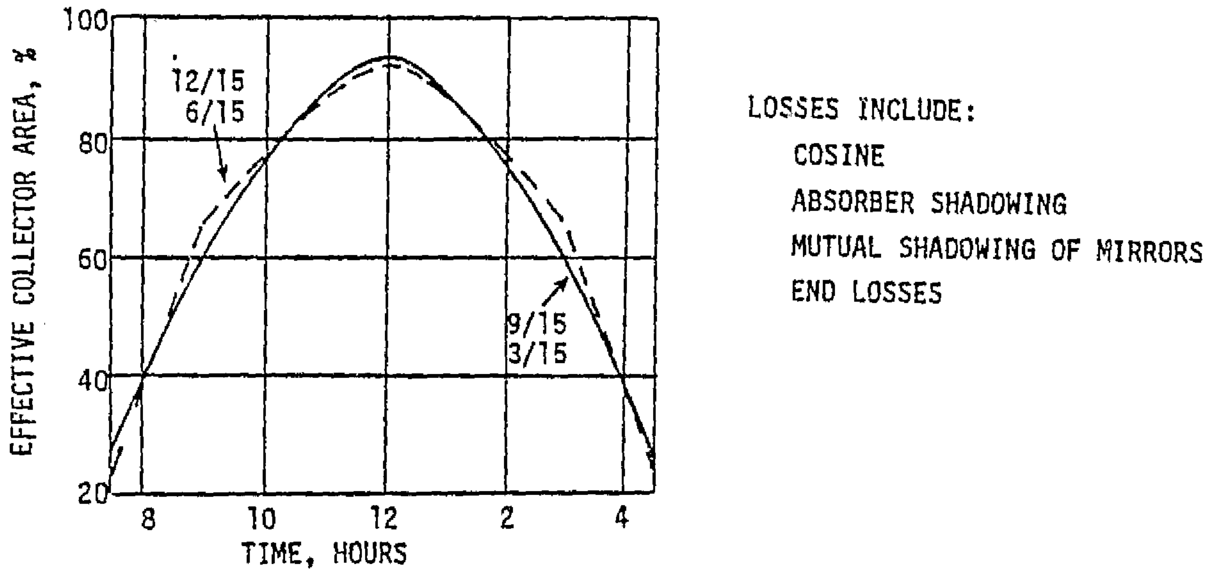


FIGURE FROM PROC. ERDA CONF. ON CONCENTRATING SOLAR COLLECTORS
PAGE 2-51, SEPT., 1977

Figure 3.2-26. Segmented Mirror Daily Losses

field produces output which peaks strongly in the summer and falls off in the winter whereas an East-West oriented system collects reasonably constant quantities of solar energy throughout the year. For the segmented mirror collectors, the plane of the collector is tipped at an angle equal to the local latitude to improve performance.

Collector spacing was determined by the criteria of no shading at 10 AM solar time during the winter solstice. Previous MDAC studies have shown that this criteria generally results in maximizing the energy collected/aperture area. At a latitude of 35°, the resultant ground cover ratio or packing factor is 0.46. At lower packing densities, the energy/aperture area decreases slightly because the thermal losses from the longer interconnecting plumbing is greater than the additional energy collected from less shading. At larger packing densities from the nominal, shading losses dominate.

The choice of collection temperature is dependent on two opposing effects: high temperatures give higher power conversion efficiencies, but lower the annual collector field efficiency; conversely, lower temperatures reverse this trend. System efficiency is obtained by multiplying the two subsystem efficiencies; the optimum collection temperature produces a maximum in the system efficiency curve. Annual collector field efficiency was computed by calculating the hour-by-hour performance for a typical week in each of the four seasons. For this temperature optimization study, hourly direct insolation data from Phoenix, Arizona (SOLMET data tape - 1962) was utilized. As reported in Reference 30, the typical weeks were selected based on satisfying long-term insolation patterns for the region of interest.

Annual collector efficiencies as a function of average collector fluid temperature are shown in Figures 3.2-27 and 3.2-28. The fluid temperature is the average of the field inlet and outlet temperatures. Distributed collector efficiency was assumed to be independent of the heat transport fluid. A steam Rankine cycle with a radial outflow turbine (η = 0.8) was assumed for the power conversion subsystem. As can be seen in these figures, the combination of increasing collector performance with decreasing cycle performance with reduced temperature creates an optimum point for both the parabolic troughs and segmented mirror concepts.

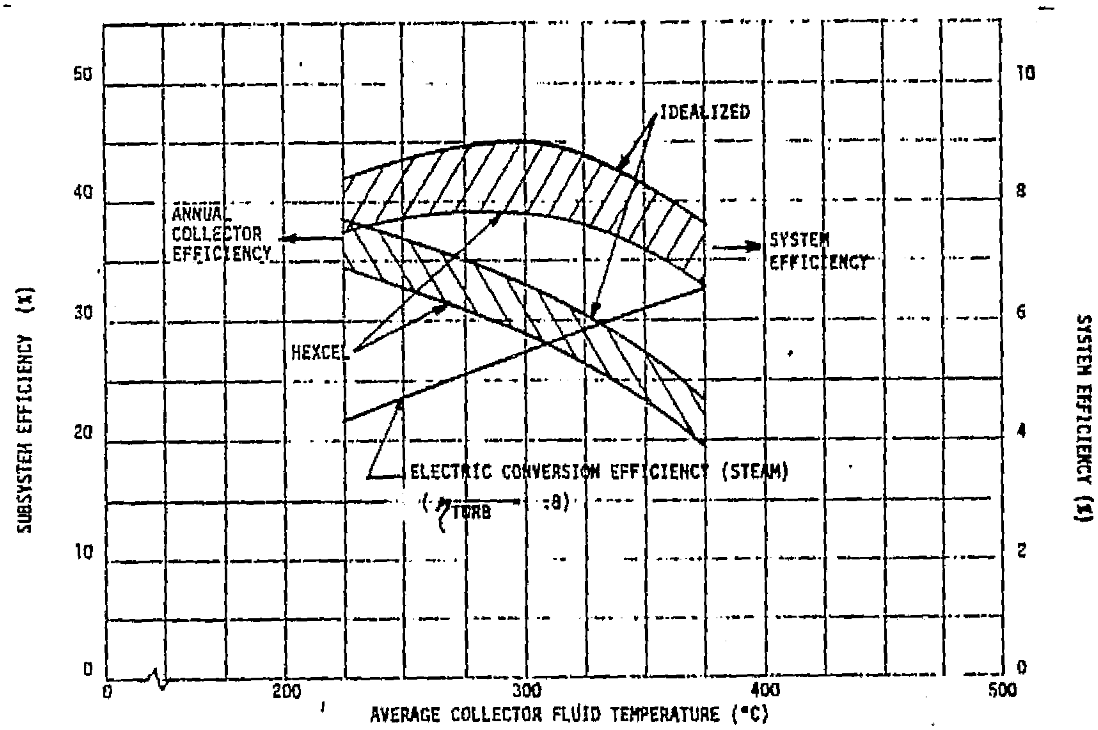


Figure 3.2-27. Parabolic Trough Collection Temperature Optimization

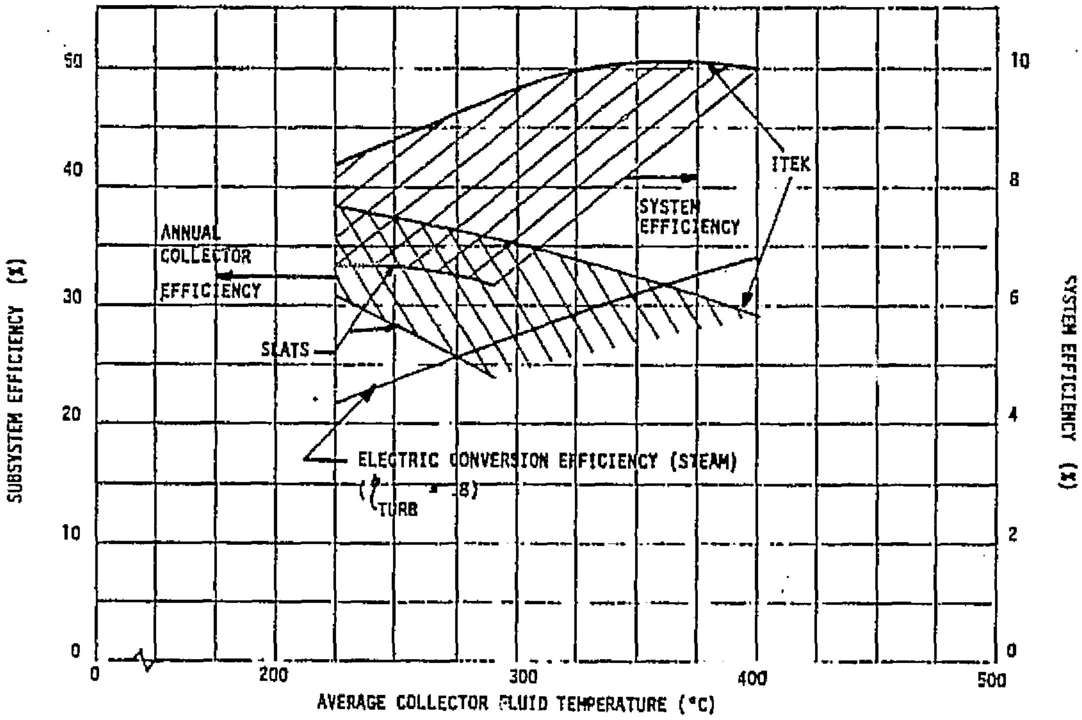


Figure 3.2-28. Segmented Mirror Collector Temperature Optimization

The selection of the heat transport fluid will dictate equipment which impacts reliability, program risk, and cost. Heat transfer oils were used for distributed collectors because a fluid with an elevated melting point is not feasible with a distributed collector field and steam imposes severe economic, performance and operational penalties. The optimum temperatures indicated in Figures 3.2-27 and 3.2-28 for the candidate distributed collectors are within the temperature limitations of heat transfer oils with the exception of the Itek segmented mirror collector. However, the performance penalty is only 3 percent if Syltherm 800 is used. This small reduction is acceptable considering the disadvantages of alternative high temperature fluids.

To provide a common basis of comparison with the central receiver concepts, the distributed collector subsystems were sized to match the central receiver system load factor. The load factor was calculated using the same sample Barstow insolation data. Thermal energy output profiles for the idealized parabolic trough and Barstow insolation data are presented in Figure 3.2-29.

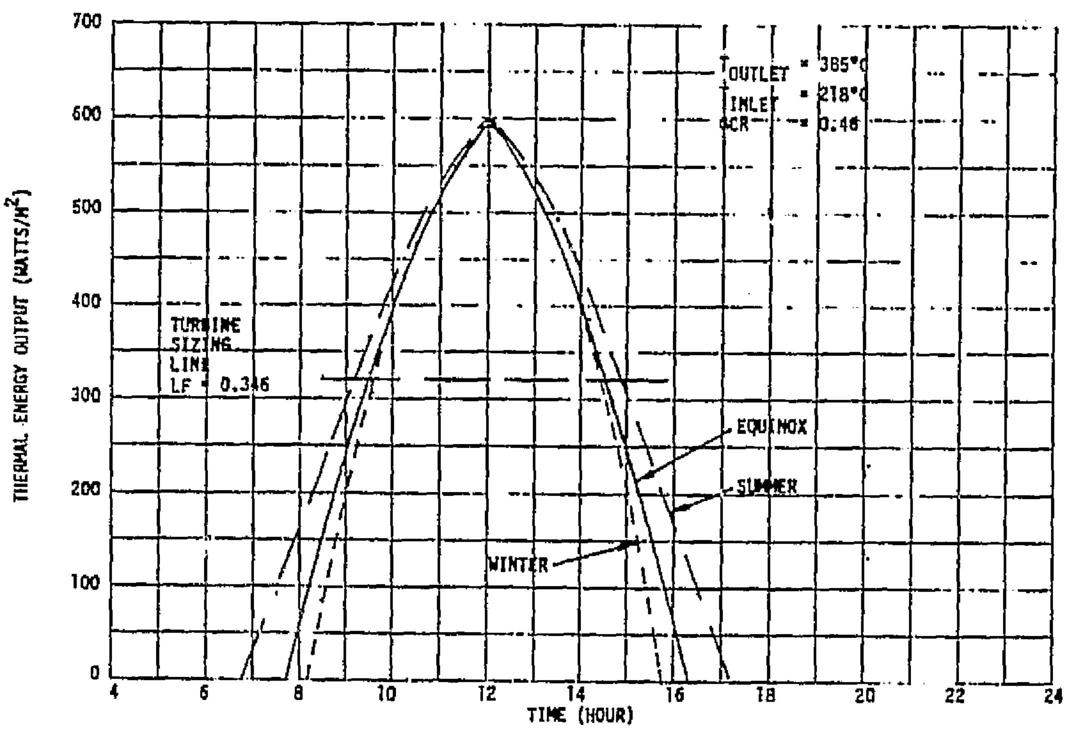


Figure 3.2-29. Idealized Parabolic Trough Performance for Reference Barstow Data

Table 3.2-11 summarizes the performance of the candidate distributed collector designs. Steady state and warm-up line losses were based on a distributed collector field design developed for the Solar Total Energy System - Large Scale Experiment No. 2 (Reference 31). The collector field design consisted of parabolic troughs with a total aperture area of 7000 m^2. Ground cover ratio was 0.47 and Therminol-66 was the selected heat transfer fluid. The total pipe length was 64.9m and consisted of nominal pipe sizes between 2-1/2 and 4 inch.

The distributed collector thermal load in this study was sized to provide 1.1 MWe (gross) and included the effects of the gear box and generator efficiencies (0.98 and 0.963, respectively). Collector field aperture area requirements were calculated based on matching the central receiver load factor (0.346 assuming 35 cloudy days).

Table 3.2-11

DISTRIBUTED COLLECTOR PERFORMANCE SUMMARY
STEAM CYCLE - RADIAL FLOW TURBINE

Collector	Parabolic Trough	Parabolic Trough	Segmented Mirror	Segmented Mirror
Company	Hexcel	-	Suntec	Itek
Temperature, Outlet (°C)	343	385	274	399
Temperature, Inlet (°C)	213	218	177	218
Collector Fluid	Therminol 66	Syltherm 800	Caloria HT-43	Syltherm 800
Average Energy Collection KW-HR/M^2/Day	2.88	3.10	2.91	3.28
Line Losses	0.966	0.962	0.971	0.962
Field Warm-up Loss	0.90	0.90	0.90	0.90
Cycle Efficiency, %	25.7	27.3	21.8	28.0
Thermal Load, MWth	5.21	4.93	6.12	4.81
Collector Area, M^2 @ LF = 0.346	17,480	15,410	20,330	14,190
Thermal Storage, MW-HR	13	12	13.2	10

Cost Estimates

For purposes of comparison with the central receiver, the distributed collector costs include the collector, energy transport, and energy storage costs. The distributed collector system is assumed to terminate at a central collection point, i.e., at the inlet to the steam generator. Current cost estimates of $200/m^2 for a parabolic trough were obtained in Reference 31 and includes site construction, packing, shipping and inspection. Of this total, the collector module price estimate was approximately $130/m^2 and was obtained from the Acurex Corp. A recent survey (Reference 34) of six manufacturers of parabolic troughs produced an average F.O.B. factory price of $150/m^2. Cost information for the segmented mirror was obtained from the Suntec Corp. and was in the range of $278-331/m^2. The greater current cost of the SLATS collector is due, in part, to the limited production status. A current cost estimate for the Itek unit was not obtained because, until test modules or prototypes are produced, meaningful cost data cannot be established.

The projected cost of collectors is dependent on assumed production rates. For the parabolic trough, an installed cost of $\$130/m^2$ has been estimated (Reference 35) for 1985. This assumes that collector production rates must reach the level where any one manufacturer is producing on the order of 10,000-100,000 m^2 per year in 1985.

Energy transport costs were based on an optimized field layout. Components included piping, pipe hangers and supports, valves, pumps, fluid, insulation, and blending tank. The energy transport cost estimate for the nearly 7000 m^2 field was \$180K, or $\$25.8/m^2$.

Energy storage was sized to accommodate the thermal energy in excess of the turbine demand. A dual media thermocline storage concept was assumed for Caloria HT-43 and Therminol 66 fluids. The storage concept selected for Syltherm 800 was a three-tank trickle charge. Cost estimates included the required ancillary equipment.

Breakdowns of capital costs for four distributed collector designs are shown in Figure 3.2-30. The power conversion costs have been excluded from this comparison because they are assumed constant for the different systems. The top

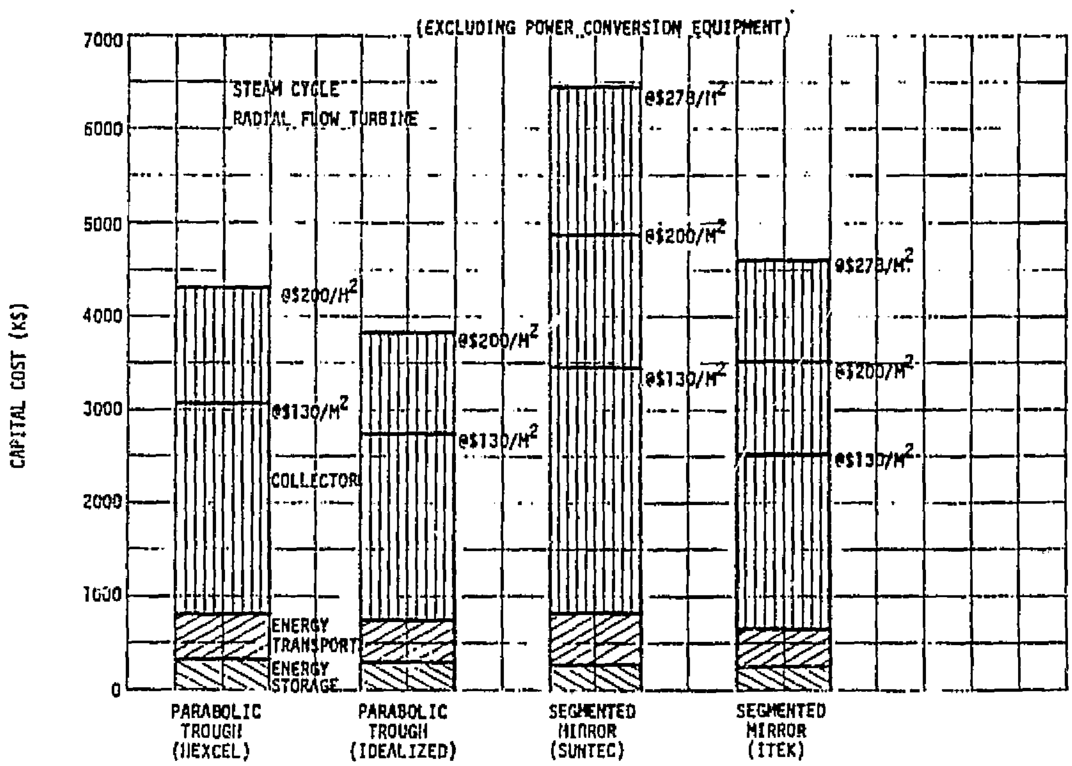


Figure 3.2-30. Distributed Collector Cost Summary

C-2

bar in Figure 3.2-30 represents the costs with current collector module costs and lower bars are shown to indicate system costs with projected decreases in collector costs. The capital cost summary should be interpreted in terms of near term and long term availability. The collector, heat transport, and thermal storage capital cost subtotal for the near term candidates is $4,300K for the Hexcel collector and $6460K for the Suntec collector. Projected long term performance improvements and reductions in unit collector costs lowers the parabolic trough capital cost to $2750K. A larger reduction in the parabolic trough system costs is obtained by the projected decrease in unit cost than with projected increases in performance. A greater uncertainty exists with long term projections for the segmented mirror. The most optimistic view is to assume that the breadboard performance of the Itek collector is attainable and that the same long term cost of $\$130/m^2$ as the parabolic trough can be achieved. Under this scenario, the capital cost of $2500K would be slightly lower than the idealized parabolic trough.

3.2.7.3 Candidate Concept Selection

The initial candidate selection compared the central receiver and distributed energy collection as a precursor to the actual system selection process. In all cases, the distributed collector approach was found to be significantly more expensive than the corresponding central receiver assuming the same power conversion subsystem for both approaches.

An example of the comparative costs between a central receiver and distributed collector approach is shown in Figure 3.2-31. In both cases, the energy collection portion of the systems are sized to provide an identical annual electrical load factor assuming identical radial flow steam turbine power conversion equipment. The data for the distributed collector equipment are taken from material presented in Section 3.2.7.2 while the central receiver information was presented in Section 3.2.7.1.

The most significant cost difference between these approaches involves the collector subsystem. Using the Nth cost numbers of $\$130/m^2$ and $\$78.5/m^2$,

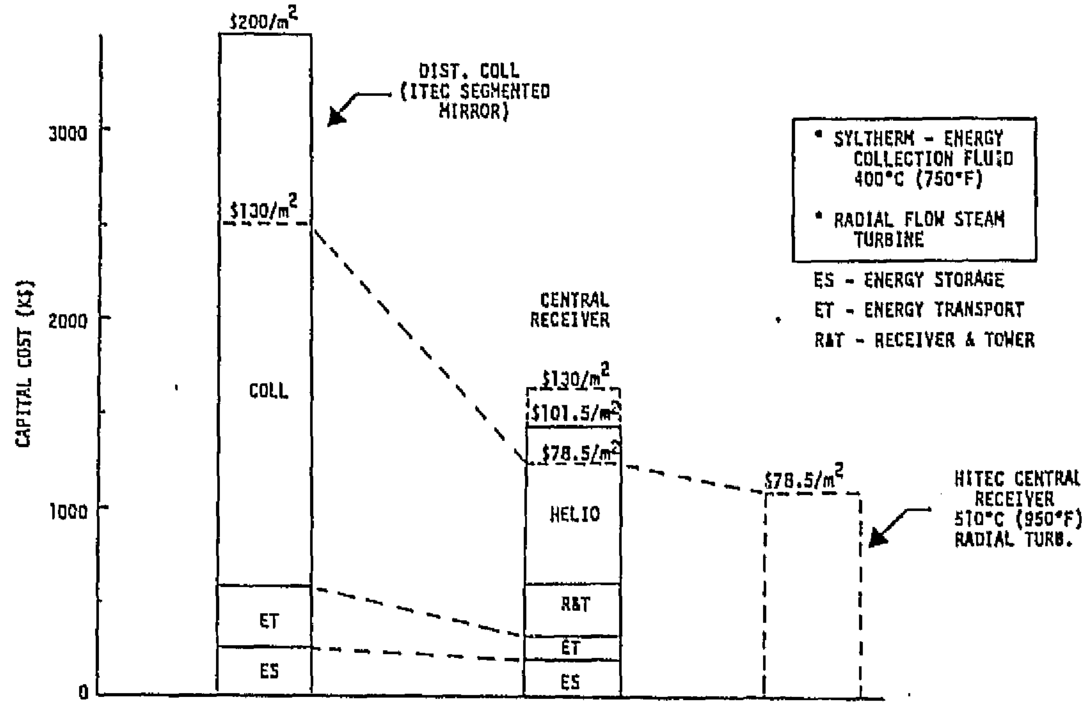


Figure 3.2-31. Comparison Between Collector Concepts (Central Receiver vs Distributed Collector)

respectively, for the distributed collectors and the heliostats, a factor of 2 difference exists between the collector subsystem costs. Even assuming an identical cost of $\$130/m^2$ for both approaches, the central receiver design offers a substantial economic advantage. This difference is attributable to the required oversizing of the distributed collector field in order to match the load factor produced by the central receiver system. This is because of the inferior annual collection capability of single-axis tracking devices.

In comparing the cost elements associated with the energy transport and energy storage subsystems, an additional advantage favoring the central receiver system is realized. From an energy transport standpoint, the distributed collector approach requires a significant quantity of expensive horizontal piping to distribute and collect the heat transfer fluid. By contrast, the central receiver system requires only separate riser and downcomer lines which run up and down the tower along with short horizontal sections.

The cost difference for the energy storage subsystems occur because of the difference in storage capacities required to provide the same load factor. This difference is directly attributable to the poorer distributed collector performance especially during early morning and late afternoon periods. As a result, a significantly larger quantity of thermal energy must be gathered and stored during the midday period for the distributed collector system in order to match the load factor of the central receiver system.

Two additional factors should be considered before drawing any final conclusions. First, performance limits for single-axis tracking collectors in general require them to be operated below 400°C (750°F). The central receiver system on the other hand experiences an optimum system performance at much higher fluid temperatures. As a result, additional improvements in cycle efficiency can be realized over the case indicated in the comparative evaluation shown in Figure 3.2-31.

A second factor which supports a higher temperature for the central receiver involves fluid freeze up. The only practical fluids for distributed collectors must have freeze points below the minimum expected ambient temperature. The best of these fluids have operating temperature limits of about 400°C (750°F). Higher temperature fluids such as heat transfer salts, which freeze well above normal ambient temperature, may be used in central receiver systems where adequate freeze protection provisions can be made without excessive cost and substantial operating problems.

An indication of the additional cost benefit that can be realized in utilizing a high temperature central receiver system is shown in the righthand bar of Figure 3.2-31. It shows that a further cost reduction relative to the Syltherm central receiver system can be realized in adopting a high temperature fluid design.

Based on the previous discussion, distributed collector subsystem designs are unattractive from a cost and performance standpoint relative to the central receiver system. In addition, the distributed collector approach requires higher cost energy transport and energy storage subsystems. As a result, systems based on the central receiver design were selected.

Table 3.2-12 presents a listing of the subsystem combinations selected for further analysis (all based on the central receiver design). This selection is based on data presented in Sections 3.2.4 and 3.2.7.1. They in general reflect the most cost effective concepts subject to various technology readiness constraints which are derived from the 3-1/2-, 4-1/2-, and 6-1/2-year startup periods.

Table 3.2-12

CANDIDATE SYSTEMS SELECTED

Receiver Coolant	Receiver Configuration	Thermal Storage	Prime Mover
Hitec	Two Zone	Dual-Media Thermocline	Radial Turbine
	Single Zone	Two-Tank	Axial Turbine
Syltherm	Two Zone	Dual-Media Thermocline	Supercritical Organic
	Cavity	Trickle Charge	Radial Turbine
			Axial Turbine
Caloria	Two Zone	Dual-Media Thermocline	Subcritical Organic
	Cavity	Two-Tank	Radial Turbine
Saturated Steam	Single Zone	Pressurized Water	Radial Turbine
Air or He	Cavity	(Battery)	Gas Turbine

Candidate designs which have been rejected include: (1) liquid sodium systems due to high cost, system complexity, and institutional factors; (2) saturated steam systems employing a reciprocating steam engine due to high cost and low performance; (3) Caloria receivers combined with axial steam turbines because of high cost and the availability of subcritical organic Rankine equipment with superior efficiency; and (4) Brayton cycle using thermal storage because of the need to use tower-mounted equipment to maintain acceptable gas turbine performance.

Table 3.2-13 shows how these candidate systems are allocated to the potential startup periods. In general, as longer startup times are assumed, higher operating temperatures and more advanced equipment is assumed.

Table 3.2-13 (Page 1 of 2)
CANDIDATES FOR THREE PROJECT DURATIONS

Receiver Fluid	3-1/2 Years	4-1/2 Years	6-1/2 Years
HTS			
Temperature Limit (°C)	430-510	510	510-580
Thermal Storage	Two-Tank	Two-Tank	Dual-Media Therm.
		Dual-Media Therm.	
Prime Mover	Axial Turbine	Axial Turbine	Radial Turbine
		Radial Turbine	
Syltherm			
Temperature Limit (°C)	400-454	450-480	450-480
Thermal Storage	Trickle Charge	Trickle Charge	Trickle Charge
		Dual-Media Therm.	Dual-Media Therm.
Prime Mover	Axial Turbine	Axial Turbine	
		Radial Turbine	Radial Turbine
		Supercritical Organic	Supercritical Organic
Caloria			
Temperature Limit (°C)	300-316	316	316
Thermal Storage	Two-Tank	Two-Tank	
	Dual-Media Therm.	Dual-Media Therm.	Dual-Media Therm.
Prime Mover	Subcritical Organic	Subcritical Organic	
		Radial Turbine	Radial Turbine
Saturated Steam	X		
Temperature Limit (°C)		500-600	500-600
Thermal Storage		Pressurized Water	Pressurized Water
Prime Mover		Radial Turbine	Radial Turbine

Table 3.2-13 (Page 2 of 2)
CANDIDATES FOR THREE PROJECT DURATIONS

Receiver Fluid	3-1/2 Years	4-1/2 Years	6-1/2 Years
Air	X		
Temperature Limit (°C)		680-820	680-820
Storage		Battery	Battery
Prime Mover		Gas Turbine (Open)	Gas Turbine (Open)
Helium	X		
Temperature Limit (°C)		680-820	680-820
Storage		Battery	Battery
Prime Mover		Gas Turbine (Closed)	Gas Turbine (Closed)

3.3 SUBSYSTEM OPTIMIZATION/DESIGN

Subsystem optimization was performed for each of the candidate systems which survived the initial screening process. Based on these analyses, the subsystems were reassembled into candidate systems in a manner which reflects the most favorable design for each system and provides the most realistic estimates for subsystem cost and performance. This is in contrast to the cost and performance analyses during screening which made extensive use of cost and performance scaling relationships. With the more detailed information developed in this task, a final selection was made of the candidate systems to be designed for the 3-1/2-, 4-1/2-, and 6-1/2-year startup times. This selection process is described in Section 3.4 of this report.

Before considering the detailed subsystem analyses, it is important to establish an overall design philosophy which can then give direction to the analysis activities. In striving for the goal of a high commercialization potential by the late 1980's, it is important to develop the most cost effective system which can take advantage of the technology developed by that time. It is appropriate to view the 3-1/2-, 4-1/2-, and 6-1/2-year startup times as alternative steps along the path of development for the ultimate system which will be available by the late 1980's. As a result, the subsystem optimization analyses gave consideration to the long term potential of each subsystem as well as optimizing the subsystem for shorter term applications. During the final system synthesis and selection process, the three preferred systems were selected on the basis of both technology readiness and the degree of achievement of the ultimate system design. For example, if a radial steam turbine was determined to be superior on a long term basis, all of the systems would be purposely configured using a water/steam power conversion cycle. For short term systems (3-1/2 years) in which the radial flow turbine equipment may not be available, off-the-shelf axial turbine equipment was selected because of the desire to proceed toward verification of the ultimate system.

The individual subsystems optimized in this task are identified in Figure 3.2-32 in terms of major elements and the subsystem interrelationships

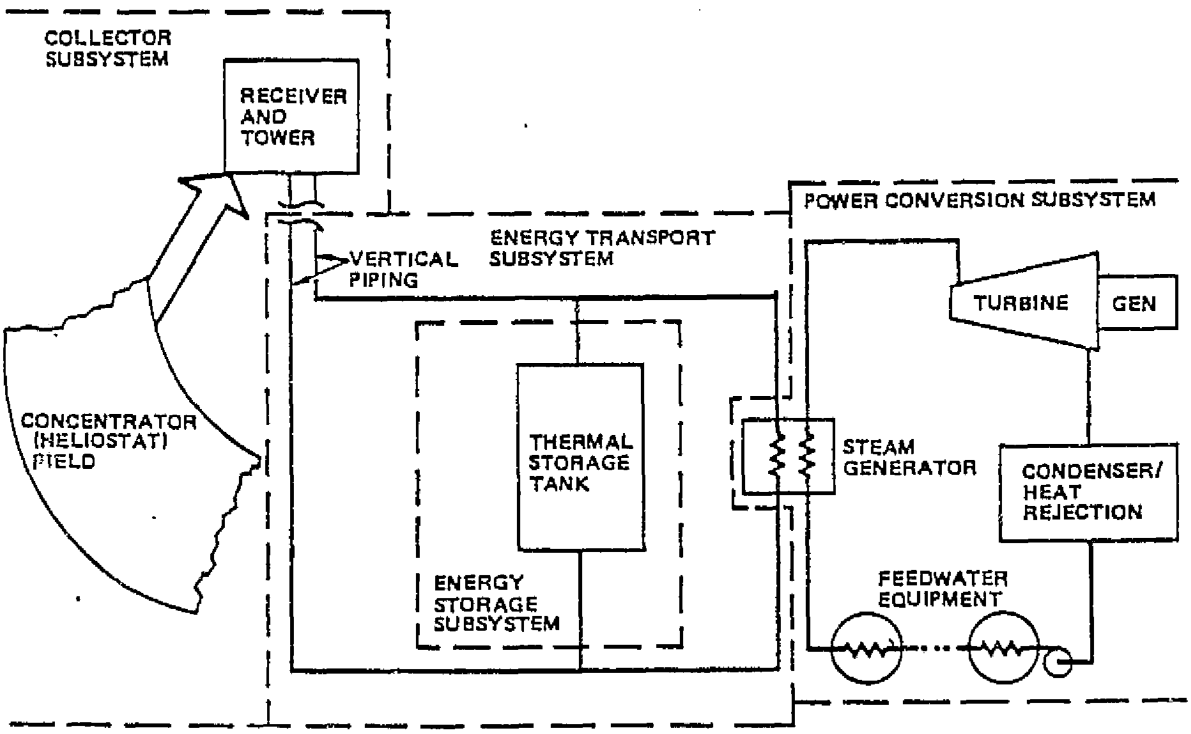


Figure 3.2-32. Definition of Subsystem

which exist. For systems which employ electrical storage, this schematic must be modified by moving the energy storage subsystem downstream (in an energy flow sense) of the power conversion subsystem. All subsystems were configured and analyzed on the basis of eventual synthesis into systems having a 1 MWe net plant output with a capacity factor of 0.4. Each of the subtask discussions presented in this section treat the optimization analyses carried out on the individual subsystems.

3.3.1 Concentrator Design

The objective of this effort was to select a concentrator design suitable for a small central receiver system. The concentrator is defined to include the heliostats, the control and power distribution, and the concentrator control function and hardware interface with the plant controller.

3.3.1.1 Concentrator Requirements

The complete set of design requirements is contained in DOE Specification 001 (Reference 37). Some of the more significant requirements are summarized in Table 3.3-1.

3.3.1.2 Baseline Concentrator Description

The baseline concentrator is made up of three assemblies. The heliostat assembly is shown on Figure 3.3-1. The other two assemblies are: (1) the concentrator controller which is collocated with and may be an integral part of the plant controller, and (2) field electronics consisting of power and data feeders.

Table 3.3-2 shows a subsystem hardware tree down to the component level.

Heliostat Summary Description

The heliostat (Figure 3.3-1) is divided into four subassemblies, based on the physical pieces of hardware delivered to the field. These subassemblies are the reflector panel (one half of the reflective unit), the drive unit (including the pedestal), the foundation, and the heliostat electronics (including controllers and control sensors).

Table 3.3-1
CONCENTRATOR DESIGN REQUIREMENTS

Category	DOE Requirement	Small Central Receiver System Requirement
Performance	Operate sunrise to sunset	Same
	Clean reflectivity $\geq$ 0.91	Same
	Maximum cost effective reflector area (set at 49 M^2)	Same
	Tracking accuracy optimized (Set at 1.7 mr RMS)	Greater errors may be acceptable
Environmental	Maximum operational wind (Set at 16 M/S)	Same
	Maximum survival wind 40 M/S	Same
	Sudden winds up to 21 M/S	Same (may be unnecessarily high)
	Temperature -30°C to +50°C	Same
	Earthquake, Seismic Zone 3 (0.25 G's)	Same
	Snow/ice load 350 Pa	Same
	Hail up to 25 MM at 23 M/S	Same
Operational	Availability > 0.97 (Estimated > 0.999)	Same
	All units interchangeable	Same
	Maintain beam control at all times	Same
	Fail safe operation	Same
	Safe stow capability (Inverted stow provided)	Same
	Easy removal for maintenance	Same
	Access space provided	Same
	30-year design life	Same
	Reflector design for periodic cleaning	Same
	Easy service with normal skills	Same

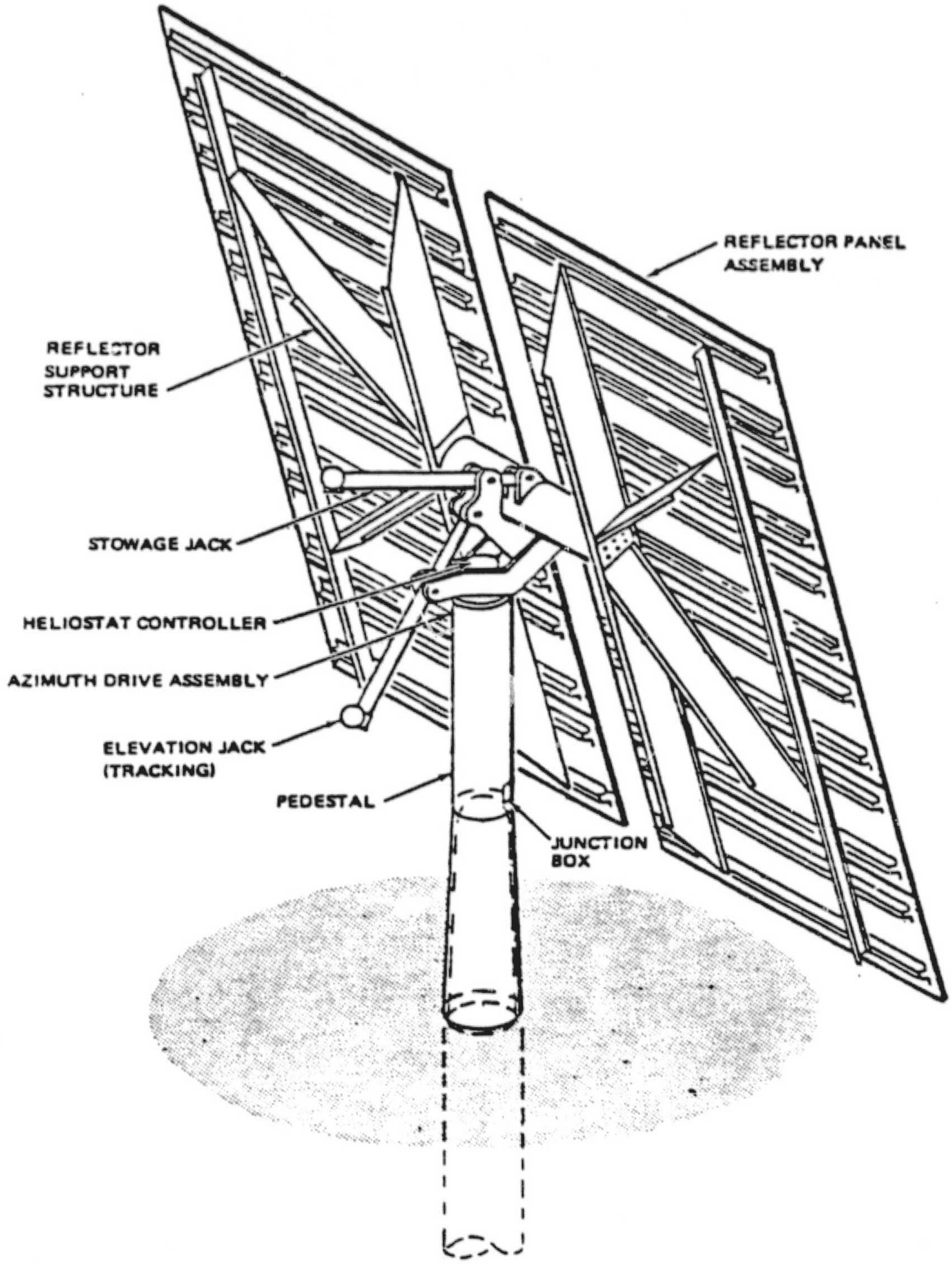


Figure 3.3-1. Primary Baseline Heliostat

Table 3.3-2
BASELINE HELIOSTAT HARDWARE TREE

Subsystem	Assembly	Subassembly	Component
• Concentrator - (Field of heliostats)			
	• Heliostat - (Includes controller)		
		• Reflector panel - (two panels make a reflective unit)	
			• Mirror module
			• Support structure
		• Drive unit	• Azimuth drive
			• Elevation drive
			• Pedestal
		• Foundation	
		• Heliostat Electronics	• Heliostat controller
			• Motor
			• Pedestal junction box
	• Concentrator Controller (May be part of Plant Controller)		
		• Console	• Key board
			• Cathode ray tube
			• Control panel
		• CPU	
		• Storage	
		• Field interface	
		• MCS interface	
		• Time pickup	
	• Field electronics	• Power distribution	
		• Data distribution	

Reflector — Each reflector panel is composed of six mirror modules and a support frame. The mirror modules are 1.22 by 3.35 m (48 by 132 inches) and made of a 1.5 mm (0.060 inch) second surface mirror laminated to a 4.8 mm (0.1875 inch) glass back panel. The clean reflectivity is varied from 0.92 to 0.95, depending on iron content and chemical state. The mirror modules are bonded to stringers which are, in turn, bolted to the cross beams. The outer cross beam is supported by two diagonal beams. All beams and stringers are made by continuous roll-forming from coiled sheet stock.

This design was derived to achieve a direct production cost reduction compared to the Barstow Pilot Plant configuration (Paragraph 3.3.1.3), and provides an indirect cost reduction by use of a thinner glass with higher reflectivity. The total reflector area is increased commensurate with the drive unit loads. Of significance to Small Power Systems is that the reflector is in two panels which are fully prealigned, transportable by common carrier, and can be installed with a minimum of field labor and equipment. Errors in alignment of the panel can be easily adjusted by methods already demonstrated by MDAC.

Drive Unit — The drive unit is composed of a rotary azimuth drive, a double jack elevation drive, and a pedestal. All drive motors are three-phase, 480 VAC. A 162:1 Helicon input reducer provides the first azimuth stage reduction. The output is through a 242:1 Harmonic drive reducer. The elevation jacks utilize a Helicon input gear affixed to the shaft of a ball screw. The two jacks are connected by a drag link. One jack provides tracking motion while the other provides the additional motion required for stowage. The main beam is an 0.4m (16-inch) diameter tube with flange ends onto which the reflector panels are bolted. The tube has brackets which attach to a hinge line on one side and the tracking actuator on the opposite side, providing the final linkage of the elevation drive. The pedestal is an 0.6m (24-inch) diameter tube with a slight flare on the lower end which matches the tapered top of the foundation and provides a friction joint to the foundation. The top of a pedestal is closed by a dome which bolts to the circular spline of the Harmonic drive.

The drive unit is delivered to the field with the heliostat electronics installed.

This design incorporates a number of improvements, such as a lower-cost, more efficient jack design, lower-cost gears and bearings, and a pedestal design that allows simple field installation. The drive unit with its central main beam also allows a rapid and efficient field installation of the reflector panels in two pieces. The design requires no scheduled maintenance. Removal and replacement of failed parts may be accomplished easily at the component or subassembly level. Repairs are simple, require no special tools, and utilize a piece part remove-and-replace approach.

Heliostat Electronics — The heliostat controller is located in a housing on the top of the drive unit. The controller receives and transmits commands from the collector controller and responds to requests for data. A microprocessor calculates the motor revolutions required to maintain tracking and activates the motor controllers. The motor controllers switch the motor on and off to produce the required motion. The motor revolutions sensors detect motor revolution and direction, and the controller maintains a count of the accumulated revolutions. A nonvolatile memory retains motor counts and alignment data in the event of a loss of power. The field wiring terminates at a junction box located on the pedestal. A "tee" junction provides the power to operate the heliostat. Data are routed to the heliostat controller, decoded, and relayed to the next heliostat in the link if not addressed to the receiving heliostat. Acknowledgment of receipt of a message and status are also transmitted.

The design of an integrated pedestal, drive, and electronics unit permits complete assembly and automated functional checkout testing to be done in the factory.

Foundation — The foundation is a drilled pier, 0.6 m (24 in.) in diameter. The pier extends about 1.2 m (4 ft) above grade and 6 m (20 ft) below. A tapered steel shell establishes the mounting surface to the pedestal and serves as a form for the protruding end of the pier. This design speeds field installation, reduces costs, and decreases the amount of steel required for the pedestal by over 272 kg (600 pounds).

Field Electronics Summary Description

The field electronics distribute power and data to the heliostats. Those loops are illustrated in Figure 3.3-2.

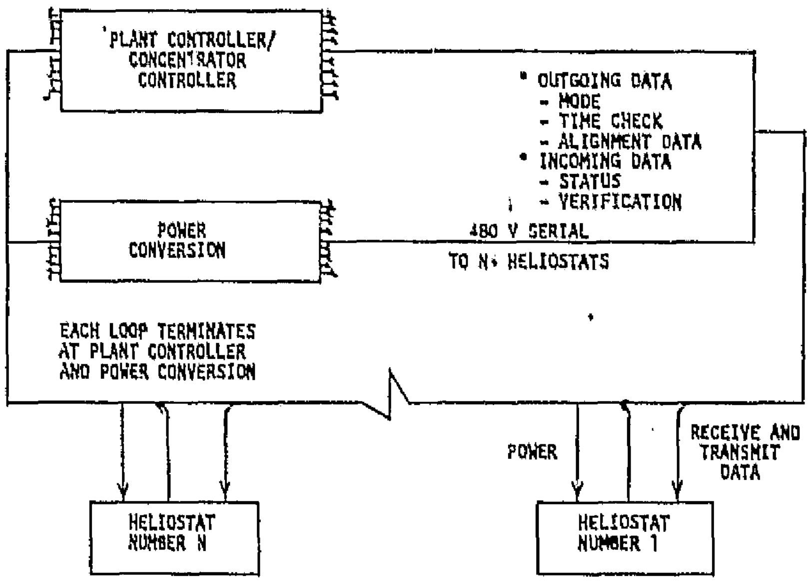


Figure 3.3-2. Collector Field Electronics

A field distribution panel receives power from the power conversion subsystem and data from the concentrator controller. Power and data are dispatched in the same cable to a "daisy chains" of heliostats; i.e., heliostats connected by a single cable which tap power and data off that cable. Each cable is terminated at both ends at the distribution panel. Hence, power may be fed either way on a cable if the cable fails open as in a break. A short circuit in a cable will, of course, trip the breaker in the distribution panel and cause the loss of power to all heliostats in the chain.

The control signals carried by the cable are all processed by the first heliostat in the chain. Those signals which are addressed to other heliostats are simply repeated, hence routed to the next heliostat. Signals addressed to the Nth heliostat are received by that heliostat and an acknowledgment signal is transmitted. The acknowledgment signal, which may include requested data on heliostat status, is relayed to the field distribution center at the end of the chain. From the panel, data are relayed directly to the heliostat array controller.

Each heliostat has the capability to continue to operate autonomously in the event of a loss of data signals. If no data are received in a specified length of time, the heliostat will continue to track. The collector controller will monitor the signals received from the communications loops. The controller will notify the operator when an anomally is detected.

3.3.1.3 Barstow Pilot Plant Heliostat Baseline Design Description

The heliostat design selected for the DOE 10 MWe Pilot Plant is illustrated in Figure 3.3-3. This design is the precursor to the baseline design described above and may be utilized for the shorter schedule options for Engineering Experiment No. 1. A brief description of this design is given below.

Mirror Module — The mirror module is a bonded sandwich consisting of a second-surface silvered mirror of low iron float glass, a foam core, and a thin, galvanized steel back sheet. Total reflective surface area is 45.3 m^2 (487 ft^2).

Support Structure — The support structure consists of a tubular main beam and four channel cross beams. Twelve mirror modules are back bolted to the cross beams with shallow cups to spread the load.

Drive Unit — Azimuth rotation is obtained by three reduction stages. The first stage is integral with a 240-VAC, three-phase induction motor, the second stage is a worm/gear pair, and the third is a Harmonic drive unit. The elevation drive employs two machine screw jack actuators coupled with a drag link to provide for the required 180-degree rotation. Each jack is driven by a similar gear motor. The azimuth housing and drag link are castings.

Pedestal/Foundation — A tubular steel pedestal is attached to the drive unit on the upper end and to the foundation on the lower end by bolted flanges. The foundation may be either a precast spread footing or a drilled pier. The anchor bolts are wired to the reinforcement in either case.

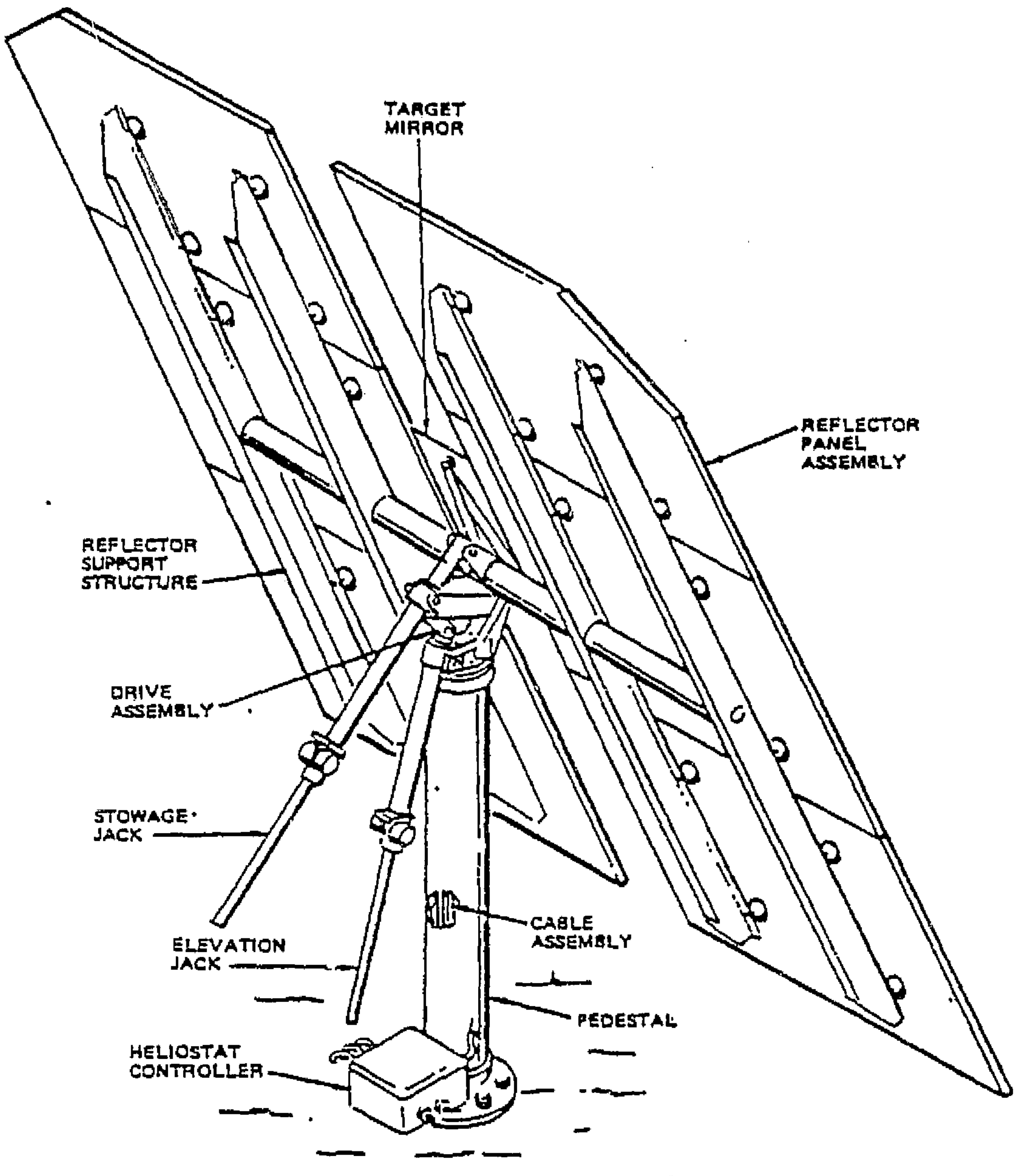


Figure 3.3-3. DOE 10 MWe Pilot Plant Heliostat

Controls — The heliostat employs open-loop control with motor revolution counters for tracking and four-bit absolute encoders on both gimbal axes for periodic update restart capability.

A heliostat controller located on each heliostat retains the motor revolution counts and generates error signals from data transmitted by field controllers. The motor controller section of the heliostat controller then executes the required motor revolutions indicated by the error signal.

Field controllers are located to service approximately 24 heliostats. The field controllers serve as a data interface with the collector controller and calculate time, ephemeris, and gimbal axis position data to transmit to the heliostat controller.

The field electronics (Figure 3.3-4) include primary feeders of power and data to the field controllers. Both hookups are serial. Branching networks from the distribution panel connect approximately 24 heliostats in a serial or daisy chain arrangement. Similarly, a serial connection is used between the field controllers and the heliostat controllers.

While this concentrator design could be utilized with no modifications, minor modifications to the design are recommended to simplify transportation and final assembly. Those modifications include:

(1) Dividing the main beam into three pieces and placing a bolted joint at the inboard cross beam.
(2) Assembling reflector panels completely in the factory so that alignment is ensured.
(3) Assembling the center portion of the main beam to the drive unit in the factory.

3.3.1.4 Baseline Heliostat Costs

The baseline heliostat costs will depend strongly on the heliostat production rate and cost reduction activities sponsored by DOE for the near term. Figure 3.3-5 shows a range of expected costs for heliostats as a function of cumulative units produced. The ordinate, $\$/m^2R$, shows the cost per unit area of the heliostat normalized to unit reflectivity. The upper boundary of the

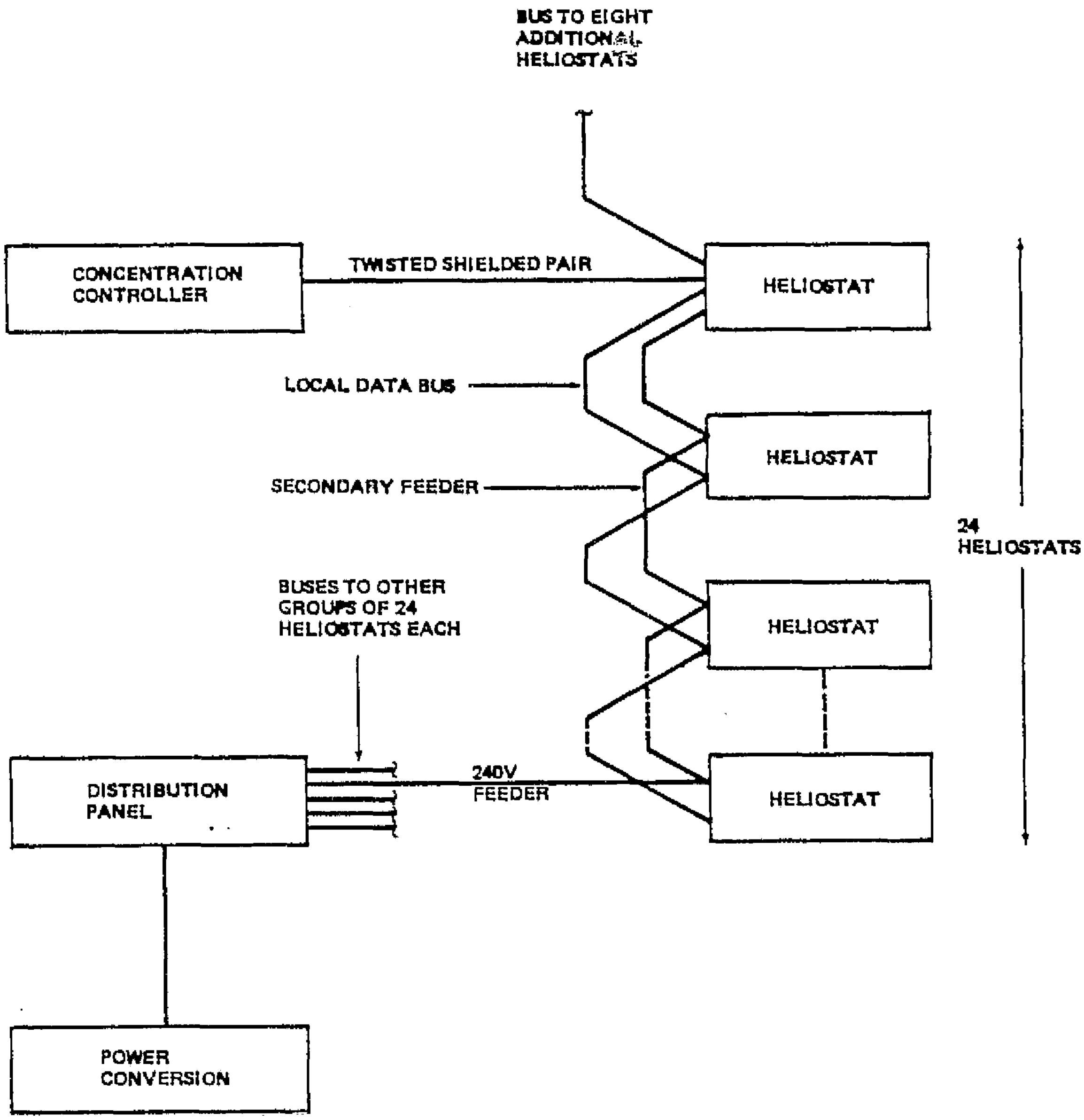


Figure 3.3-4. Branch-Collector Field Network

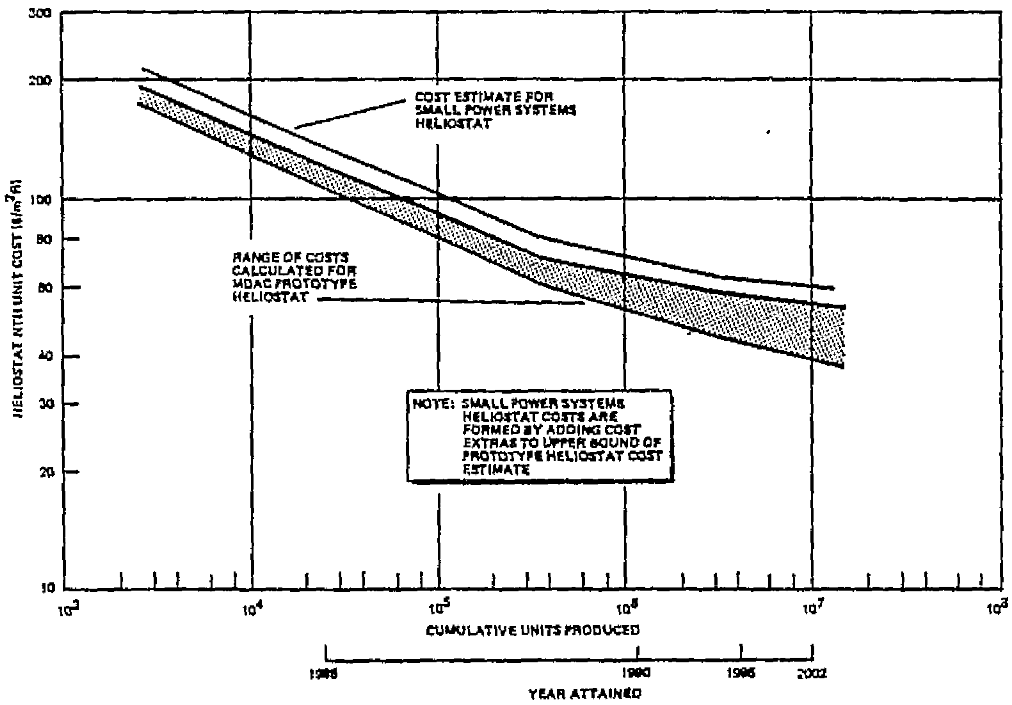


Figure 3.3-5. Expected Heliostat Cost Reduction

range represents costs on a commercial business basis, assuming no guaranteed Government market. The lower curve shows projections based on a guaranteed Government market. The abscissa also shows the year in which the cumulative production is attained, based on DOE projection of a heavily stimulated market.

The baseline data for Figure 3.3-5 are for deployment of heliostats in large collector fields. Typically, there will be about 18,000 heliostats in a field for a 100 MWe plant. Automated installation equipment is used and subassembly is located close to the deployment point. For a small power system, the automated installation equipment is probably not economically justified. Subassembly may not be close to the deployment point. Moreover, special cant angles and curvature must be built into the heliostat reflector panels. For these reasons, higher costs are expected.

The following upward cost adjustments were made:

(1) Transportation costs between the subassembly facility and the deployment site were increased by a factor of 5 (mean radius of 300 miles).

(2) Installation costs were increased by a factor of 3 (one three-man crew installs one heliostat per hour)

(3) Assembly costs for the reflector panels were increased by a factor of 3 for the special handling.

The upper curve of Figure 3.3-5 then represents the projected estimates of the costs of the heliostat.

3.3.2 Concentrator Field Optimization

The purpose of the concentrator field optimization was to establish sizing requirements for the concentrator field, receiver, and tower which result in the lowest cost of thermal energy on an annual basis. Due to optimization work being carried out in parallel on other subsystems, the sizing requirements were developed on a parametric basis as a function of annual energy. In order to bracket the analysis, annual thermal energy requirements between 11,000 and 19,000 MWH were considered to satisfy the 0.4 capacity factor requirement for power cycles with conversion efficiencies in the range of 0.35 to 0.20.

3.3.2.1 Field Optimization Methodology

The optimization analysis, which was carried out by the University of Houston, utilized well established computer codes which have been exercised extensively in support of other DOE contracts (References 38 and 39). The objective of the codes is to determine the most cost effective approach to the gathering and delivery of thermal energy to the base of the tower over a representative one year period. The resulting subsystem characteristics are, of course, dependent on the nature of the inputs assumed for the analysis. Table 3.2-3 presents a listing of the principal study inputs along with typical values for the current study.

Before initiating the actual optimization procedure, the collector field is divided into a 14 by 15 array with each element serving as a basic computational cell. For each of these cells, a data base is established which

Table 3.2-3

SUMMARY OF INPUT DATA

Heliostat Cost (Total)	$\$81/m^2$
Heliostat Wiring Costs (Included in above cost)	
Cable	\$8.16/m
Trenching	\$6.10/m
Receiver Cost	
4 m x 4 m aperture	\$89K
4.5 m x 4.5 m aperture	\$107K
Tower Cost @ 40 m Optical Height	\$69K
Riser/Downcomer Cost	$\$23 \left(\frac{\text{Power}}{3.7\ \text{MWt}}\right)^{0.5} K$
Pump Cost (25 HP @ 3.7 MWt)	\$165/HP
Land Cost	\$5K/Acre
Fixed Cost	\$30K
Heliostat Area	$49\ m^2$
Receiver Loss Model	0.05 (Incident Power)$_2$ + 0.433 MWt (area/lbm^2)
Heliostat Error Budget	2.83 mr (1σ)

contains annual cell performance information as a function of heliostat spacing in two orthogonal directions. This performance information, which reflects blocking and shadowing data, is the basis for the subsequent optimization analysis.

The optimization procedure starts by defining the spacing of heliostats in each cell so that each cell has a figure of merit (i.e., cost/annual energy) equal to an assumed input value. The individual cells are then ordered in terms of decreasing performance, and the field is then reconstructed on a cell-by-cell basis.

This reconstruction process starts by determining the cost of annual energy for a system powered by the single, highest performance cell. Clearly, because the costs include consideration of the receiver, tower, piping, and pump, the cost/annual energy for a field containing only one cell is very large. The cells are added sequentially with the cost energy ratio being recomputed each time an additional cell is added until a minimum value is reached. At this point, the field trim is established since the inclusion of any more cells would result in an increase in the cost annual energy ratio. The minimum cost energy ratio represents the output figure of merit which is then compared with the input value. If a difference exists, the process is iterated until a convergence is established between the input and output values.

Implicit in the figure of merit are the influences of all cost and performance considerations which can be allocated to the individual heliostats. These factors include:

A. Shading and blocking of adjacent heliostats.
B. Guidance error model.
   1. Slope errors of reflectors.
   2. Tracking errors.
C. Aberration model for canted heliostats.
D. Heliostat aim strategy.
E. Cost model.
   1. Heliostats (including guidance, etc.)
   2. Tower.
   3. Receiver.
   4. Plumbing in tower.
   5. Land for heliostat.
   6. Wiring for heliostat.
   7. Receiver feed pump.
F. Energy loss model.
   1. Mirror reflection and receiver absorption.
   2. Receiver absorptivity versus angle of incidence.
   3. Reradiation and convection from receiver.
   4. Atmospheric losses between heliostat and receiver.
   5. Interception losses at receiver.

The interception factor data between individual heliostats and the receiver were calculated off line and used as inputs to the optimization analysis. A description of the approach used to define the interception factors for each cell is presented in the next section of this report.

The information developed as a result of this optimization analysis includes a specification of the optimized cost of annual energy, the annual energy absorbed into the receiver working fluid, the peak power level, field shape, and heliostat spacing data for each of the remaining computational cells. Simple changes in tower height and receiver size (expressed in terms of revised interception factors) will result in a new set of collector subsystem performance and design data. This process was repeated until a sufficient parametric data base was established to cover the range of interest from 11,000 to 19,000 MWH of annual thermal energy.

3.3.2.2 Receiver Interception Factor

The average annual receiver interception factor (AIF), which is a primary input to the concentrator field optimization analysis, is defined as the ratio of the total annual energy collected within the aperture to the total annual energy redirected by the heliostat field. This value depends on both the ambient temperature which can influence heliostat surface shape and heliostat location within the field. Separate studies were carried out to determine the effect of each of these parameters on the AIF using the McDonnell Douglas optical analysis computer code (CONCEN).

Temperature — The mirror panels show bending deformation at temperatures other than that at which the assembly was bonded (21°C [70°F]). At higher temperatures the bending is concave on the mirror side; at lower temperatures the mirror side becomes convex. Structural analysis has indicated that a model approximating a slope deviation linear with distance from a point somewhat displaced from the center may be employed. A suitable algorithm defining the temperature effect has been incorporated in the CONCEN code.

If it is anticipated that the mean operating temperature will be somewhat higher than the 21°C (70°F) bonding temperature, partial compensation for the thermal bending can be obtained by reducing the curvature during the bonding process. For example, to establish a curvature of 0.0025 m^{-1} at 32°C (90°F), a curvature of 0.00164 m^{-1} would be required at the bonding temperature. These effects on AIF as a function of receiver aperture width are shown in Figures 3.3-6 through 3.3-11. In Figure 3.3-6, the 6.5 m x 6.5 m panel curvature is 0.0025 m^{-1}, which is correct focusing at a slant range of 200 m and an ambient temperature of 21°C (70°F). Curves are shown for temperatures of -1°, 21°, and 43°C (30°, 70°, and 110°F). The AIF, as expected, is highest for the 21°C (70°F) curve. In Figure 3.3.7, the initial panel curvature is 0.00164 m^{-1}, correct for focusing at 32°C (90°F). The performance at 21°C (70°F) is slightly reduced, while the AIF is markedly improved at 43°C (110°F) and degraded at -1°C (30°F). Figures 3.3-8 and 3.3-9 show similar curves for a heliostat at 100 m slant range with the curvature being maintained for the 200 m focal length. Since the panels are under curved for the shorter range, the effect of elevated temperature is to curve the panels closer to the best focus condition. Consequently, the 43°C (110°F) curves lies above the 21°C (70°F) curve. Reducing the curvature to 0.00164 m^{-1} mainly drives the -1°C (30°F) curve downward (Figure 3.3-9). When

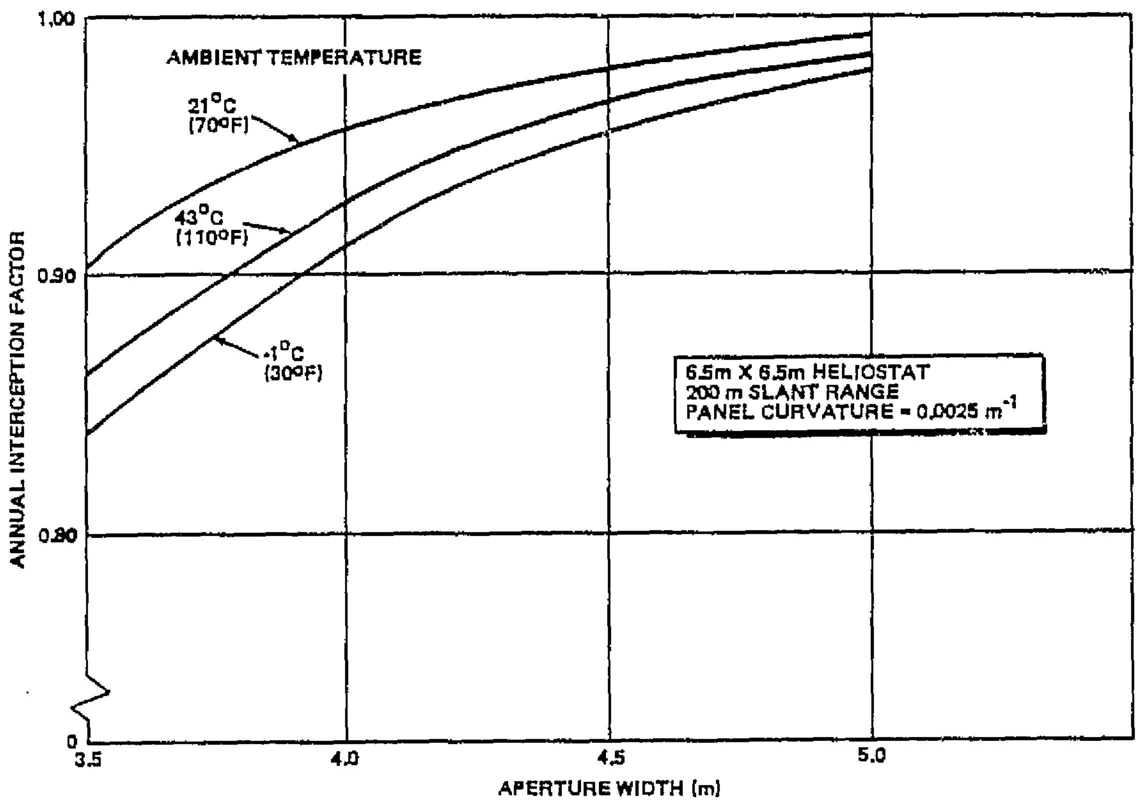


Figure 3.3-6. Temperature Effects on AIF (1)

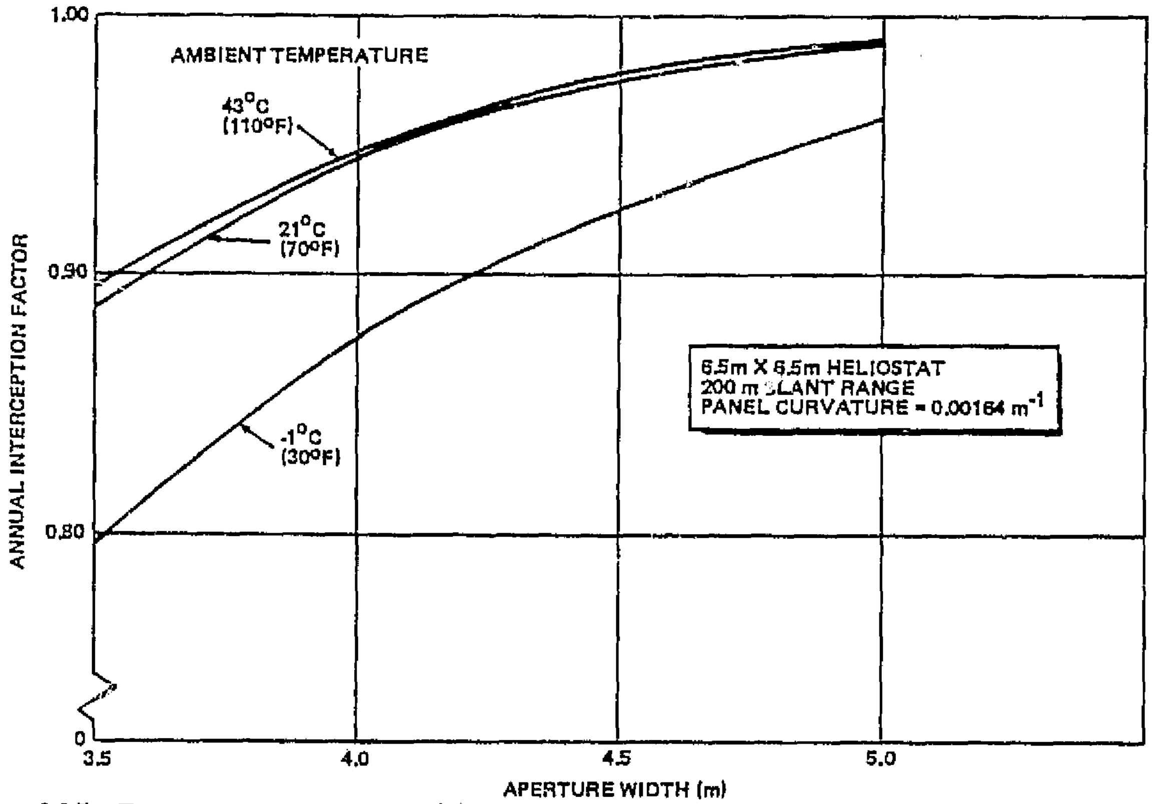


Figure 3.3-7. Temperature Effects on AIF (2)

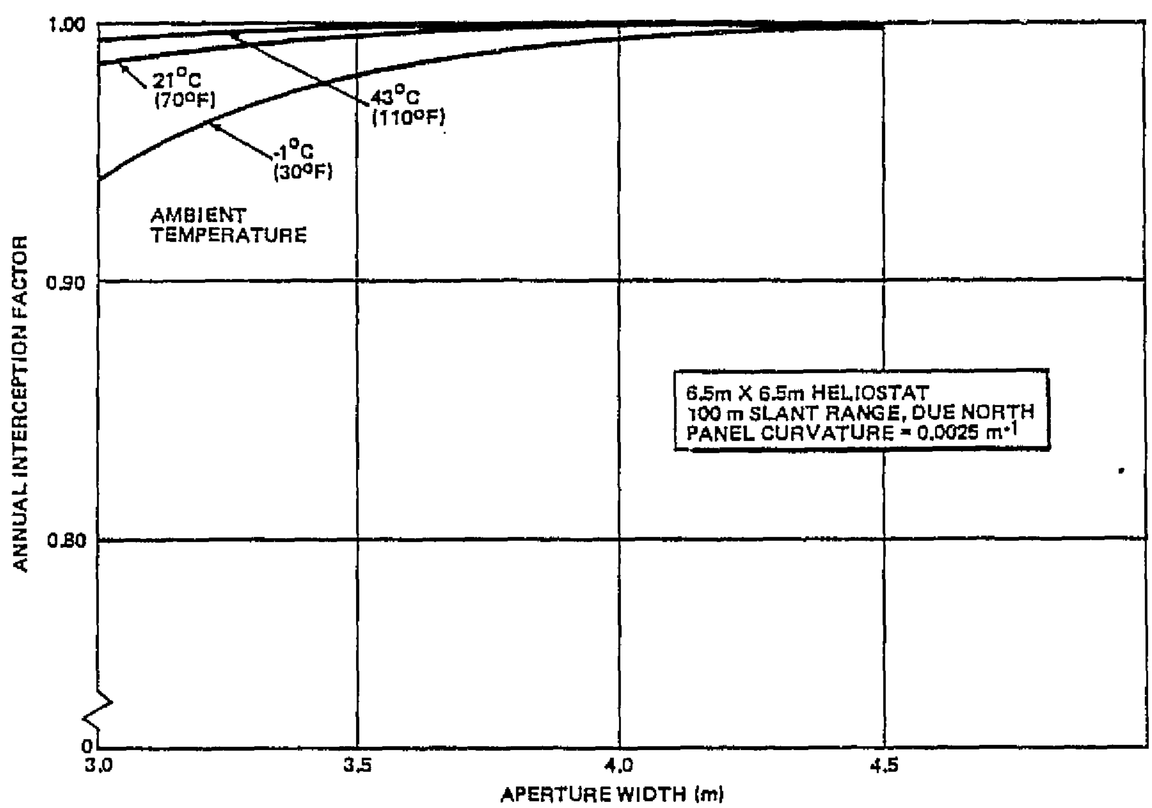


Figure 3.3-8. Temperature Effects on AIF (3)

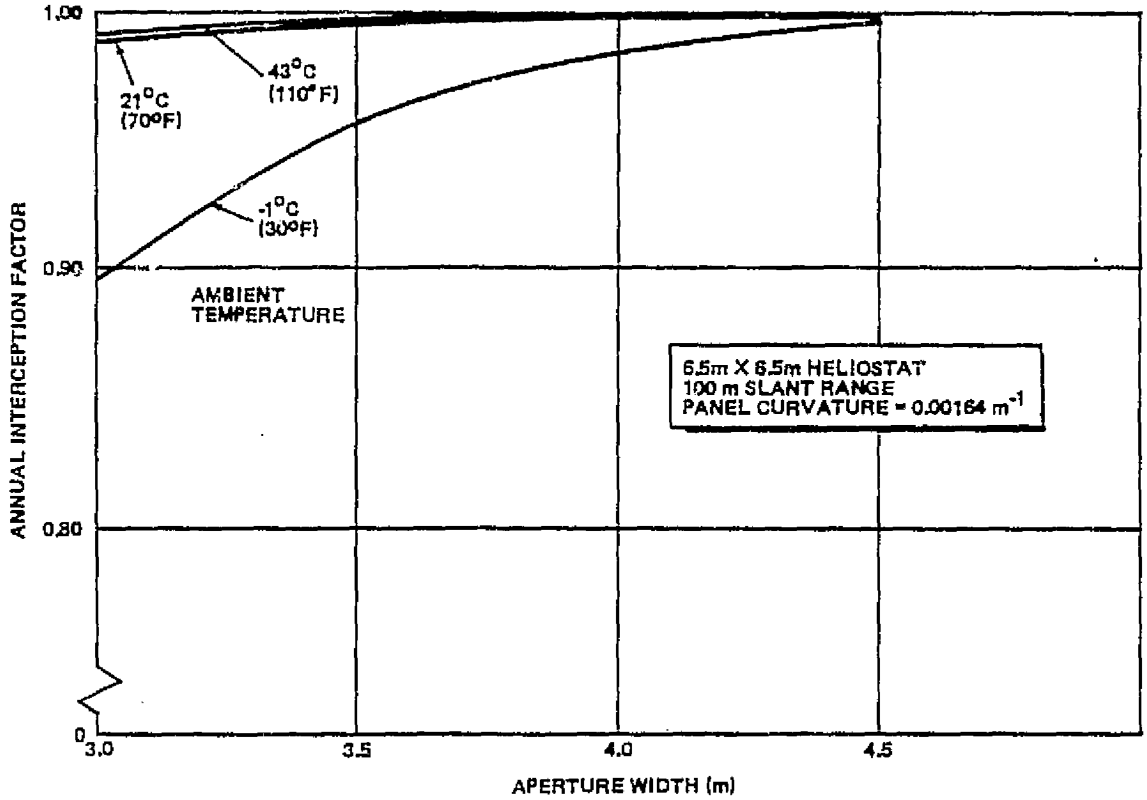


Figure 3.3-9. Temperature Effects on AIF (4)

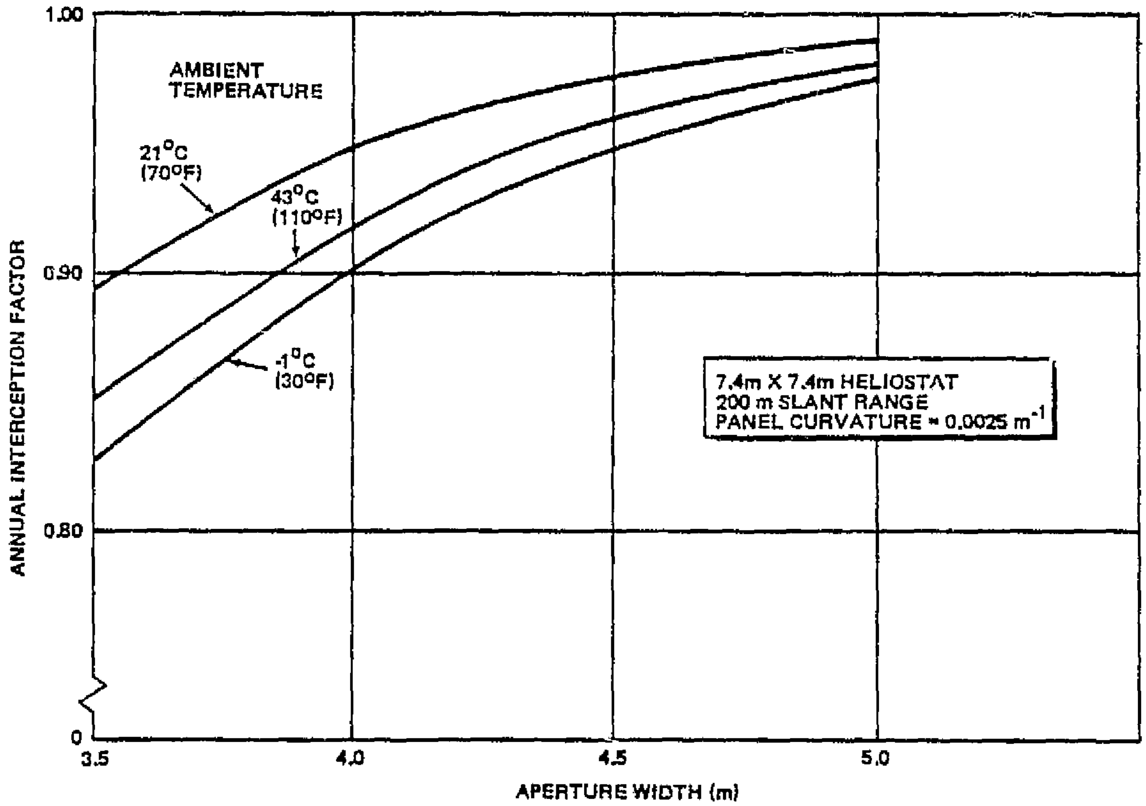


Figure 3.3-10. Temperature Effects on AIF (5)

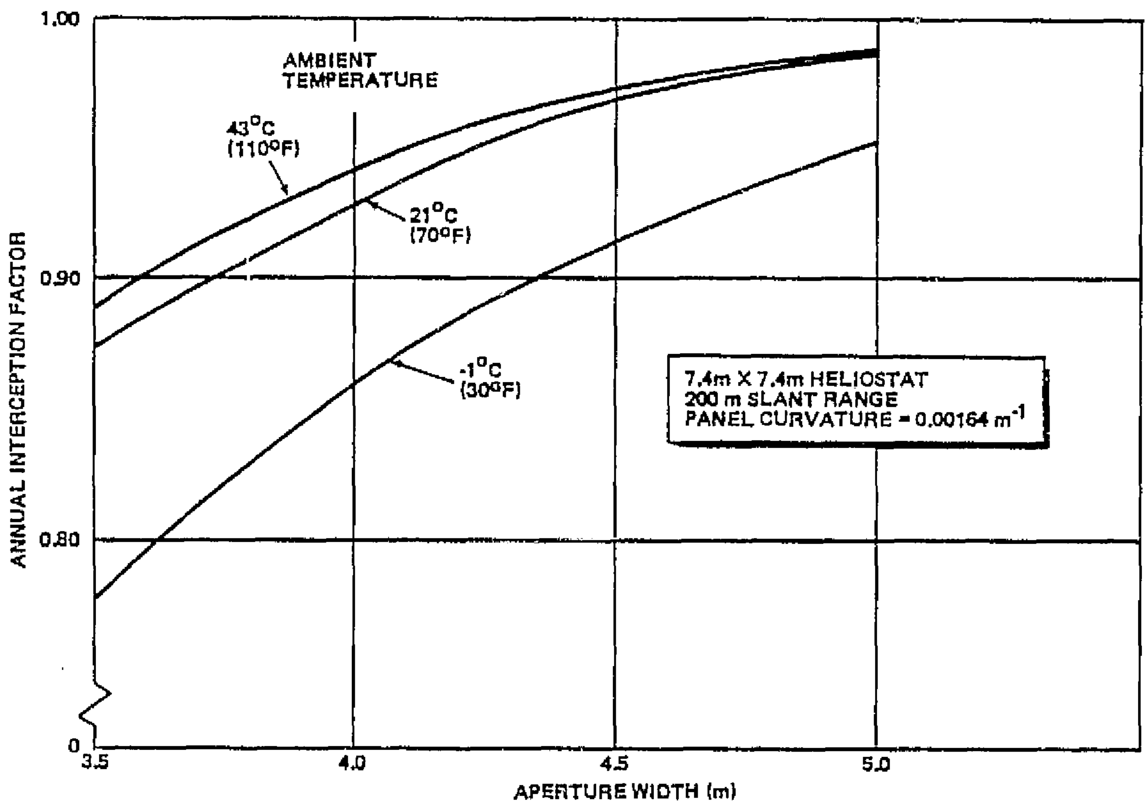


Figure 3.3-11. Temperature Effects on AIF (6)

larger heliostat mirrors (7.4 m x 7.4 m) are used at a slant range of 200 m, similar effects due to temperature variation to those with the 6.5 m heliostats are observed, with all curves displaced downward, as expected, due to the larger mirror size.

Location Effects — A preliminary analysis was made with the CONCEN code of the variation in annual interception factor with location in the north field. Heliostat mirror size 7.4 m x 7.4 m was assumed. The height of the receiver aperture center above the heliostat mirror center was taken to be 40 m. The receiver axis was assumed to be tilted downward 30° from horizontal, and the ambient temperature was 32°C (90°F). Table 3.3-4 contains values of the AIF for twelve locations in the collector field, for square receiver apertures of 4 m x 4 m and 4.5 m x 4.5 m. From this data a contour diagram giving the AIF contours throughout the field was developed. Figure 3.3-12 shows such a contour diagram for a receiver of aperture 4 m x 4 m. The departure of the contours from circular symmetry about the tower is due to the effects of average incident angle cosine reduction and off-axis aberration of the mirrors at large angles away from north.

Table 3.3-4
RECEIVER INTERCEPT FACTORS

Field Location			AIF	
N	E	Curvature	4 m Aper.	4.5 m Aper.
34.6 m	103.9 m	0.0025 m^{-1}	0.7943	0.8438
103.9	138.6	0.0025	0.8281	0.8814
138.6	103.9	0.0025	0.9156	0.9525
138.6	173.2	0.0025	0.6919	0.7650
173.2	0	0.0025	0.9662	0.9858
173.2	69.3	0.0025	0.9277	0.9625
173.2	138.6	0.00222	0.7917	0.8611
207.8	103.9	0.00212	0.8224	0.8904
242.5	0	0.00203	0.8367	0.9011
242.5	103.9	0.00187	0.7528	0.8317
311.8	0	0.00159	0.6642	0.7526
311.8	103.9	0.00151	0.6115	0.6996

3.3.2.3 Field Optimization Results

The results of the concentrator field optimization analysis carried out by the University of Houston are shown in Figure 3.3-13 for different tower heights and receiver aperture dimensions. The "figure of merit" parameter represents the capital cost divided by the annual thermal energy delivered to the base of the tower expressed in ($/MWHt per year). Cost factors considered include heliostats, land, wiring, tower, receiver, piping, pumps, and a fixed cost which is independent of the specific system under consideration.

The indicated values of the Figure of Merit were based on an insolation model defined by the University of Houston. This model predicts an insolation level which is 91.8 percent of the measured Barstow data when taken over a complete annual cycle. As a result, the predicted values of the Figure of Merit are about 8 percent higher than would be expected if the Barstow insolation model were used.

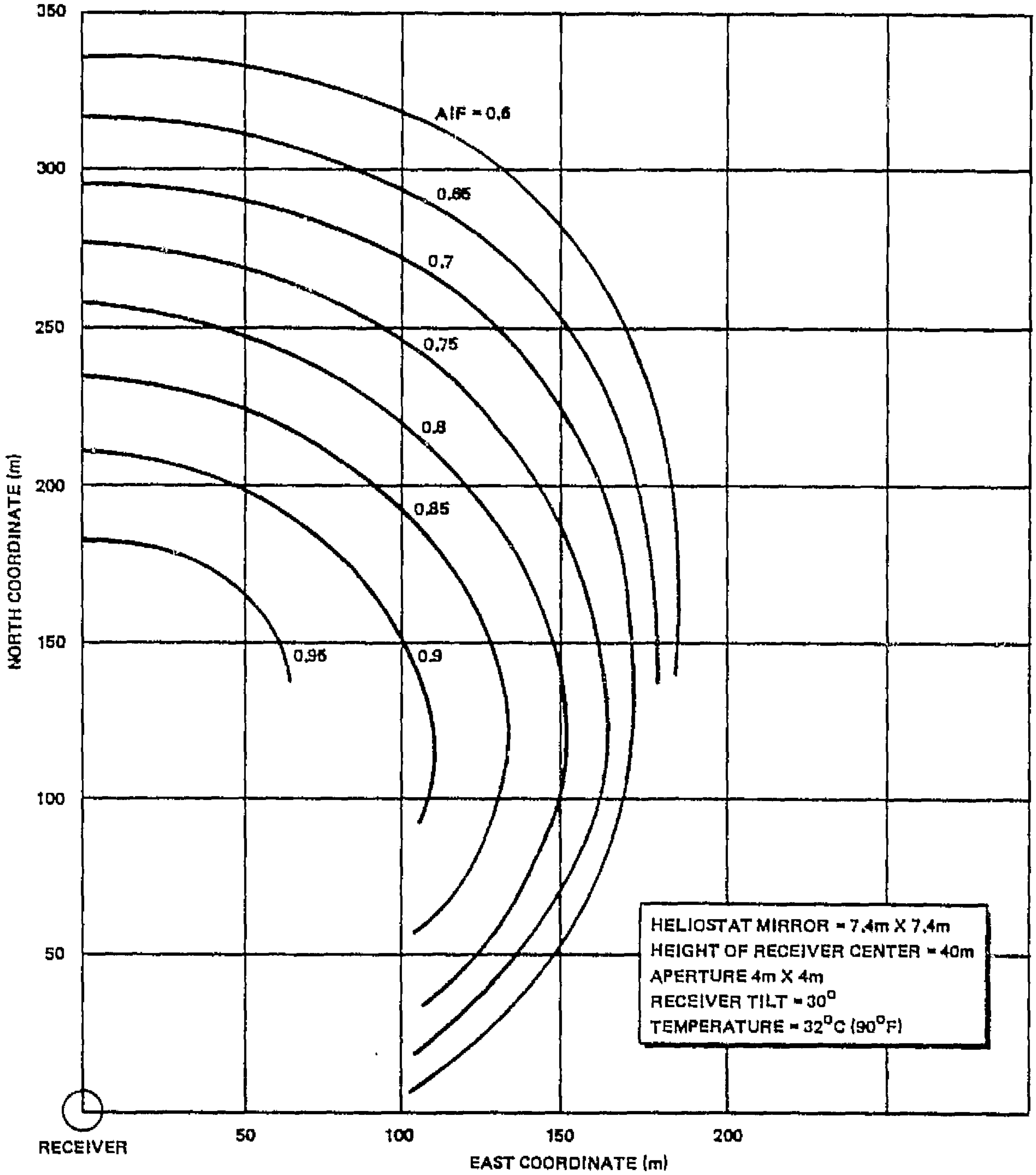


Figure 3.3-12. Annual Interception Factor Contour Diagram

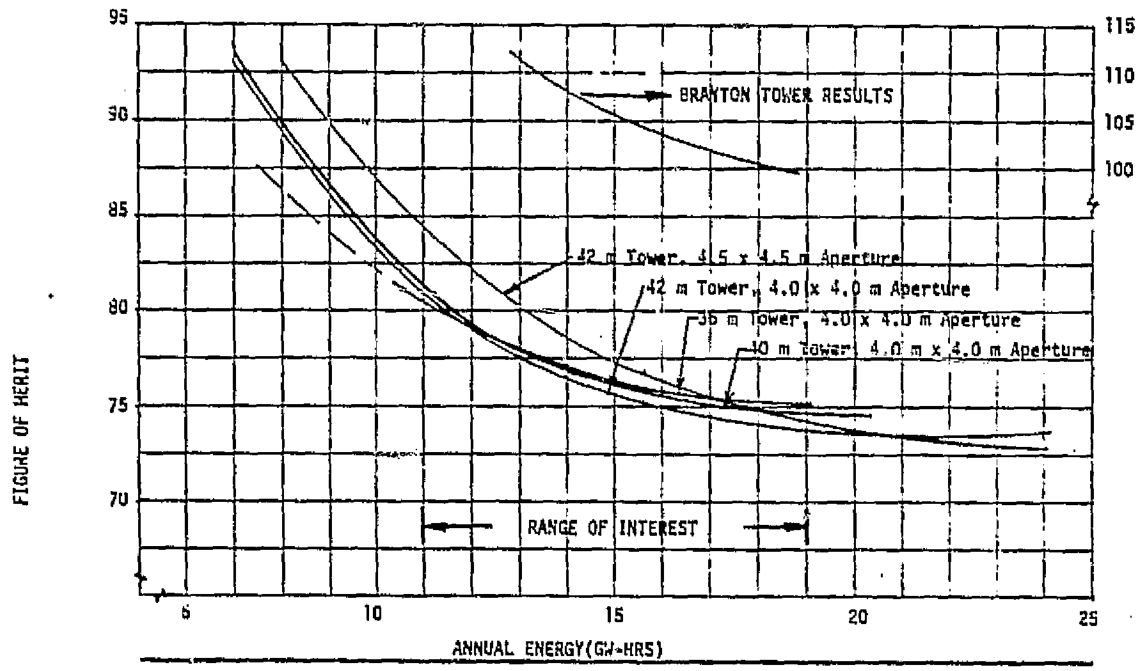


Figure 3.3-13. Collector Field Optimization Results

The indicated range of interest in annual energy is based on the thermal energy that would be required to produce an annual load factor of 0.4 for power conversion cycle efficiencies from 20-35 percent. This corresponds to annual thermal energy requirements from 19-11 GWHt, respectively.

The first study concerned the effect of aperture size on performance. For this purpose, several computer runs were made using a 42 m tower height with a 4.0 m x 4.0 m aperture and a 4.5 m x 4.5 m aperture. The results illustrated in Figure 3.3-13 show that the 4.0 m x 4.0 m aperture provides less expensive energy in the range of interest.

A second study was made to determine the impact of reducing tower height, by using a 4.0 m x 4.0 m aperture with a 40 m and 36 m tower height. The 40 m tower is almost identical to the 42m tower at the lower annual energy levels, but becomes more costly as annual power increases. The 36 m tower shows some potential for lower cost energy at low power levels and slightly higher cost energy above the 12 GW-hr annual energy level.

A third study was made to evaluate the effect the elevated receiver temperatures representative of gas turbines would have on collector field performance. The receiver loss model was altered to represent a 3.7 m x 3.7 m aperture operating at 815°C and a solar absorption coefficient of 0.99. Tower cost was increased to $100,000 due to increased loading and receiver cost was estimated at $250,000. Results show a 30-40 percent increase in the Figure of Merit (cost of energy) for the gas turbine system.

The Rankine cycle results show that energy cost is not highly dependent on tower height near the optimum height and that a single tower/receiver size can cover a substantial range of power levels.

A summary of the results for the various tower heights with a 4.0 x 4.0 m aperture is given in Table 3.3-5.

Table 3.3-5
SUMMARY OF COLLECTOR FIELD OPTIMIZATION RESULTS

Tower Height (m)	Number of Heliostats	Annual Energy (GW-Hrs)	Power at Equinox Noon (MW)	Figures of Merit	Efficiency
Rankine Cycle					
36m	151	10.85	4.83	81.38	0.562
36m	192	14.06	6.14	77.00	0.573
40m	158	11.70	5.13	79.83	0.579
40m	203	14.90	6.52	76.16	0.574
40m	235	17.20	7.46	74.97	0.573
42m	211	15.70	6.87	75.04	0.582
42m	253	18.75	8.11	73.54	0.580
42m	301	21.89	9.38	73.31	0.569
Brayton Cycle					
42m	222	12.46	6.095	113.55	0.439
42m	258	15.09	7.168	105.38	0.456
42m	310	18.50	8.55	99.35	0.467

3.3.3 Performance Potential of Alternate Cycles and Fluids

In defining the candidate systems for each of the three startup periods, it is desirable to maximize the overall cycle efficiency subject to existing heat transfer fluid and hardware constraints. In order to properly match the two into an optimized system, it is necessary to understand the capabilities and limitations of each.

From a heat transfer fluid standpoint, the operating temperature levels of each fluid along with corresponding degradation rates and fluid makeup costs are the single greatest factors in establishing the overall system design. Key thermodynamic fluid properties which are important in setting specific design requirements for individual hardware items include heat capacity, thermal conductivity, viscosity, vapor pressure, and freeze point. Details related to the applicability and limitations of various candidate fluids along with potential degradation mechanisms which can influence system design are discussed in the next section.

From a hardware standpoint, the selection of candidate systems can be equally influenced by the limitations of existing equipment as well as any anticipated near-term hardware development. These factors are related to performance, materials, temperature limitations, and design flexibility. In defining the candidate systems, these factors must be combined with those of the heat transfer fluids to define viable systems. These factors will be treated in Section 3.3.3.2.

3.3.3.1 Collection Fluid Performance

The heat transfer fluid must remain chemically unchanged over its thermal operating range for long periods of time. This requires that it be chemically compatible with its physical environment, and that it be inert within itself, i.e., it should not decompose or react with itself. These properties can be termed thermochemical stability. The temperature operating range is limited by its thermochemical stability at high temperatures and by its increase in viscosity (or solidification) at low temperatures. Generally, organic heat transfer fluids have a relatively wide working temperature range which is dependent upon the molecular structure of the particular fluid. Inorganic heat transfer fluids tend to have more limited useful temperature ranges but have the advantage of exhibiting a smaller change in viscosity over the useful temperature range and of possessing higher thermochemical stability at high temperature.

The decomposition of organic fluids is similar to breaking a chain at its weakest link. The upper useful temperature limit depends upon the weakest bond or combination of bonds in the fluid molecule. Certain chemical structures are known to be more stable than others, such as Biphenyl and Terphenyl. However, prediction of lifetimes for these fluids based on chemical considererations is not reliable. Experimental life tests are thus required to allow performance predictions to be made when these materials are incorporated in engineering designs.

Temperature Limits — A compendium of heat transfer fluid characteristics is maintained up to date at Rocketdyne. Data are added as it becomes available from in-house experiments as well as from efforts at various other laboratories. Commercially available heat transfer fluids and their conventional operating temperature ranges are shown in Table 3.3-6. The upper temperature limits to which a given fluid can be used is not an exact figure but depends upon the particular use intended and upon economic considerations involving cost of rejuvenation and replacement. Fluid degradation is a strong function of temperature near the upper limit of use but allows some flexibility depending upon how expensive the fluid is and upon what quantities are being used. Factors involved in these fluid degradations and cost considerations are discussed below in the next two subsections. The section immediately following discusses the important physical properties influencing performance for likely candidate heat transfer fluids.

Fluid Properties — The most important heat transfer fluid properties for a set of fluids are summarized in Table 3.3-7. In addition, several derived coefficients based on these properties are included to aid in evaluating their relative merits in light of the intended end use. Figures of merit are shown in Table 3.3-8. The volumetric heat capacity is given as a relative figure of merit of the ability of the fluid to act as a thermal storage media. Similarly, the heat transfer film coefficient has been calculated for typical expected end-use conditions. The higher the heat transfer film

Table 3.3-6

COMMERCIAL HEAT TRANSFER FLUIDS AND THEIR RECOMMENDED USABLE TEMPERATURE RANGES

FLUID NAME	USABLE TEMPERATURE RANGE, °F			
	LOW		HIGH	
	°C	°F	°C	°F
PARTHERM	249	480	593	1100
HITEC	149	300	538	1000
THERMINOL 88	145	293	399	750
SYLTHERM 800 (SILICONE B)	-40	-40	399	750
DOWTHERM A	12	54	388	659
THERMINOL 77	15.6	60	371	700
DOWTHERM G	-11	12	343	650
THERMINOL 66	-28	-18	343	650
MARLOTHERM L	-70	-94	350	662
MARLOTHERM S	-35	-31	350	662
CALORIA HT-43	- 9	15	316	600
MOBILTHERM 123	-18	0	316	600
SUN OIL 21	-18	0	316	600
THERMINOL 55	-40	-40	316	600

coefficient, the less costly will be the associated heat transfer equipment such as the receiver and the steam generator. Another important consideration is the relative pumping power required to transport a given quantity of energy from the receiver to the steam generator and any excess into thermal storage. The figure of merit shown in Table 3.3-8 for pumping power requirements depends upon the system configuration used. Generally, the required

Table 3.3-7

HEAT TRANSFER FLUID PROPERTIES OF SIX CANDIDATE FLUIDS

PROPERTY	HEAT TRANSFER FLUID					
	HITEC	DRAW SALT	SYLTHERM 800	THERMINOL 88	THERMINOL 66	CALORIA HT-43
SPECIFIC GRAVITY	1.77 @ 427°C (800°F)	1.82 @ 427°C (800°F)	0.67 @ 316°C (600°F)	0.88 @ 316°C (600°F)	0.825 @ 288°C (550°F)	0.654 @ 288°C (550°F)
VISCOSITY	1.49 cp @ 427°C (800°F)	1.65 cp @ 427°C (800°F)	0.29 cp @ 316°C (600°F)	0.33 cp @ 316°C (600°F)	0.45 cp @ 288°C (550°F)	0.516 cp @ 288°C (550°F)
THERMAL CONDUCTIVITY	$\frac{0.57 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{0.33 \text{ BTU/}}{\text{HR·FT·°F}}\right)$	$\frac{0.57 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{0.33 \text{ BTU/}}{\text{HR·FT·°F}}\right)$	$\frac{.116 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{.067 \text{ BTU/}}{\text{HR·FT·°F}}\right)$	$\frac{.114 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{.066 \text{ BTU/}}{\text{HR·FT·°F}}\right)$	$\frac{.0976 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{.0564 \text{ BTU/}}{\text{HR·FT·°F @ 550°F}}\right)$	$\frac{.085 \text{ WATTS}}{\text{M °K}}$ $\left(\frac{.049 \text{ BTU/}}{\text{HR·FT·°F @ 550°F}}\right)$
SPECIFIC HEAT	$\frac{1560 \text{ JOULES}}{\text{KG °K}}$ (.373 BTU/15°F)	$\frac{1548 \text{ JOULES}}{\text{KG °K}}$ (0.37 BTU/LB°F)	$\frac{2071 \text{ JOULES}}{\text{KG °K}}$ (.495 BTU/LB°F)	$\frac{2318 \text{ JOULES}}{\text{KG °K}}$ (.554 BTU/LB°F)	$\frac{2531 \text{ JOULES}}{\text{KG °K}}$ (0.605 BTU/LB°F)	$\frac{2845 \text{ JOULES}}{\text{KG °K}}$ (0.68 BTU/LB°F)
MELTING POINT	142°C (288°F)	221°C (430°F)	-40°C (-40°F)	60°C (140°F) TO 145°C (293°F)	-27.8°C (-18°F)*	-9.4°C (15°F)*
CORROSIVITY	MILD	MILD	VERY LOW	VERY LOW	VERY LOW	VERY LOW
TYPICAL TΔ	167°C (350°F)	167°C (350°F)	149°C (300°F)	149°C (300°F)	65.6°C (150°F)	65.6°C (150°F)

* POUR POINT

Table 3.3-8

FIGURES OF MERIT FOR SIX CANDIDATE HEAT TRANSFER FLUIDS

	HEAT TRANSFER FLUID					
	HITEC	DRAW SALT	SYLTHERM 800	THERMINOL 88	THERMINOL 66	CALORIA HT-43
VOLUMETRIC HEAT CAPACITY	$\frac{2760 \text{ KJOULES}}{M^3\ °K}$ $(41.2\ BTU/FT^3 °F)$	$\frac{2816 \text{ KJOULES}}{M^3\ °K}$ $(42\ BTU/FT^3 °F)$	$\frac{1388 \text{ KJOULES}}{M^3\ °K}$ $(20.7\ BTU/FT^3 °F)$	$\frac{2038 \text{ KJOULES}}{M^3\ °K}$ $(30.4\ BTU/FT^3 °F)$	$\frac{2085 \text{ KJOULES}}{M^3\ °K}$ $(31.1\ BTU/FT^3 °F)$	$\frac{1857 \text{ KJOULES}}{M^3\ °K}$ $(27.7\ BTU/FT^3 °F)$
HEAT TRANSFER FILM COEFFICIENT**	$\frac{5.1 \text{ KW}}{M^2\ °K}$ @ 427°C $\left(\frac{900 \text{ BTU}}{HR\ FT^2 °F}\right)$ @ 800°F	SIMILAR TO HITEC	$\frac{1.9 \text{ KW}}{M^2\ °K}$ @ 427°C $\left(\frac{340 \text{ BTU}}{HR\ FT^2 °F}\right)$ @ 800°F	$\frac{2.6 \text{ KW}}{M^2\ °K}$ @ 316°C $\left(\frac{450 \text{ BTU}}{HR\ FT^2 °F}\right)$ @ 600°F	$\frac{1.8 \text{ KW}}{M^2\ °K}$ @ 316°C $\left(\frac{320 \text{ BTU}}{HR\ FT^2 °F}\right)$ @ 600°F	$\frac{1.25 \text{ KW}}{M^2\ °K}$ @ 288°C $\left(\frac{220 \text{ BTU}}{HR\ FT^2 °F}\right)$ @ 550°F
PUMPING POWER FIGURE OF MERIT $(\rho\ C_p\ \Delta T)$*	231	236	100	146	75	66.7

* HIGH VALUES INDICATE LOW PUMPING POWER REQUIREMENTS

** CALCULATED AT 2.44 M/SEC (8 FT/SEC) FLUID VELOCITY pump power is inversely proportional to the value of the term $\rho C_p(\Delta T)$. Comparing the values of the three figures of merit from Table 3.3-8 shows that the molten salt heat transfer fluids have a distinct advantage over the others except in regard to the high melting point.

It should be noted that the properties of the organic type heat transfer fluids are not as well defined as those of the molten salts since these fluids are actually a mixture of various chemical compounds, sold commercially under brand names, with the exact composition known only to and controlled only by the manufacturer. In contrast, the molten salts are simple mixtures of inorganic compounds which are relatively pure and whose purity can be specified more easily.

It is important to have a high heat transfer film coefficient to minimize receiver heat transfer area. Additionally, the maximum fluid temperature can be close to the bulk temperature. This will result in a less rapid rate of degradation for a given heat transfer fluid bulk temperature, or the bulk temperature can be raised to a higher value resulting in an equivalent wall temperature but with the result that performance has been increased because the heat transfer fluid can be utilized over a greater differential temperature range between the cold side and the hot side of the system.

Fluid Thermal Degradation — Heat transfer fluid degradation can occur in three different forms: (1) degradation due to thermal effects only, i.e., excessively high temperatures, (2) decomposition due to catalytic effects in which surfaces of associated equipment or thermal storage media provide reaction sites, and (3) degradation due to chemical reaction with either a heat transfer storage media, the material of the equipment with which it comes in contact or with contaminants entering the system such as air or water. These three categories are discussed below.

All compounds eventually undergo degradation into smaller molecules at elevated temperatures and the real differences between substances are the temperatures and rates with which these degradations take place. This phenomena when applied to substances of polymeric form such as oils, is usually referred to as "cracking". Relatively long molecular chains are split into shorter chain components which are more volatile than the parent fluid and are usually soluble in the parent fluid. Subsequent to the formation of these lighter fractions, another mechanism can begin to take place, namely, the polymerization of the smaller fractions into longer molecules when the fluid is at lower temperatures. These excessively long chains are then prone to coat out on the walls or congeal into solid particles or materials that are not soluble in the parent fluid. Thus, the products are usually light, volatile fractions; however, they may subsequently turn into heavy polymers which must be filtered out or removed in some other manner. In general, the rate of degradation is an exponential function of temperature, and the useful upper limit for a given fluid is that point at which the rate of degradation is the maximum that is cost effective in the particular system when compared with the performance advantage of more elevated temperature.

The thermal decomposition rate is occasionally greatly accelerated because of the presence of a catalyst, usually consisting of a surface such as the inside of piping and equipment or the outside of thermal storage media particles. Not only can catalysts of this type accelerate the cracking rate, but they may also accelerate the rate of polymerization of the decomposition products or promote reaction with the storage media itself. An example of this is that certain organic type heat transfer fluids in the presence of copper surfaces causes polymerization to take place at an increased rate. In general, however, materials which cause heat transfer fluids to catalytically react have not been specifically searched for or identified. The presence of such catalysts are however indicated by the fact that the decomposition rate of the number of heat transfer fluids are influenced by the surface area of solid materials to which they are exposed during thermal degradation tests. For example, the rate of decomposition of Syltherm 800 is increased by a factor of two to three when solid materials such as iron ore are added.

Fluid Property Changes — Fluid degradation can also produce changes in the fluid properties. The degradation products may be soluble in the parent fluid causing it to increase its melting point, alter viscosity, (usually increasing it) and increase its tendency to coat surfaces. In some rare cases these substances further catalyze the degradation of the parent fluid.

In addition, the vapor pressure may increase and the toxicity or health hazard may be increased.

HITEC contains 40 percent of sodium nitrite, which tends to decompose forming sodium nitrate which causes the melting point to increase. In contrast, draw salt contains no sodium nitrite initially but has been observed to contain small percentages of this material after having been aged at high temperatures for relatively long periods. In addition, the melting point has been observed to decrease with increased aging.

In the case of Syltherm 800, it has been observed that although one of the byproducts of degradation is highly soluble in the parent fluid, this material will precipitate out at relatively cool local points in the system with the possibility of clogging it. Similarly, the heat transfer oils used at lower temperatures tend to form long chain polymers which may find their way into points of low fluid velocity in heat exchangers where they may form deposits blocking the flow of heat through the heat exchanger wall. Tests at Rocketdyne have found that the heat transfer fluid, Therminol 55, is particularly sensitive to this tendency.

In contrast, Caloria HT43 does not produce coatings on heated walls as long as a non-stagnant fluid condition is maintained, and provided the upper limit of film temperature is not exceeded. In addition, this fluid has a relatively low thermal degradation rate of 7% per year when utilized under the time-temperature history conditions expected in a solar power plant having an upper thermal storage temperature of 302°C (575°F).

When a heat transfer fluid cracks, causing short chain molecules to be formed from the longer chains, the vapor pressure of these shorter chains is much higher than that of the mother fluid and will cause the pressure in the system to rise unless it is adequately vented. Venting must be done in such a way as to conform with local waste disposal regulations.

In general, the toxicity of the heat transfer fluids has been considered to be low. However, the thermal degradation products have probably not been adequately evaluated in this regard. Consequently, appropriate safeguards for operating and maintenance personnel at these plants must be provided until such time as the relative toxicity of these various byproducts has been determined. A review of the existing data has revealed that the inorganic salts, HITEC and draw salt, undergo by far smaller changes as a function of time and temperature as compared to their organic counterparts.

Heat Transfer Fluid Costs — A comparison of anticipated costs of the heat transfer fluids is given in Table 3.3-9. The initial capital cost is determined and the equivalent influence on the plant capital cost in dollars per kilowatt is given as well as the increase in power cost in mills per kilowatt hour assuming an 18 percent annual charge for capital, operations, and maintenance (not including replenishment). Similar values are given for material required to replace the fluid lost by thermal degradation. It can be seen that the molten salts are the most cost effective and the silicone fluid is the most expensive. This is why Syltherm storage will be limited to the trickle charge configuration.

Safety — The greatest hazard in the case of the oil-type heat transfer fluids is that of fire caused by accidental exposure of the oil or its degradation products to air. Spontaneous combustion can take place easily even when small leaks occur while at operating temperature.

In contrast to this, no such danger is present when molten salts are used for the heat transfer fluid. Accidental, short term exposure to the air even at the highest operating temperatures creates no problem.

3.3.[illegible].2 Alternate Cycles and Fluids Evaluation

In comparing the alternate cycles with the heat transfer fluids, temperature is the single most important issue. Figure 3.3-14 presents an operational temperature range comparison between the heat transfer fluids discussed in the preceding section and the candidate power conversion equipment.

Table 3.3-9

COST OF HEAT TRANSFER FLUID TO STORE 13.75 MWHt

FLUID	ΔT = 177°C (350°F)		ΔT = 149°C (300°F)		ΔT = 65.5°C (150°F)	
	HITEC	DRAW SALT	SYLTHERM 800	THERMINOL 88	THERMINOL 66	CALORIA HT-43
PRICE IN $/LB	0.25	0.13	2.86	0.65	0.85	0.15
FLUID CAPITAL COST IN $/KWe	90	47	907	184	440	69
COST OF CAPITAL* FOR FLUID IN MILLS/KWHe	4.7	2.4	46.7	9.4	26	3.55
REPLENISHMENT COST IN $/YEAR	284	14	25,400	3,450	5,640	2,100
TOTAL COST IN MILLS/KWHe	4.8	2.4	53.1	10.3	28	4.16

* COST OF CAPITAL AT 18%/YR

ASSUMES 2-TANK SYSTEM (NO DUAL MEDIA)

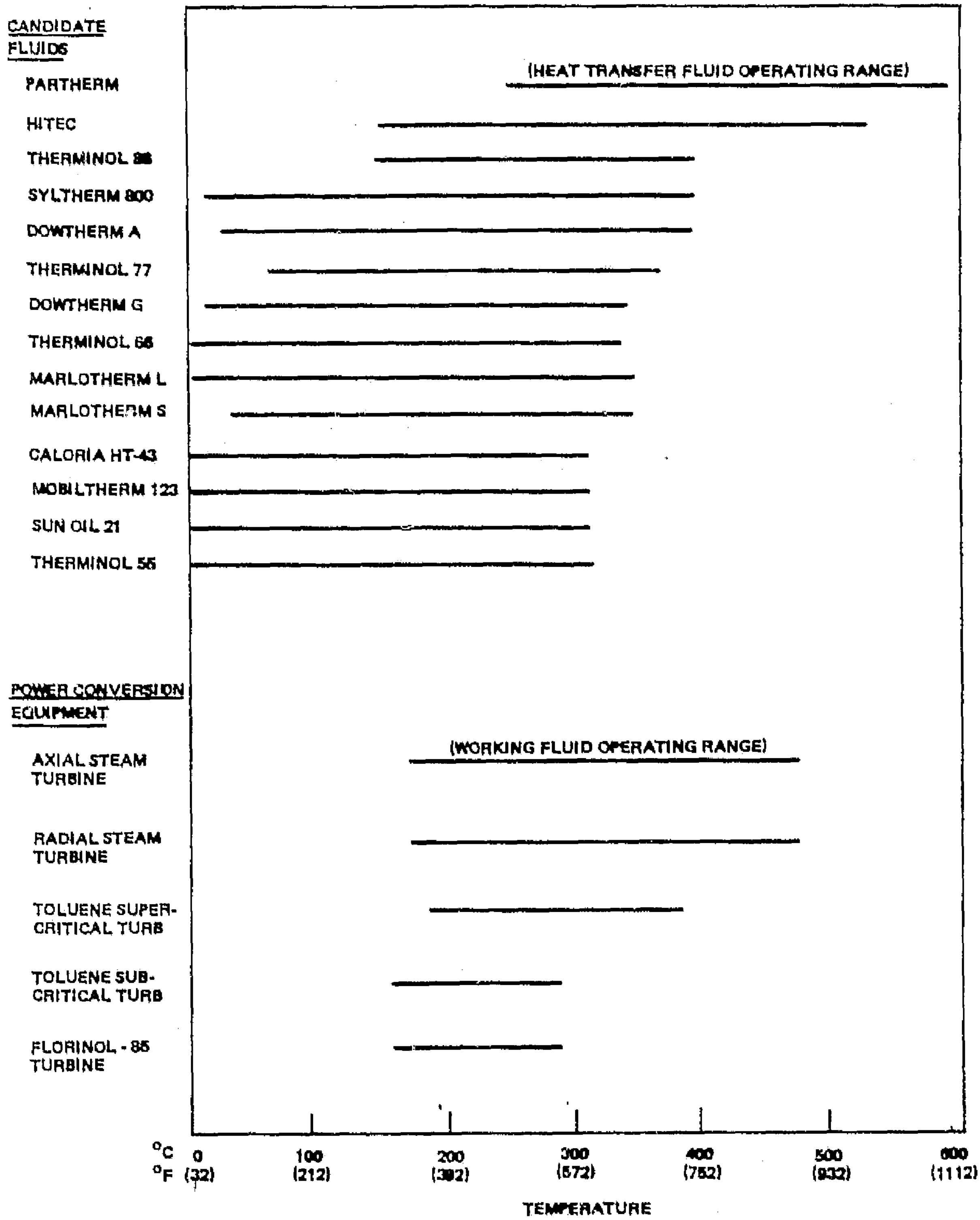


Figure 3.3-14. Compatibility Between Heat Transfer Fluids and the Power Conversion Equipment

For those combinations of heat transfer fluid and power conversion equipment in which the maximum equipment temperature exceeds that of the heat transfer fluid, the power conversion equipment would have to be operated at something less than its point of maximum efficiency. This, in general, would produce a lower cycle efficiency than could be realized with that power conversion equipment resulting in a higher system cost due to larger collector subsystem requirements.

In considering the steam cycle equipment (both axial and radial turbines), it is seen that Hitec or Partherm are the only fluids which have the temperature potential to maximize the performance of the conversion equipment. Use of any of the organic fluids as the heat transfer media would impact the performance of the water/steam cycle equipment. Some of the higher temperature organic fluids can be used with the organic Rankine cycle equipment and result in the full cycle efficiency potential being realized.

For combinations of heat transfer fluids and power conversion equipment in which the minimum heat transfer fluid temperature is greater than the minimum power conversion cycle fluid temperature, special design must be given to protecting the heat transfer fluid from exposure to excessively low temperatures. For Partherm and Hitec the minimum acceptable temperature shown corresponds to the freeze point. Through proper cycle design, these potential freeze problems can be eliminated through the use of additional regenerative heating stations or by utilizing thermal energy which completely bypasses the turbine.

The power conversion working fluid pressure, which can also significantly influence both the cycle efficiency and the design of the vapor generating equipment, must also be included in any compatibility considerations. In general, if temperature compatibility can be established, the desired power conversion subsystem operating pressure can be maintained by adjusting the lower temperature of the heat transfer fluid leaving the vapor generator based on pinch point considerations.

3.3.4 Receiver Configurations

Receiver configurations were defined for the candidate systems being evaluated.

The receiver is defined to include the surface which absorbs the sunlight redirected by the concentrator; the tubing which carries the heat transfer fluid from and to it's interfaces with the energy transport loop; the structure which supports the absorber; the valves, manifolds, sensors and flow control devices; insulation; weather protection and aerodynamic shrouds; and a terminal concentrator shroud, if used.

Four general receiver configurations were considered:

(1) Dual Zone in which a flat front panel of spaced tubes partially shade a back panel of packed tubes,

(2) Single Zone in which coolant traverses a flat panel in a serpentine manner,

(3) Single Zone with spiral tubes, and

(4) Cavities.

Sufficient information is generated for each configuration and fluid combination to permit an estimation of size, cost, thermal efficiency and pump power or pressure drop.

The general approach to establishing configurations is indicated in the following steps:

(1) Estimate the incident flux distribution, fluid routing, and flow rates.

(2) Estimate the fluid temperature at several stations along the tube by comparing the integrated flux along the tube to the total flux from inlet to outlet.

(3) Establish the required film coefficient at each station to prevent exceeding the maximum fluid film temperature allowed.

(4) Calculate the fluid flow speed, tube diameter, and number of parallel paths to provide the film coefficient with minimum pressure drop.

(5) With point-by-point heat balance, calculate temperatures and heat losses.
(6) Sum the heat losses to indicate receiver efficiency.
(7) Check receiver thermal stresses.
(8) Calculate weight and estimate aerodynamic loads.
(9) Prepare additional data as required to estimate costs of the receiver configuration.

Receiver configurations have been selected for each of the heat transfer fluids.

3.3.4.1 Receiver Requirements

The receiver configurations are required to operate with the fluids, flow rates, and requirements temperature of the concepts described in Table 3.2-13. These will be summarized, together with receiver characteristics, in Section 3.3.4.9.

The receiver efficiency should be optimized for cost/performance, considering the receiver cost and efficiency, the tower and energy transport loop costs, the concentrator costs, and the cost of providing parasitic pump power.

Environmental conditions include:

(1) Average operating wind speed of 3.6 m/s (8 mph) at a height of 10 meters.
(2) A survival wind speed of 40 m/s (90 mph) at a height of 10 meters.
(3) A neutral stability wind speed profile given by

$$\frac{V}{V_{10m}} = \left(\frac{Z}{10}\right)^{0.15}$$

where Z is the height of the receiver centerline above ground in meters.
(4) Seismic loads up to 0.25 g's at ground level.
(5) Temperatures from -30°C to +50°C at ground level.
(6) Hail up to 25 mm at 23/msec.
(7) Rain up to 25 mm/hour.

Operational requirements include:

(1) Provide for night-time heating or drawing of heat transfer fluids to prevent freezing (where required)
(2) Provide for preheat prior to morning start-up.
(3) Provide for rapid thermal response to prevent damage or out of range operation during cloud passage transients.
(4) Provide a means for draining at night or during periods of extended shutdown.
(5) Provide ready access for maintenance and repair.
(6) Maintenance and repair skills levels should be consistent with those likely to be available in a small community.

3.3.4.2 Receiver Flux Distributions

The effect on the distribution of radiation flux on the interior surface of the receiver, as the shape of the receiver cavity is varied, was determined by means of the CONCEN solar irradiation code. The flux density distribution due to radiation focused at the plane of the receiver aperture is shown in contour diagram form in Figure 3.3-15. Representative operating conditions were assumed, with a 150 heliostat collector field. With a 3.4m x 3.4m aperture, as shown, the peak-to-average flux density ratio is approximately 2.5. The spreading out of the midplane distribution at depths behind the aperture is shown in Figure 3.3-16.

Three receiver partial cavity configurations, all with apertures of 1.5m x 1.5x, were also analyzed. The flux density distribution along the cavity walls at a center horizontal section is given for the three cavity shapes in Figures 3.3-17 through 3.3-19. The cavity cross-section in each case is shown in the figures. The curves indicate that substantial discontinuities occur in flux density when the wall orientation changes abruptly, as at the junctions of flat walls. The simple conical or pyramidal shape, shown in Figure 3.3-19, exhibits a more uniform distribution than do the cavities with sides nearly parallel to the incident radiation. It is apparent that, by adjusting the shape of the cavity interior, nearly uniform flux density distribution can be obtained. The combined effects of the varying divergence of the focused beam beyond the aperture plane and the cosine of the incidence angle onto the surface may be employed to obtain the desired flux distribution.

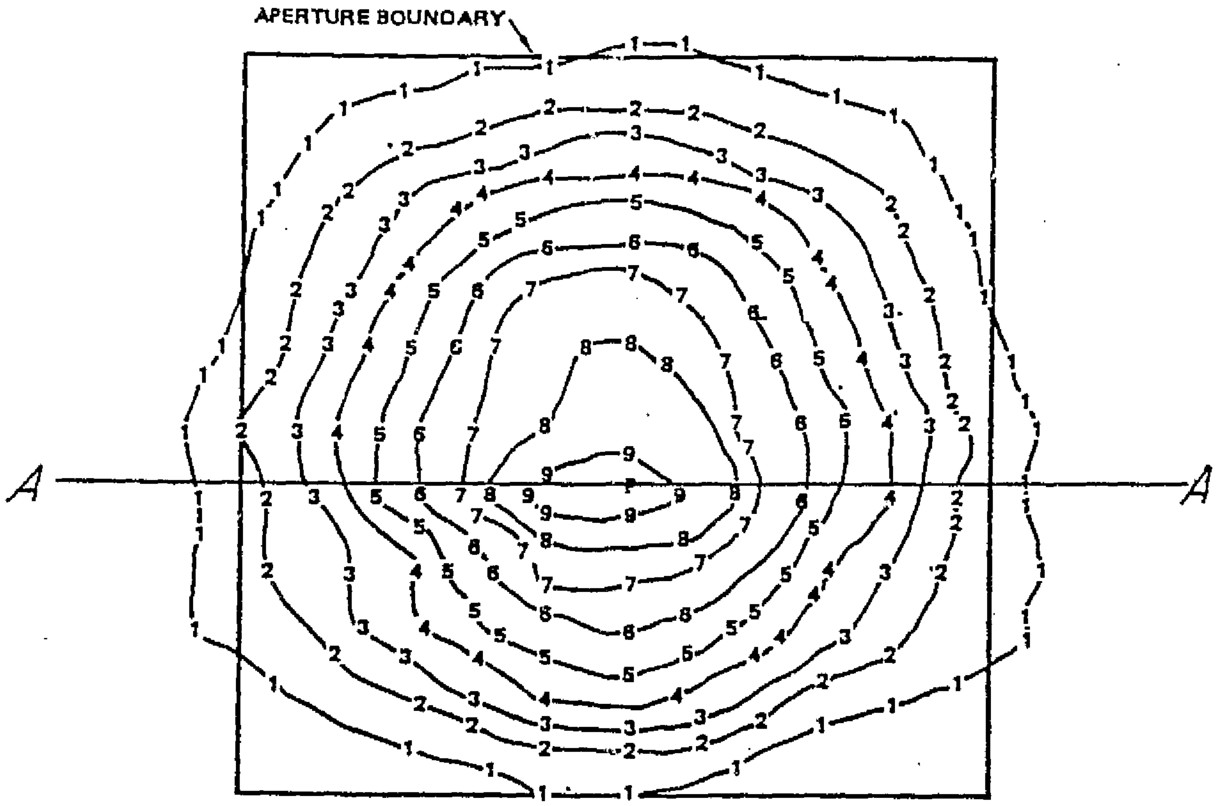


Figure 3.3-15. Receiver Flux Map

Given the flux distribution at the heat transfer surface, the fluid passages will be arranged to provide a favorable temperature profile along the path from the fluid inlet to the outlet. Generally, it will be advantageous to have the cold fluid enter the receiver at the region of peak heat flux, and for the flow path to traverse regions of successively lower heat flux as the fluid temperature increases.

3.3.4.3 Heat Transfer Correlations

Film coefficients were computed using the physical properties data listed in Table 3.3-7 and are shown on Figure 3.3-20 for Hitec, Syltherm and Caloria HT-43. The temperatures shown on the curves are average fluid temperatures, fixed by the requirements of the candidate power systems. Manufacturers' data indicate that convective heat transfer for any of these fluids, in turbulent flow, is correlated by the standard Colburn or Seider and Tate type equations. It can be seen that Hitec can provide a combination of (1) smaller film

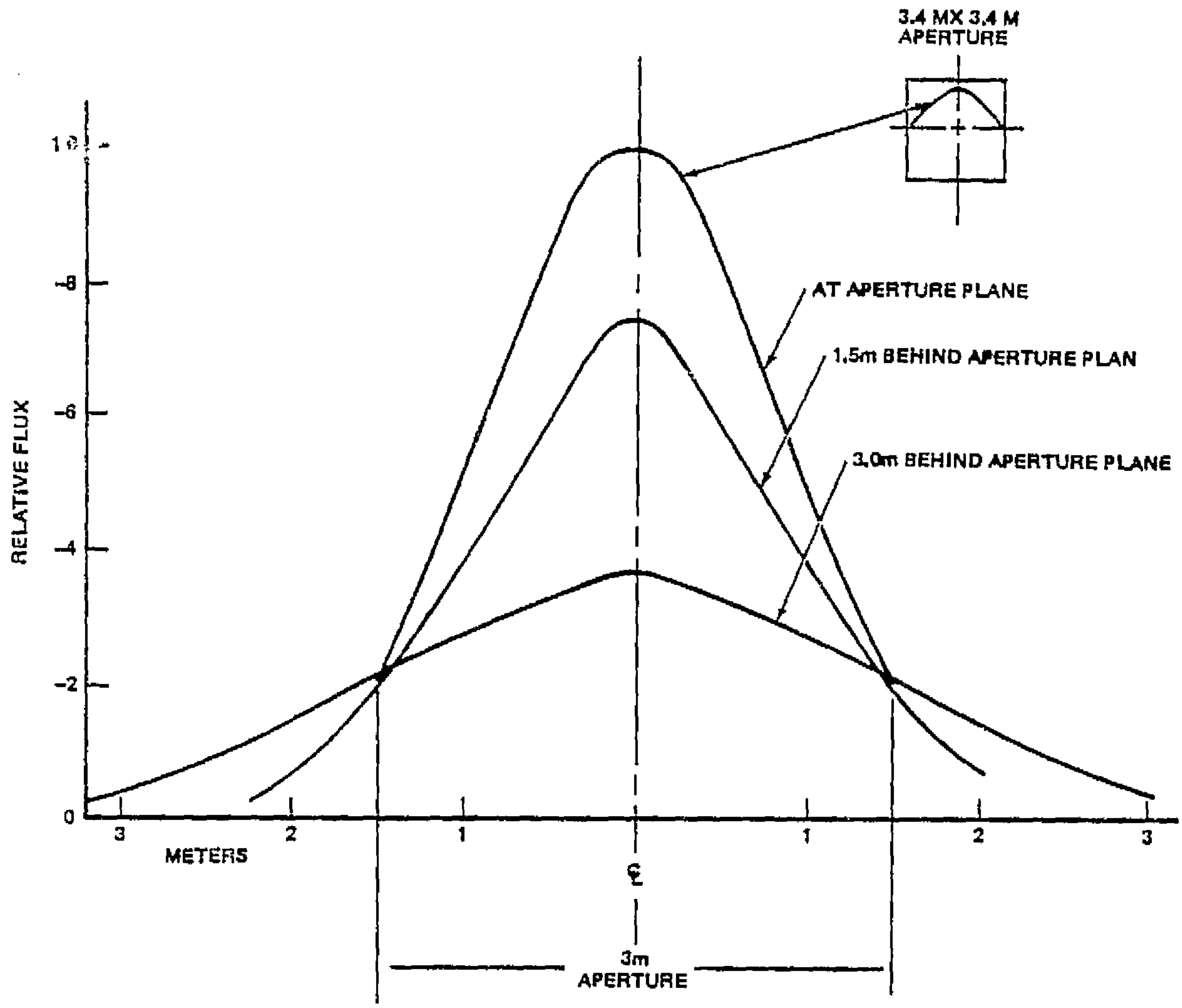


Figure 3.3-16. Receiver Flux; Horizontal Profile

temperature difference, (2) higher operating heat flux, (3) smaller area of heat transfer surface, relative to the other two fluids.

3.3.4.4 Two-Zone Receiver (50-50 Power Split)

The elements of the two-zone receiver concept are shown in Figures 3.3-21 through 3.3-23. The heat transfer surface (tube assembly containing the receiver fluid) is arranged in two planar zones at the receiver aperture as shown in Figure 3.3-21. Zone 1, a row of spaced vertical tubes, absorbs about

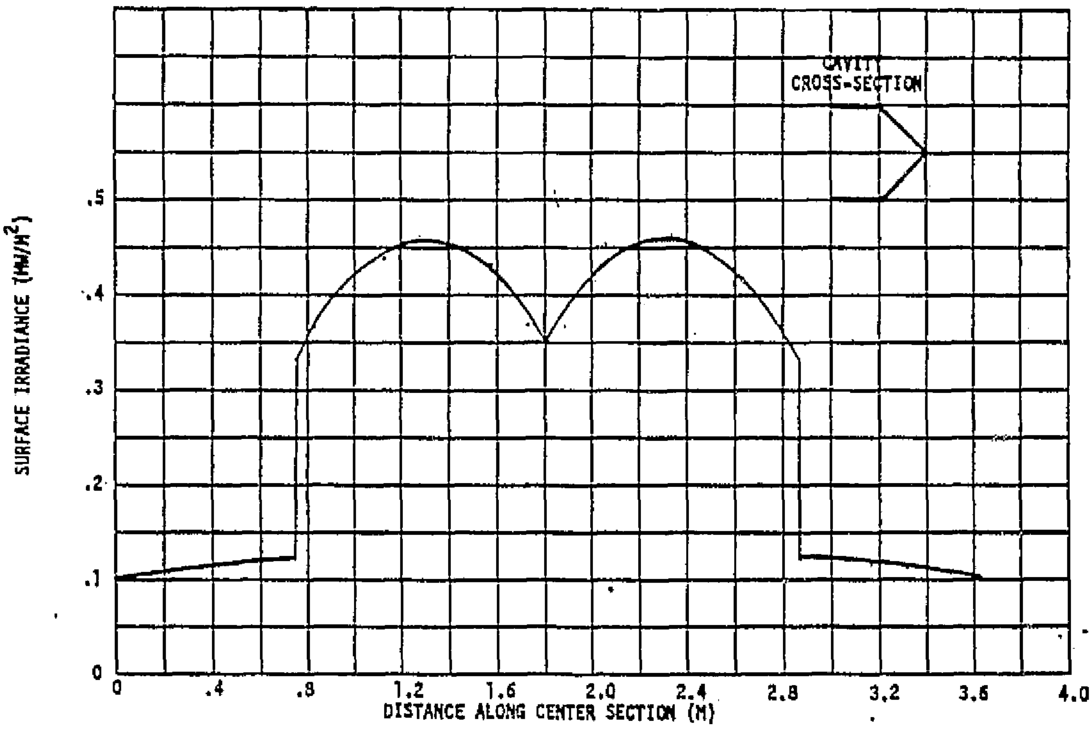


Figure 3.3-17. Flux Density Distribution on Cavity Receiver

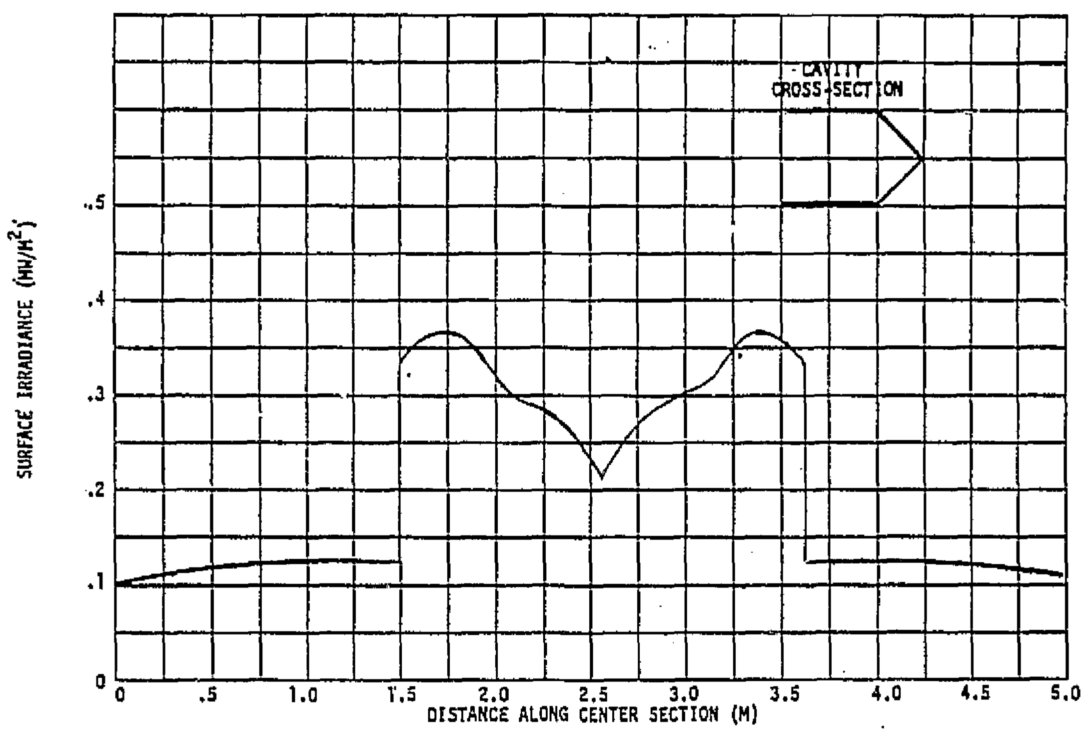


Figure 3.3-18. Flux Density Distribution on Cavity Receiver

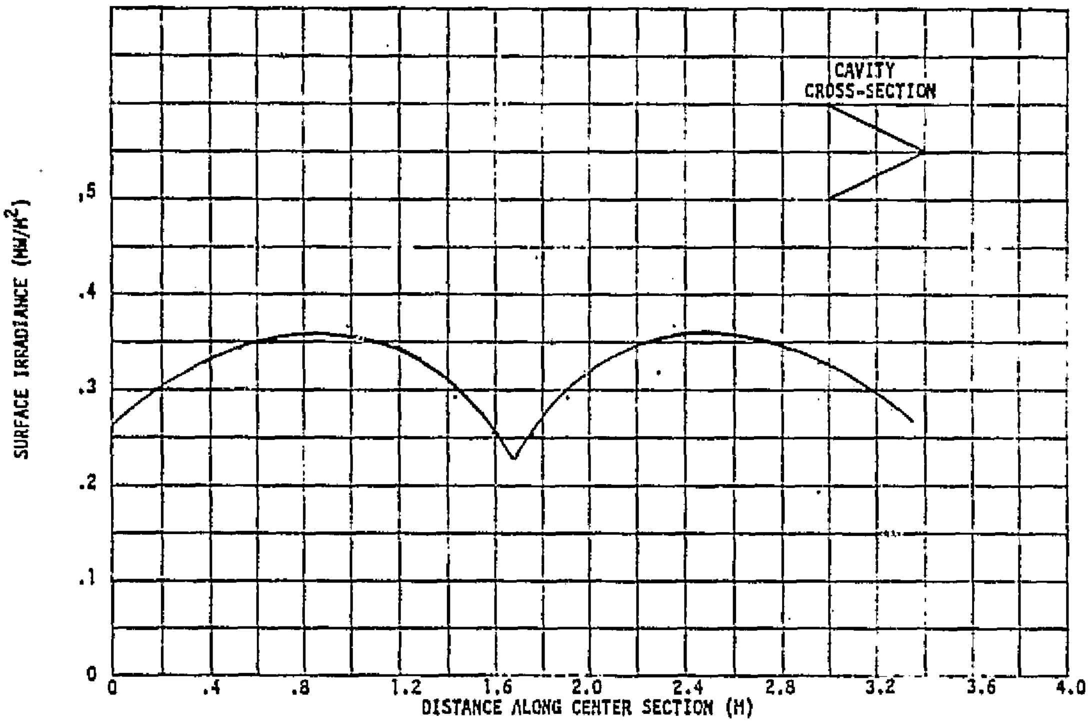


Figure 3.3-19. Flux Density Distribution on Cavity Receiver

50 percent of the incident energy. Zone 2, a row of close-packed horizontal tubes located behind Zone 1, absorbes the remaining energy. The magnitude and distribution of the Zone 2 flux can be adjusted by varying the spacing between tubes in Zone 1 and the distance between Zones 1 and 2.

Figure 3.3-22 is a sketch of the two-zone receiver configuration. The Zones 1 and 2 pipe, the reflector, thermal insulation, support structure, siding, and insulated doors are indicated in Views A-A and B-B. The side view shows the receiver atop the tower. The aperture faces North, and is tilted downward 30° off-vertical. Figure 3.3-23 is a schematic of the fluid flow paths through the receiver. The reflector, Zone 1, and Zone 2 piping are series-connected and there are two parallel circuits through the receiver. Each circuit may consist of several small diameter tubes in parallel. The fluid flow in Zones 1 and 2 is from the center (region of highest heat flux) to the edges of the aperture, in vertical and horizontal serpentine paths.

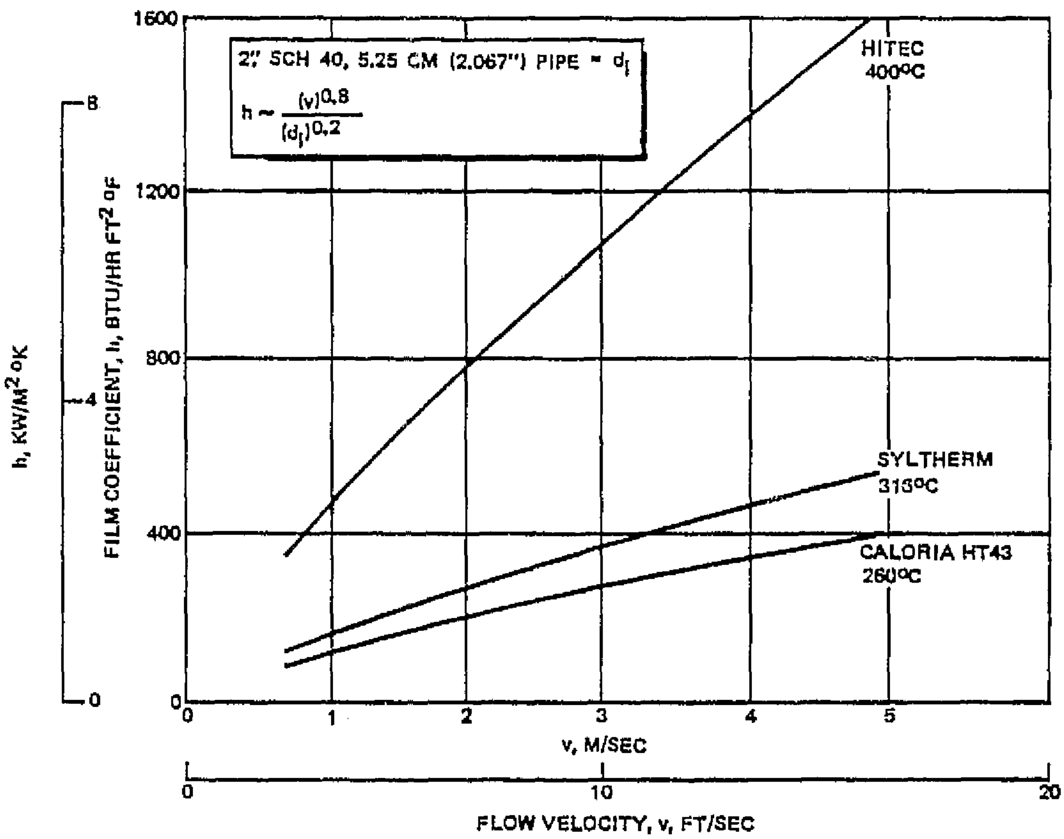


Figure 3.3-20. Heat Transfer Coefficient vs. Velocity for Hitec, Sylthern and Caloria HT 43

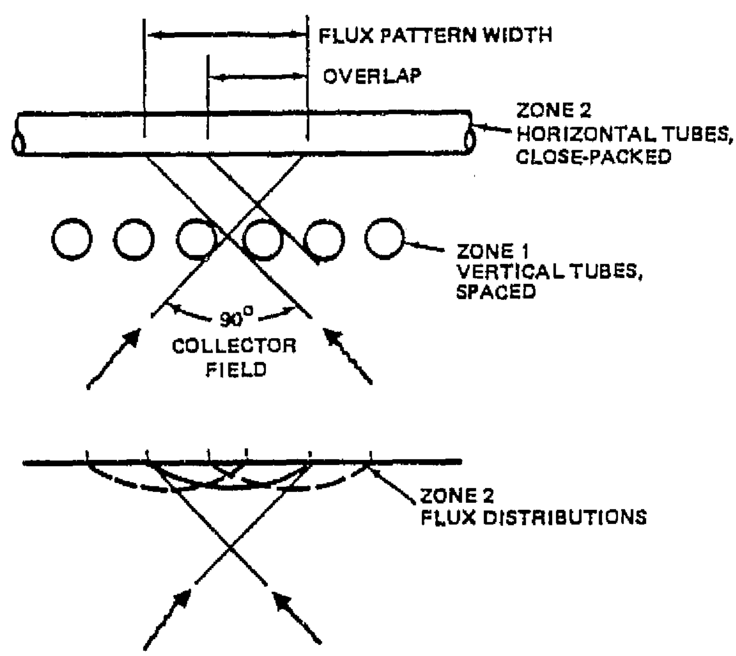


Figure 3.3-21. Two-Zone Receiver Heat-Absorbing Surface Geometry

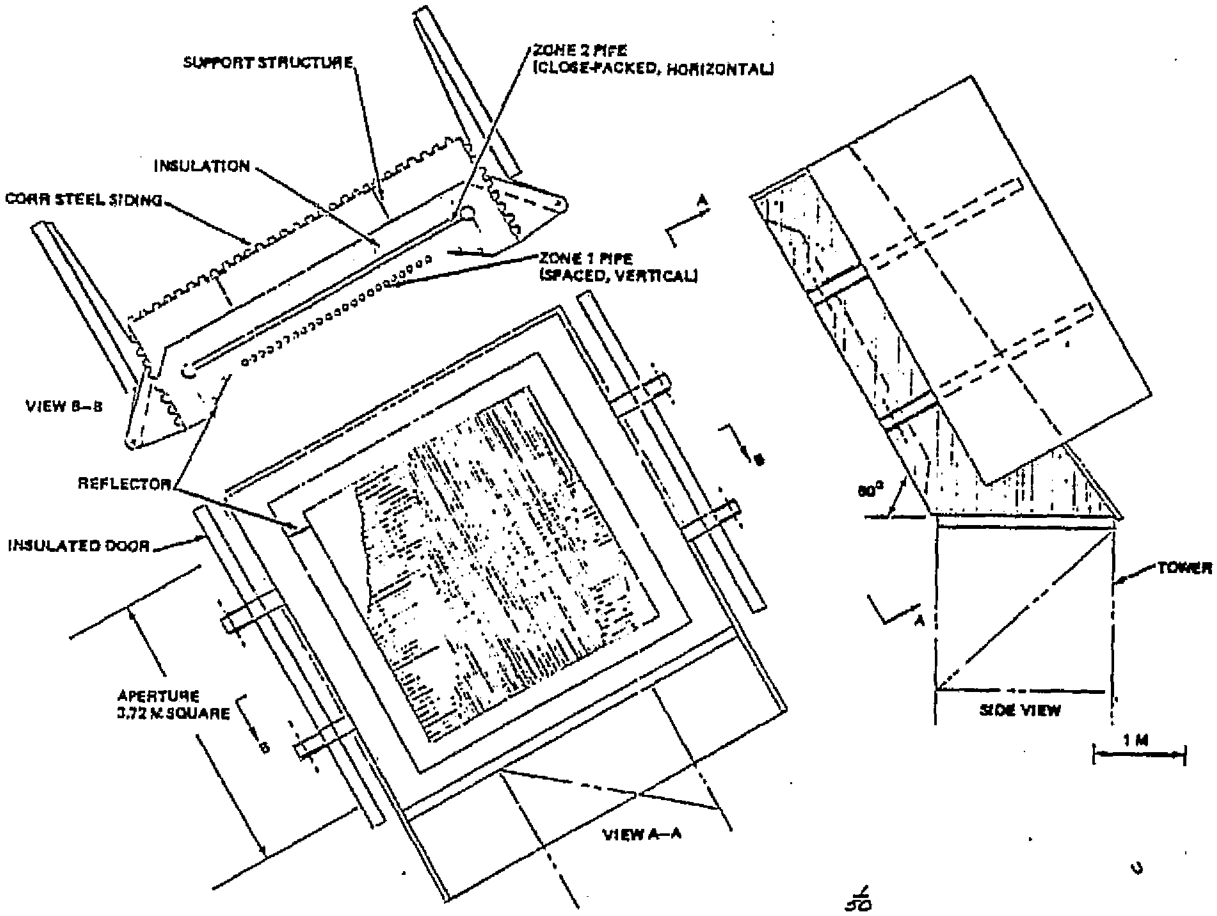


Figure 3.3-22. Two-Zone Receiver, 4.83 MWT, HTS

Briefly, the bases for the two-zone configuration are:

(1) A narrow specularly-reflecting surface surrounding the aperture shields the adjacent structure and reflects the incident energy into, but near the edges of, the aperture. It reduces losses by making possible a smaller high-temperature receiver area for the same absorbed power.

(2) The two-zone arrangement permits operation at a reduced average heat flux and provides a favorable flux distribution over the heat transfer surface.

(3) The high temperature piping (Zone 2) is partially shielded by the Zone 1 piping. (This will reduce the radiation, and probably the convection losses, from the heat transfer surface.)

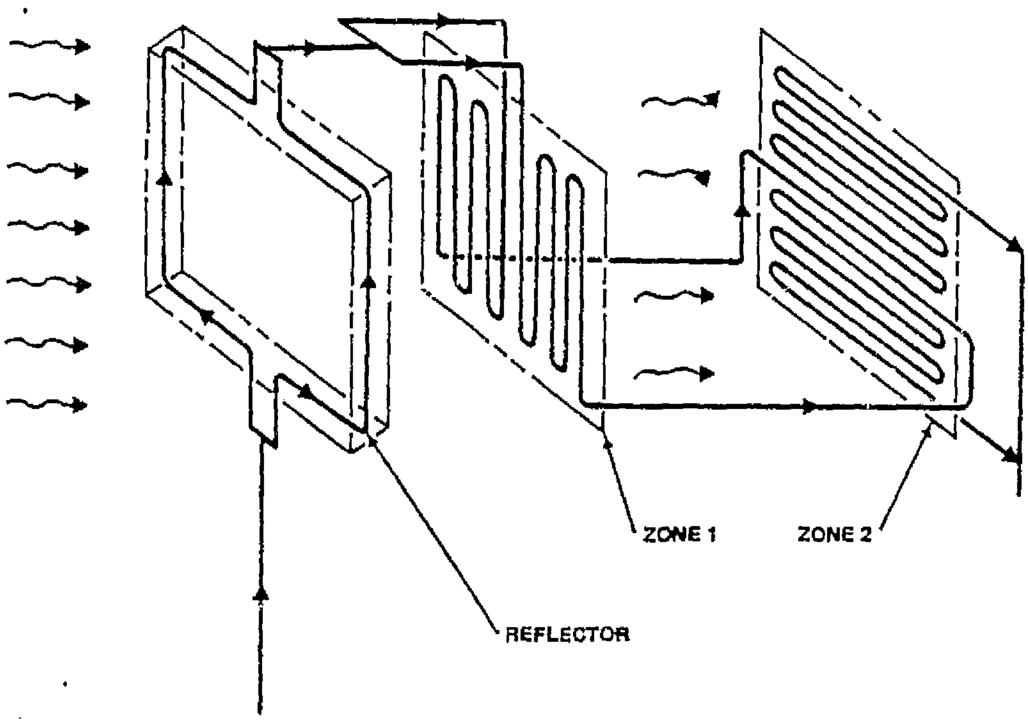


Figure 3.3-23. Tow-Zone Receiver Fluid Flow Paths

Figure 3.3-24, which shows the peak heat flux profile along the fluid flow path for the two-zone receiver, is derived from the receiver flux map (Figure 3.3-15) together with the coolant flow path (serpentine, from aperture center to aperture edge) and a particular division of power between Zones 1 and 2.

Figure 3.3-25 shows the fluid bulk temperature profiles and film temperature limits for two-zone receivers cooled by HTS, Syltherm and Caloria HT-43. The fluid inlet and outlet temperatures are fixed by the requirements of the power conversion systems. The film temperature limits are the fluid supplier's recommended maxima.

For a specified peak thermal power to the receiver aperture, the aperture area may be the minimum set by the optics of the collector field, or a larger area may be required, - depending upon the cooling efficiency of the receiver fluid. This is illustrated in Figure 3.3-26, which shows the relationship between heat transfer film coefficient and aperture size for three

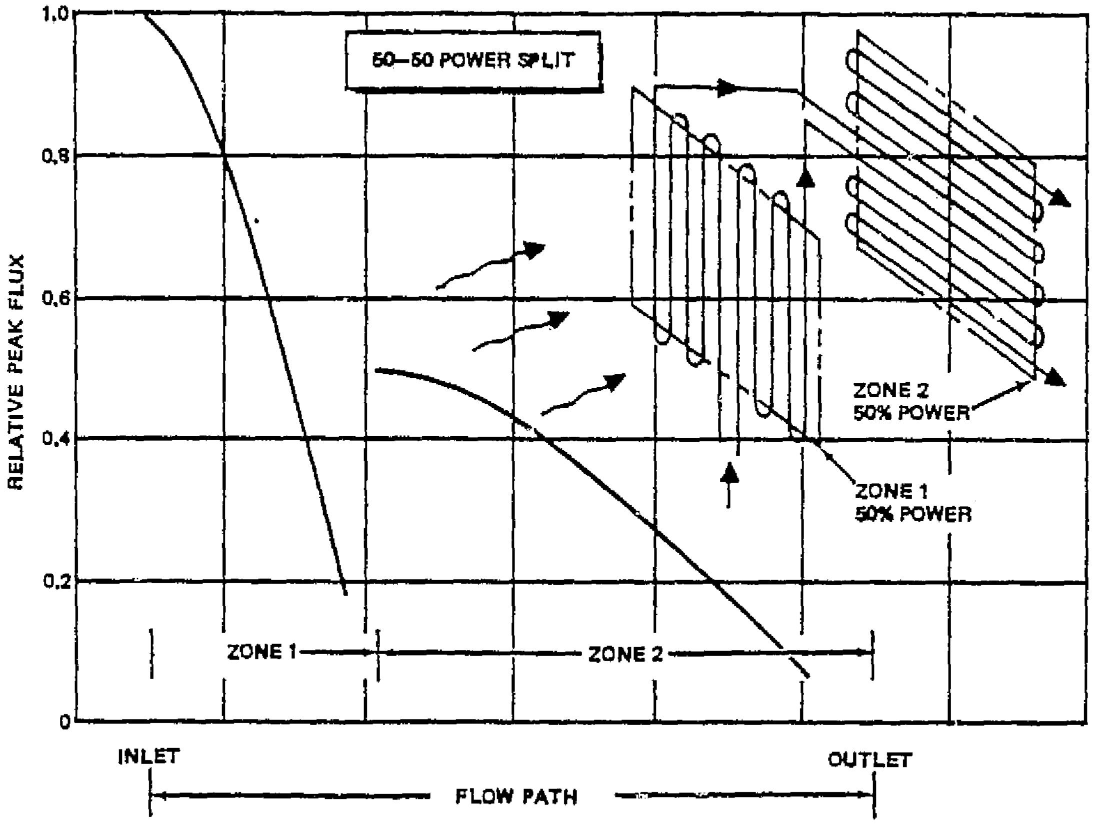


Figure 3.3-24. Reference Heat Flux Profile – Serpentine Flow Path (Two Zones)

combinations of fluid and receiver thermal power. The required peak thermal power is fixed by the system performance requirements (electrical output, load factor, insolation model) and by the power cycle efficiency. The thermal power values shown on the figure are near the minimum for these particular coolants at the operating temperature levels of Figure 3.3-25.

From Figure 3.3-20, the film coefficients for Caloria and Syltherm are in the range of 400 to 600 Btu/hr ft^2 •F, for "reasonable" fluid velocities (i.e., < 6 m/sec). Figure 3.3-26 shows that two-zone receiver systems using these fluids will require heat transfer surface areas of about two times the field optics-limited size. For a minimum area HTS receiver, the required film coefficient is about 900, which corresponds to a flow velocity of about 3 m/sec.

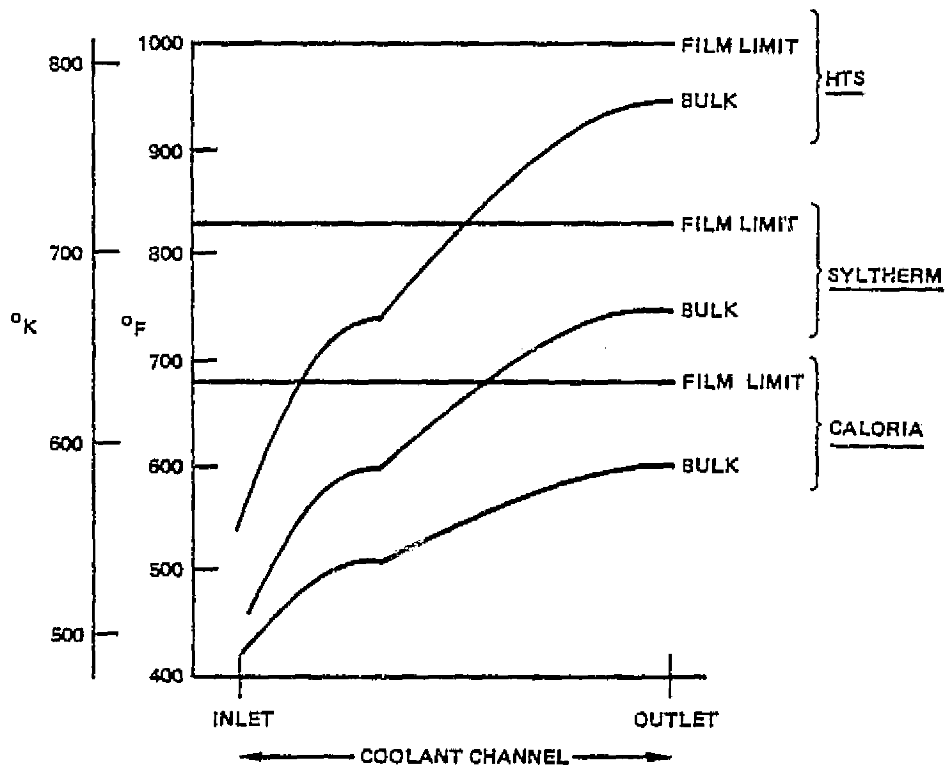


Figure 3.3-25. Two-Zone Receivers Fluid Temperature Profiles

HTS Design — Figure 3.3-27 shows bulk fluid and film temperature profiles for the HTS receiver. The "hot spot" (i.e., point of maximum film temperature) is located in Zone 2 near the outlet. The shape of the curves of Figure 3.3-27 are independent of the receiver power.

Parametric sizing data for the heat transfer surface of the minimum-aperture, 4.83 MWt, HTS-cooled, two-zone receiver are shown in Figure 3.3-28. The curves show the variation of fluid velocity, pressure drop and number of parallel flow paths with tube inside diameter. The pressure drop (and hence parasitic pumping power) decreases with decrease in tube diameter, but at the expense of increased complexity in the tube assembly (more tubes, more parallel paths). The design point shown in the figure [38 mm (1-1/2 inch) O.D. 16 Ga tubing] has six parallel flow paths and a pressure drop of (60 psi) 4 N/m^2.

The estimated total weight of the two-zone HTS receiver is 7090 Kg (15,598 lbs) as shown in Table 3.3-10. Receiver data including absorbed power, fluid flow, operating temperatures, losses and efficiency are listed in Section 3.3.4.9.

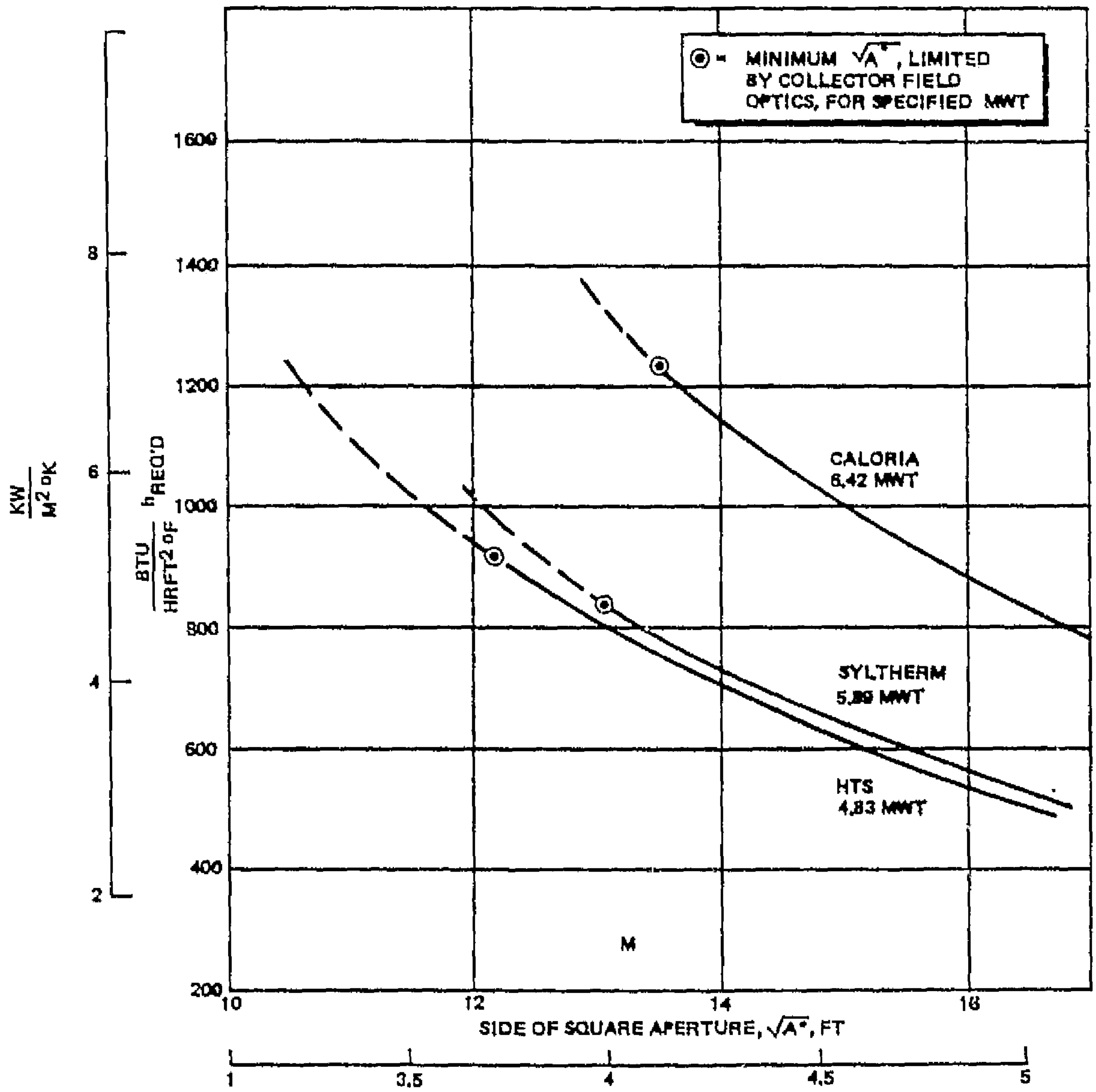


Figure 3.3-26. Required Film Coefficient Versus Aperture Size 1 MWT Z-Zone Receiver, 50/50 Power Split

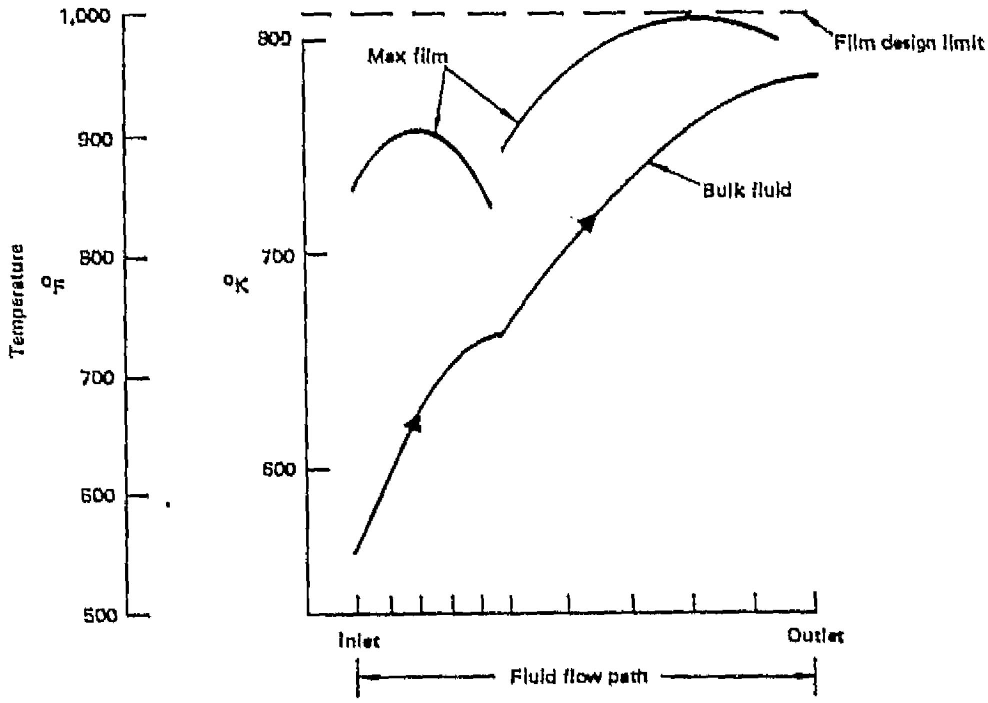


Figure 3.3-27. HTS Temperature Profiles — Two-Zone 50/50 Power Split

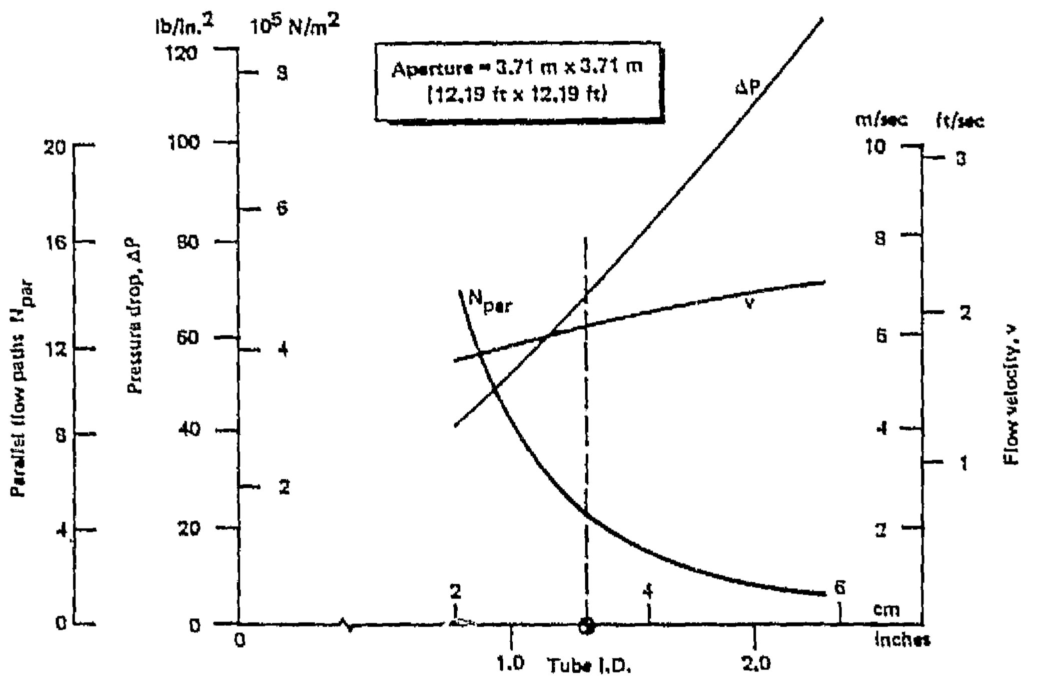


Figure 3.3-28. 4.83-MWt Receiver — HTS Fluid, Two-Zone

Table 3.3-10

TWO-ZONE HTS RECEIVER WEIGHT

Item	Weight kg (lb)
Housing/Support Structure	2,320 (5,104)
Reflector	450 (990)
Zone 1 Pipe	410 (902)
Zone 2 Pipe	1,410 (3,102)
Interconnecting Piping	230 (506)
Insulation	1,000 (2,200)
Doors	1,270 (2,794)
Total	7,090 (15,598)

Syltherm Design — Figures 3.3-29 and 3.3-30 show the corresponding temperature profiles and parametric sizing data for the two-zone receiver using Syltherm fluid. Receiver data are listed in Section 3.3.4.9.

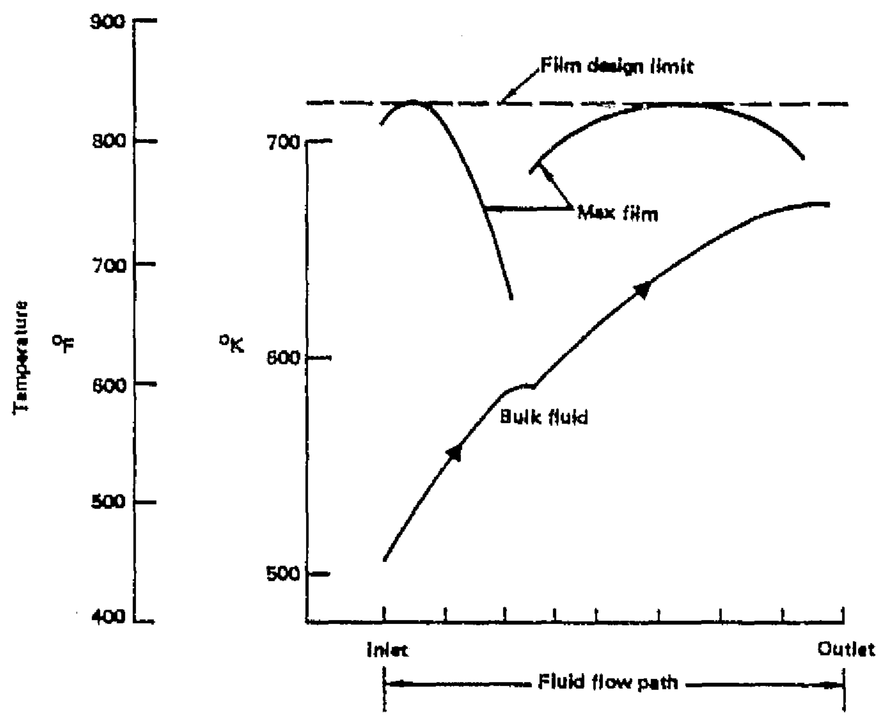


Figure 3.3-29. Syltherm Temperature Profiles — Two-Zone 50/50 Power Split

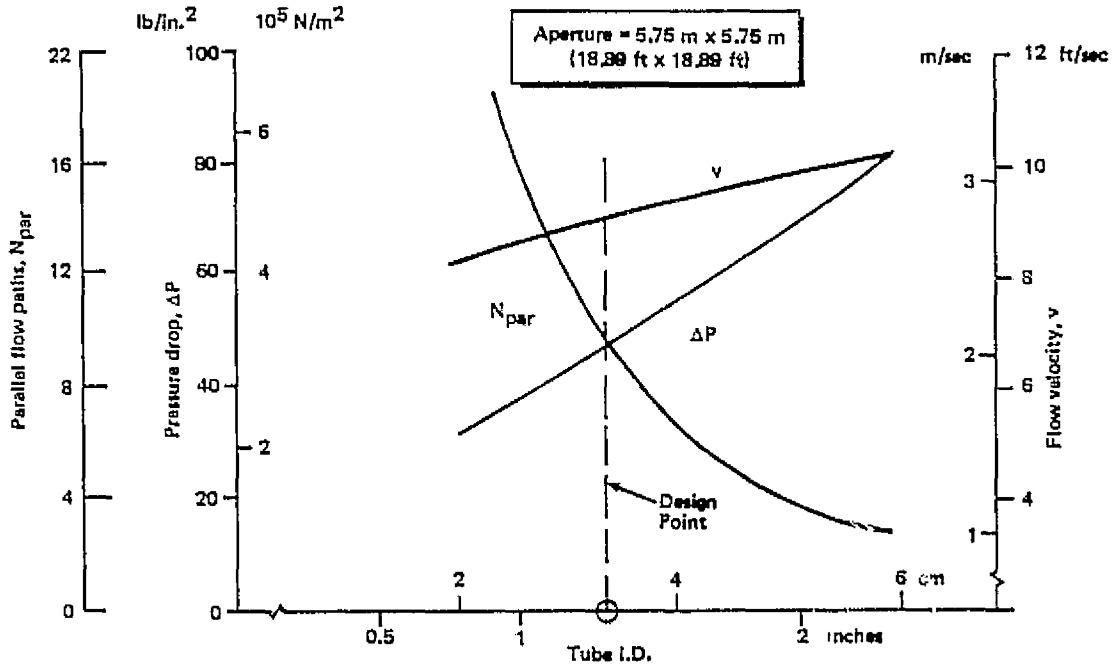


Figure 3.3-30. 5.89-MWt Receiver — Syltherm Fluid, Two-Zone

3.3.4.5 Single Zone Receiver - Serpentine

A sketch of a single-zone receiver configuration with a serpentine tubing arrangement, is shown on Figure 3.3-31. The heat transfer surface is composed of two sets of closely packed horizontal tubes arranged in a serpentine manner in one plane (single zone) at the receiver aperture. Fluid flow enters each tube set at the receiver horizontal centerline and flows horizontally across the panel in a serpentine manner. One tubing set exits at the top of the panel. The other set exits at the bottom of the panel. The horizontal routing was selected to aid receiver draining.

The heat flux profile along the fluid flow from inlet to outlet is also shown on Figure 3.3-32. Bulk fluid and film temperature profiles for this receiver configuration using Hitec, are shown on Figure 3.3-33. As indicated on this figure, the "hot spot" (point of maximum film temperature) is located near the outlets of the fluid flow path (near the upper and lower boundaries of the tubing configuration).

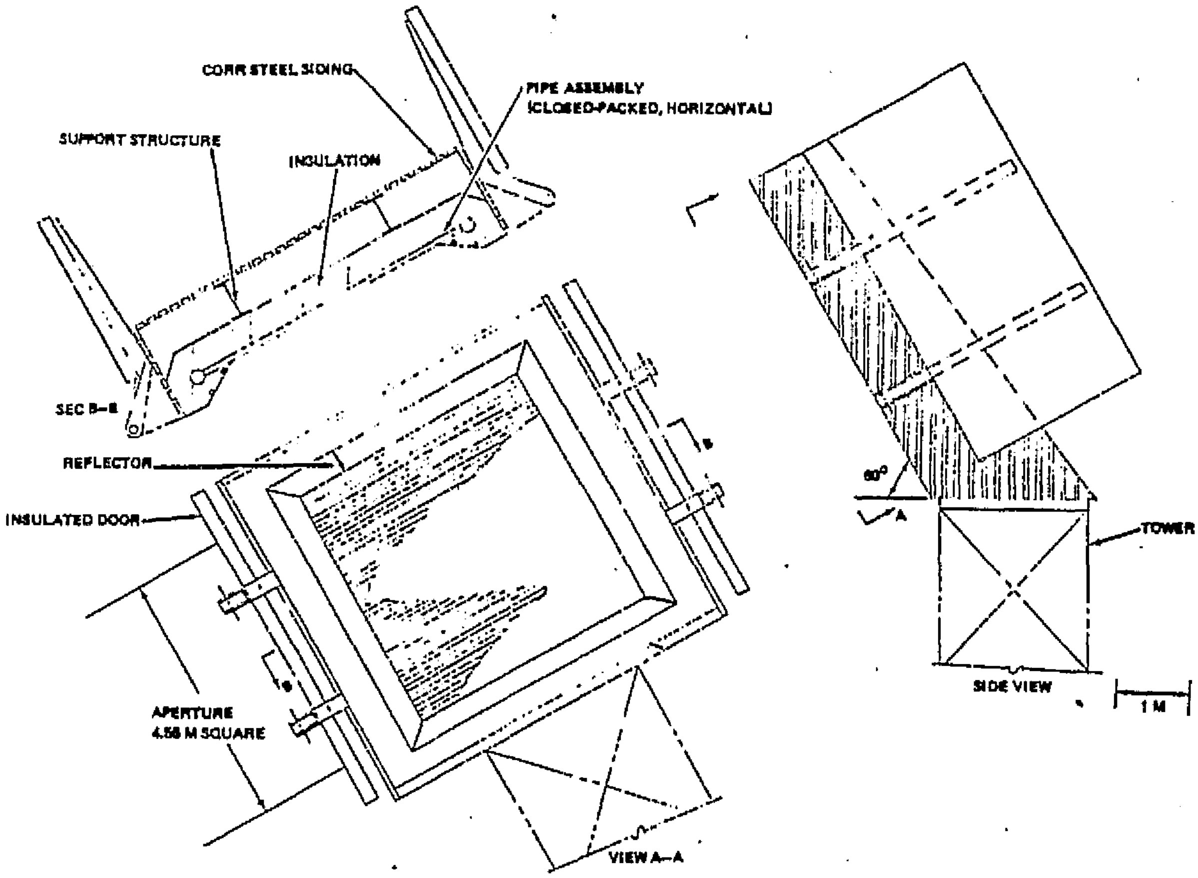


Figure 3.3-31. One-Zone Receiver, 4.83 MWT, HITEC

For this tubing arrangement, the sensitivity of the tube inside diameter to fluid velocity, pressure drop and manufacturing complexity (as measured by the number of flow paths), was estimated. As indicated on Figure 3.3-34, the selected design point for Hitec fluids is the use of 3.3 cm (1.3-inch) I.D. tubes and two parallel flow paths per set. A pressure drop of 4 N/m^2 (60 psi) is indicated for this design.

Preliminary weight estimates for the single-zone serpentine receiver have been made based upon the design deviations from the dual-zone receiver configuration. These design deviations include the reconfigured tubing arrangement, tubing size and materials. Preliminary cost estimates have also been made based on the design deviations from the dual-zone receiver concept and the unit material costs in Appendix A. Data for this design are summarized in Section 3.3.4.9.

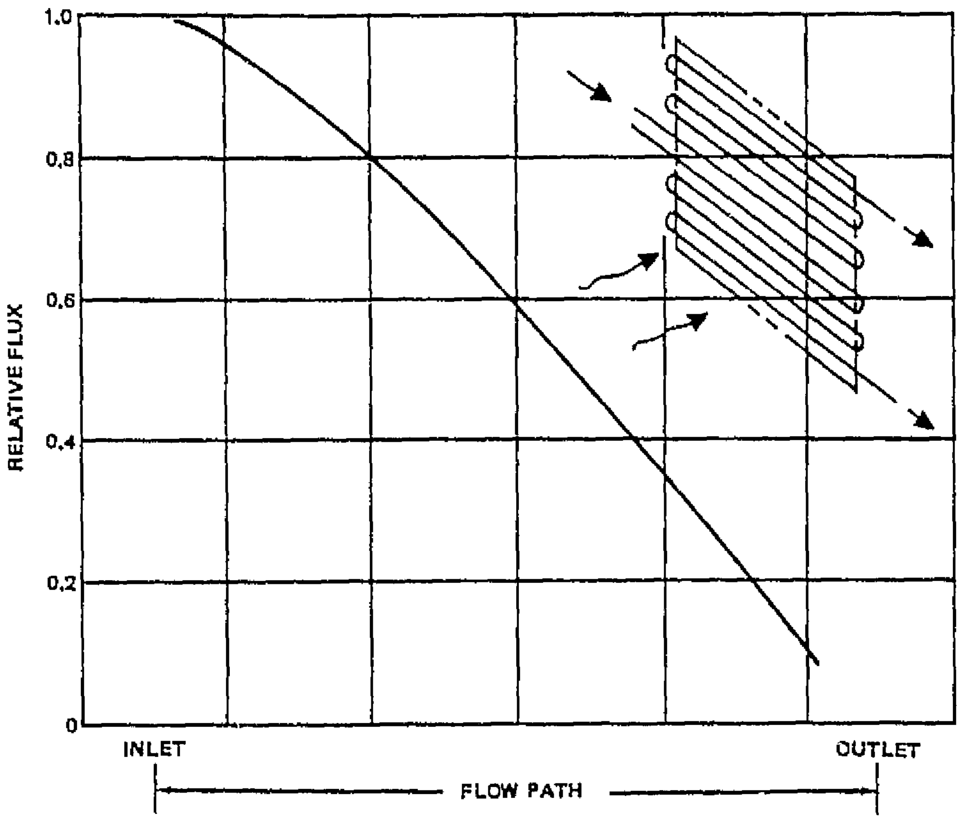


Figure 3.3-32. Reference Heat Flux Profile — Serpentine Flow Path (One Zone)

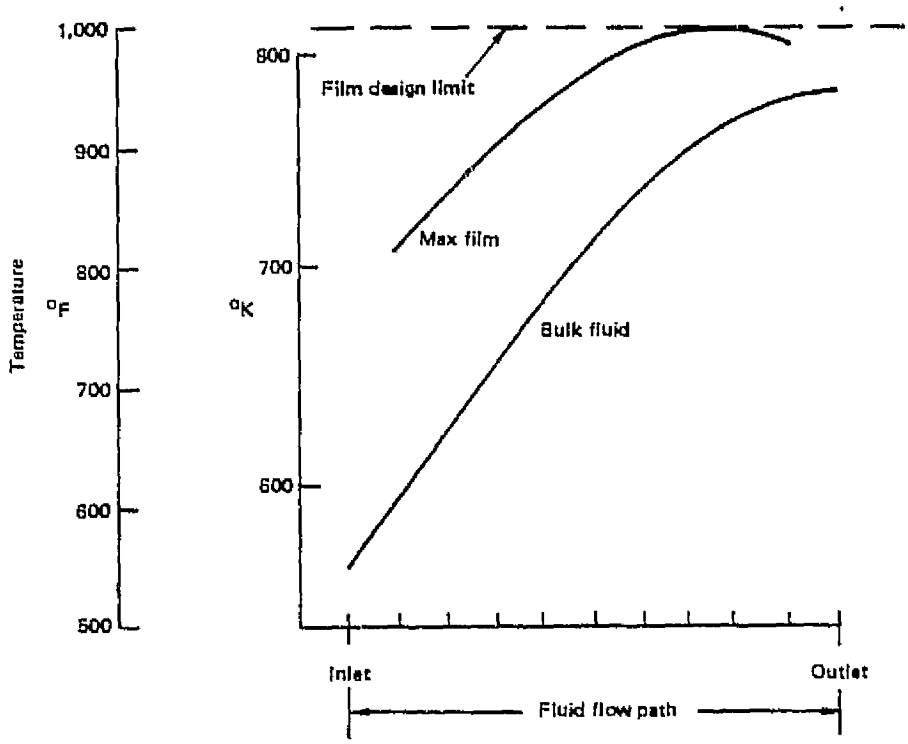


Figure 3.3-33. HTS Temperature Profiles — One-Zone Serpentine Flow Path

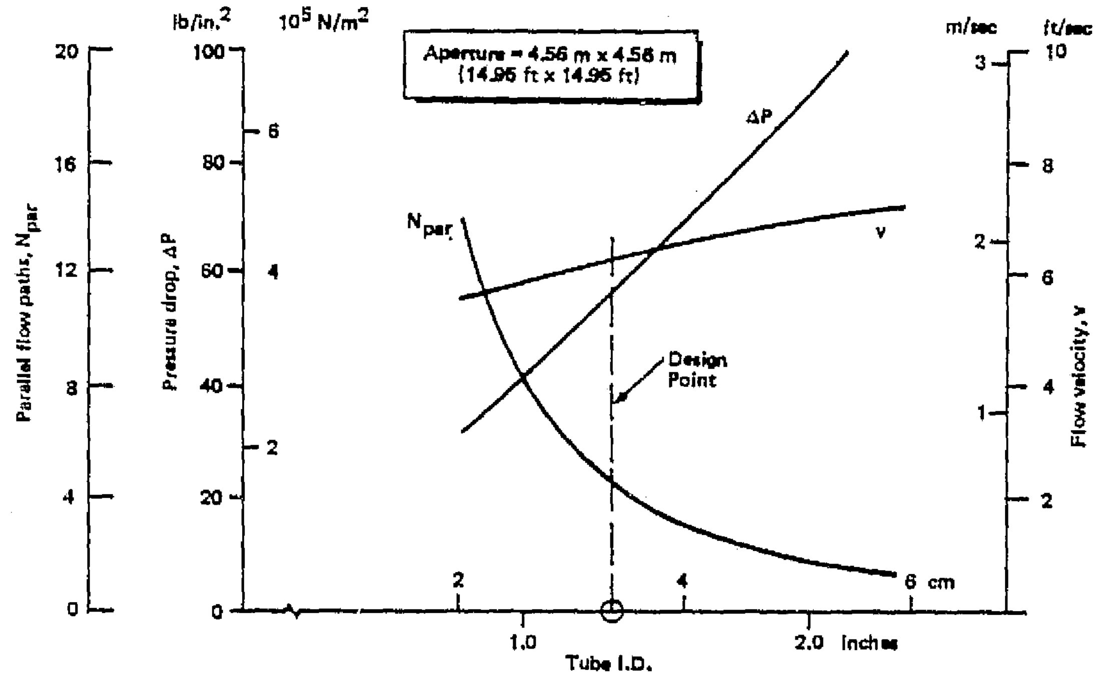


Figure 3.3-34. 4.83-MWt Receiver — HTS Fluid, One Zone

3.3.4.6 Single-Zone Receiver - Spiral

A sketch of a single-zone receiver configuration with a spiral tubing arrangement is shown on Figure 3.3-35. The heat transfer surface is composed of several tubes mounted in a spiral coil in a single zone. The receiver fluid enters the tubes at the center of the aperture, and travels in a spiral manner outward, exiting at the periphery of the coil, as shown on Figure 3.3-36. The number of tubes used is dependent upon the fluid used. For instance, for Hitec fluids, two tubes (in parallel) were required, whereas for Syltherm, three tubes were required.

The peak heat flux profile along the fluid flow from inlet to outlet is also represented on Figure 3.3-36. Bulk fluid and film temperature profiles for this receiver configuration using Hitec, are shown on Figure 3.3-37. As indicated on this figure, the "hot spot" (point of maximum film temperature) is located near the exit in a region of low heat flux.

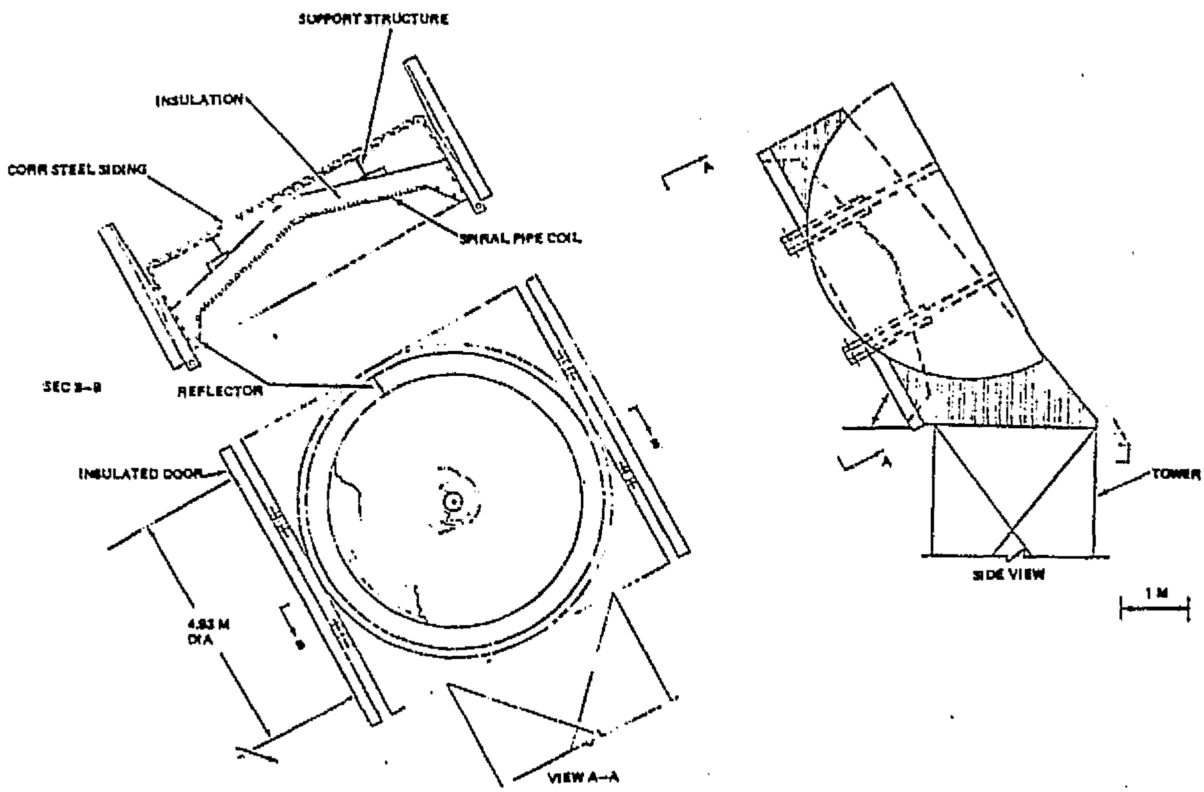


Figure 3.3-35. One-Zone Spiral Receiver, 4.83 MWT, HTS

For this tubing arrangement, the sensitivity of tube inside diameter to fluid velocity, pressure drop and manufacturing complexity (as measured by the number of flow paths) was estimated, as shown on Figure 3.3-38. The selected design point for Hitec fluids is the use of two 5.8 cm (2.3-inch) I.D. tubes. The corresponding temperature profiles and sizing parametrics for Syltherm are shown on Figures 3.3-39 and 3.3-40, respectively. The selected design point for Syltherm is the use of three 5.8-cm (2.3-inch) I.D. tubes, as shown on Figure 3.3-40.

Preliminary weight estimates for the single-zone spiral receiver have been made based upon the design deviations from the dual-zone receiver configuration. These design deviations include the reconfigured tubing arrangement, tubing size and materials. Preliminary cost estimates have also been made based on the design deviations from the dual zone receiver concept and the unit material costs given in Appendix A. Data on these receiver configurations are summarized in Section 3.3.4.9.

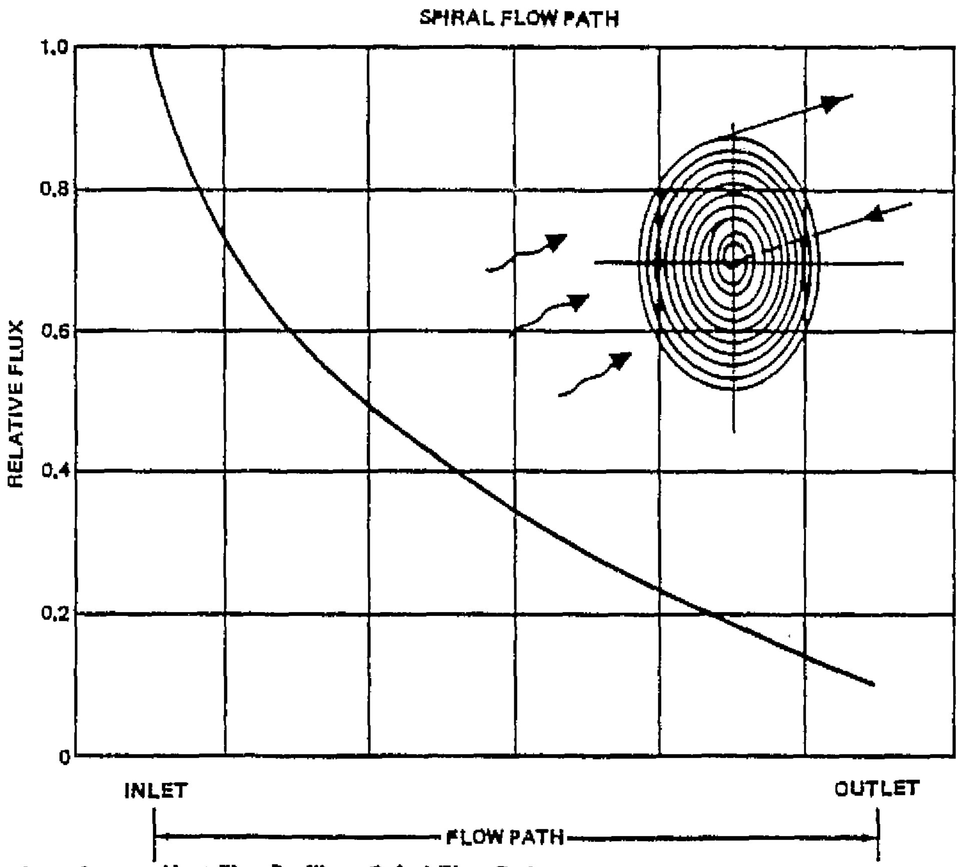


Figure 3.3-36. Reference Heat Flux Profile – Spiral Flow Path

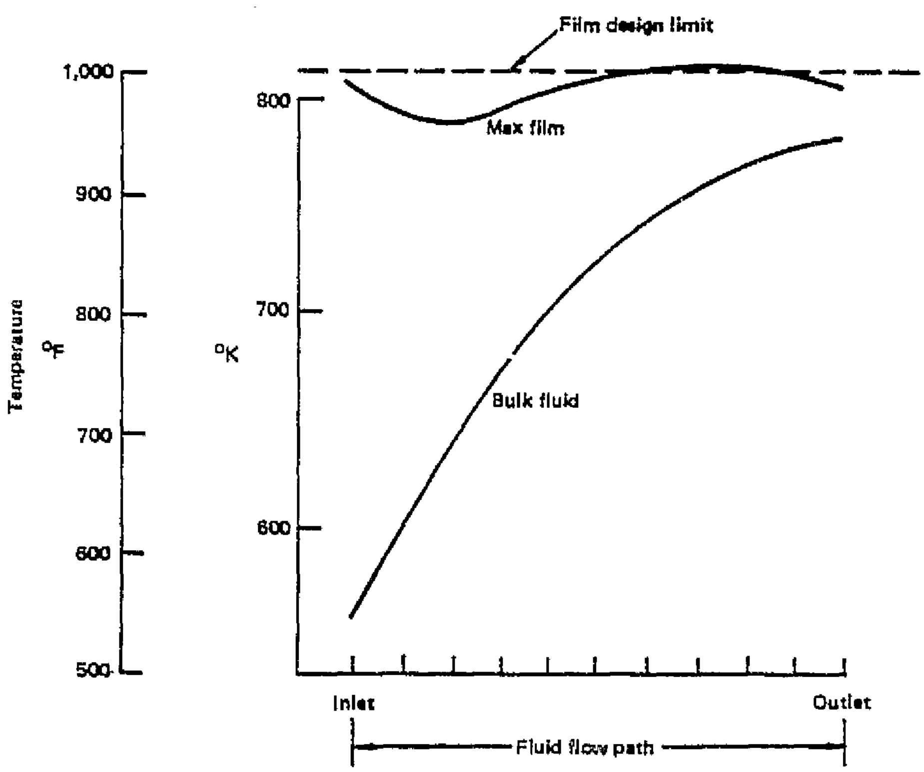


Figure 3.3-37. HTS Temperature Profiles – Spiral One-Zone Plane

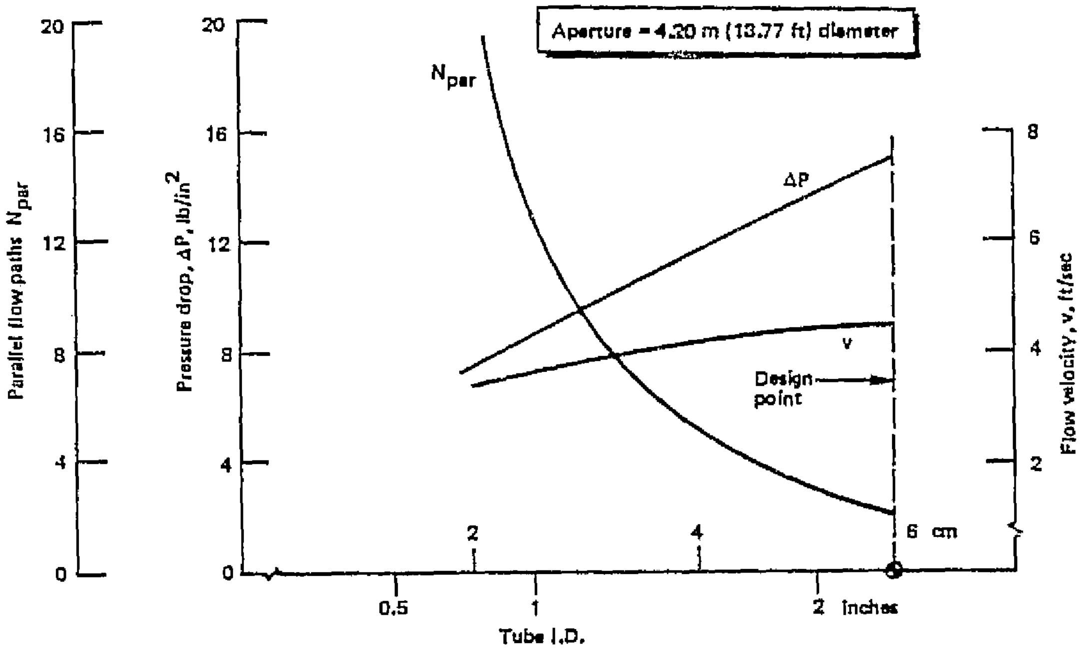


Figure 3.3-38. 4.83-MWt Receiver — HTS Fluid, Spiral

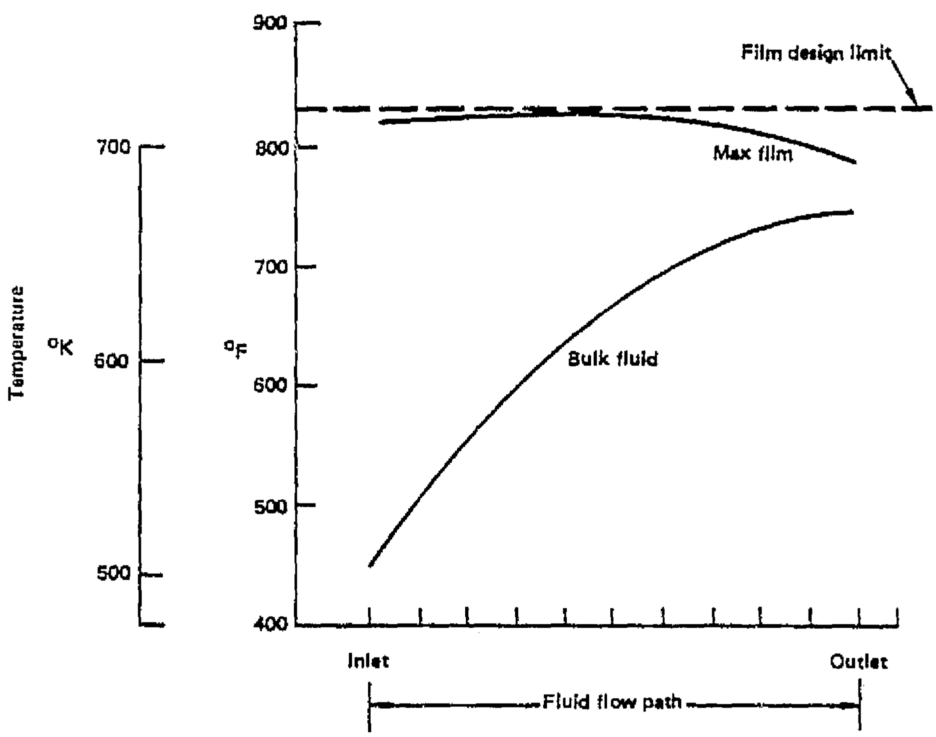


Figure 3.3-39. Syltherm Temperature Profiles — Spiral One-Zone Plane

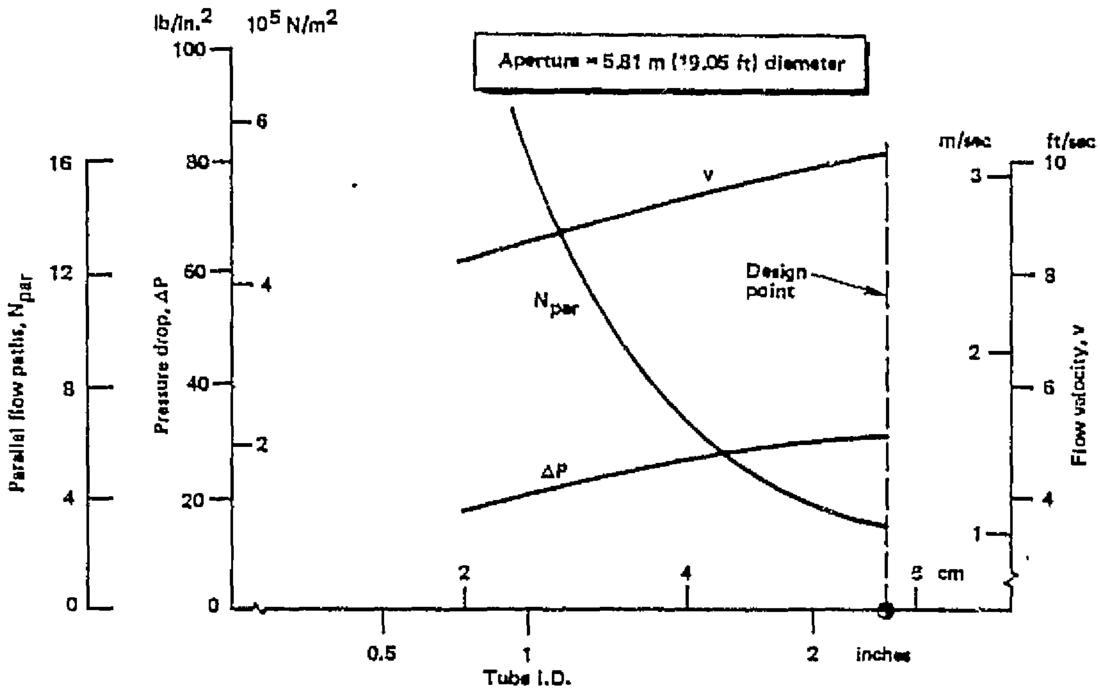


Figure 3.3-40. 5.89-MWt Receiver — Syltherm Fluid, Spiral

3.3.4.7 Cavity Receivers

Cavity-type receivers have been investigated for HTS, Syltherm and Caloria fluids and for air and helium gases. Radiant energy enters the receiver cavity through an aperture which faces the collector field. The energy is absorbed on the receiver heat transfer surface, which may cover all or a portion of the interior walls of the cavity. The aperture area will usually be set at the field-optics-limited value, in order to minimize radiation and reflection losses. The volume and shape of the cavity will be adjusted to provide the interior surface area and heat flux distribution appropriate to a particular heat transfer fluid. The relatively-inefficient heat transfer fluids (gases and oils) require large cavity volumes. The more efficient fluids (water, HTS) may require a heat transfer area no larger than the aperture.

Figures 3.3-41, 3.3-42, and 3.3-43 illustrate three of the many possible cavity receiver configurations. The receiver shown in Figures 3.3-41 has a

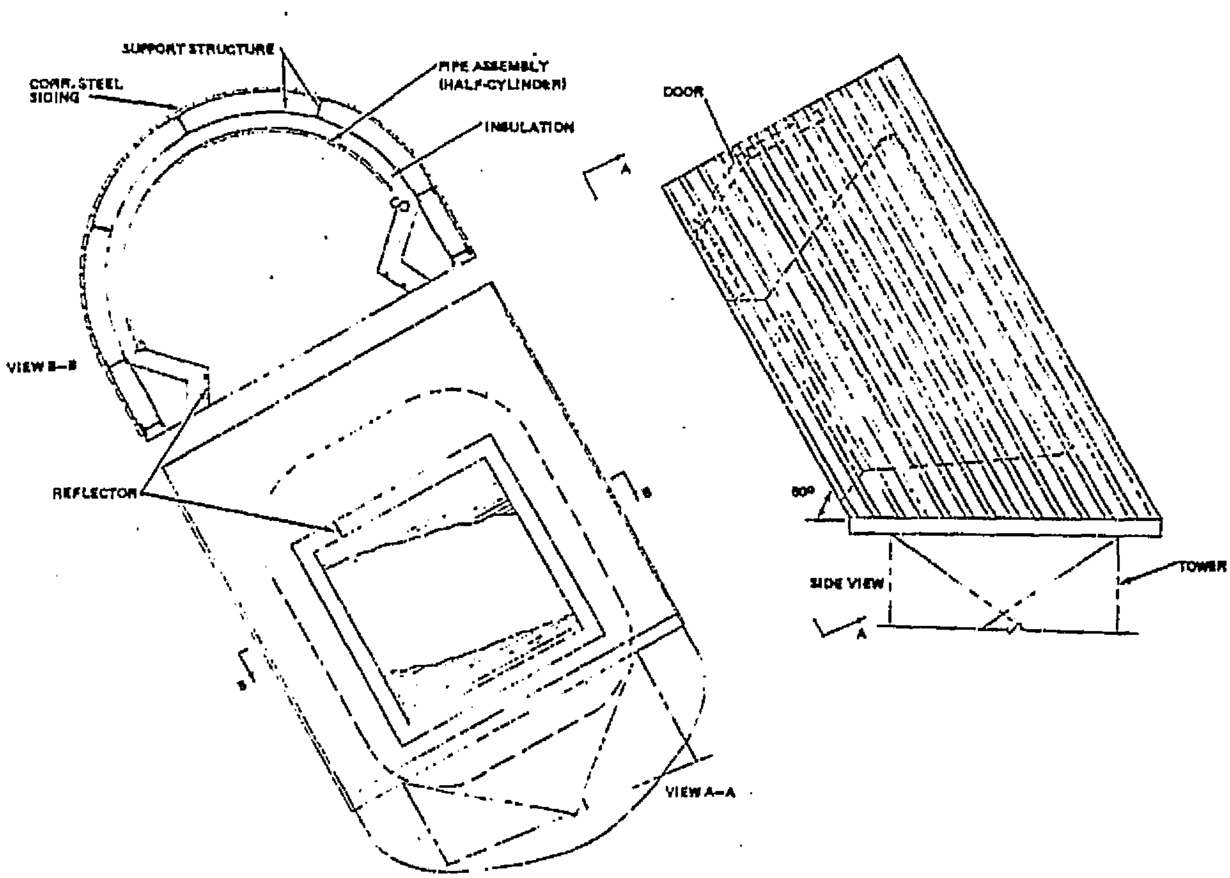


Figure 3.3-41. Semicylindrical Cavity Receiver

relatively large ratio of heat transfer surface area to aperture area. The heat transfer piping covers the illuminated portion of the cylindrical back wall of the cavity and is connected to provide one-zone, with two serpentine flow paths. This configuration is applicable to systems using the less-efficient gas or oil coolants. The receiver shown in Figure 3.3-42 is a conical cavity. The heat transfer surface is a spiral pipe coil. Fluid flow is from the apex of the cone outward to the edge. The design peak heat flux is controlled by the depth of the cone.

Figure 3.3-43 illustrates a partial cavity receiver which is a configuration having a portion of the absorber surface as an external receiver and the remainder as a cavity. The receiver may include a shroud of terminal concentrator. The shroud may be cooled or uncooled, or it may have both cooled and

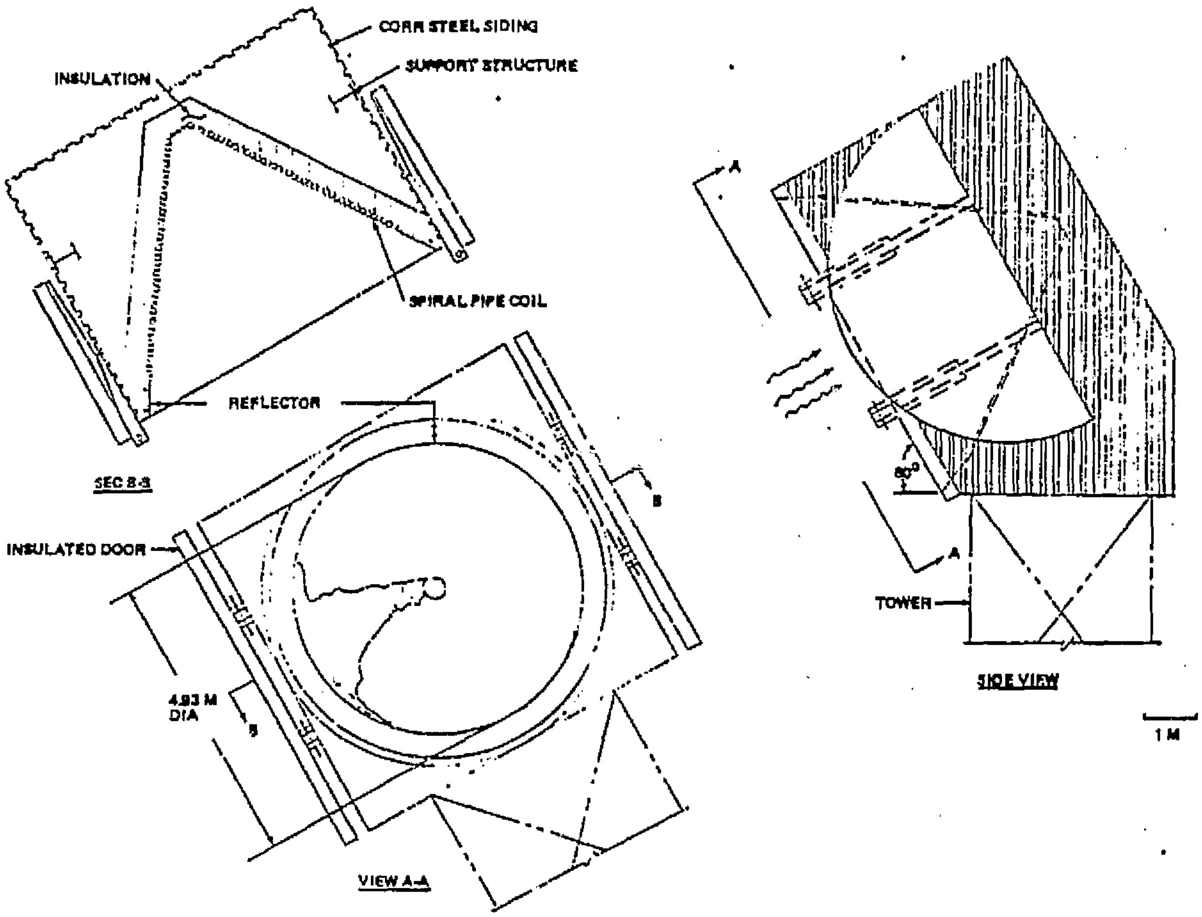


Figure 3.3-42. Cavity Receiver – Spiral Cone

uncooled regions. Inside the shroud, there is an external absorber region. The term "external" means that the optical shape factor for radiation heat transfer to other portions of the absorber surface is essentially zero. The center of the receiver is a cavity absorber. The purpose of the cavity is not so much to create an optical black body, as to create a significant local reduction in heat flux to prevent overheating of the receiver tubes and/or heat transfer fluid.

Reasons for considering a partial cavity receiver are:

1. The outer diameter of the receiver can be kept to the minimum consistent with collector field/receiver optics to maximize efficiency.
2. The colder fluid can be routed through the external absorber portion of the receiver to hold down surface temperature and losses.

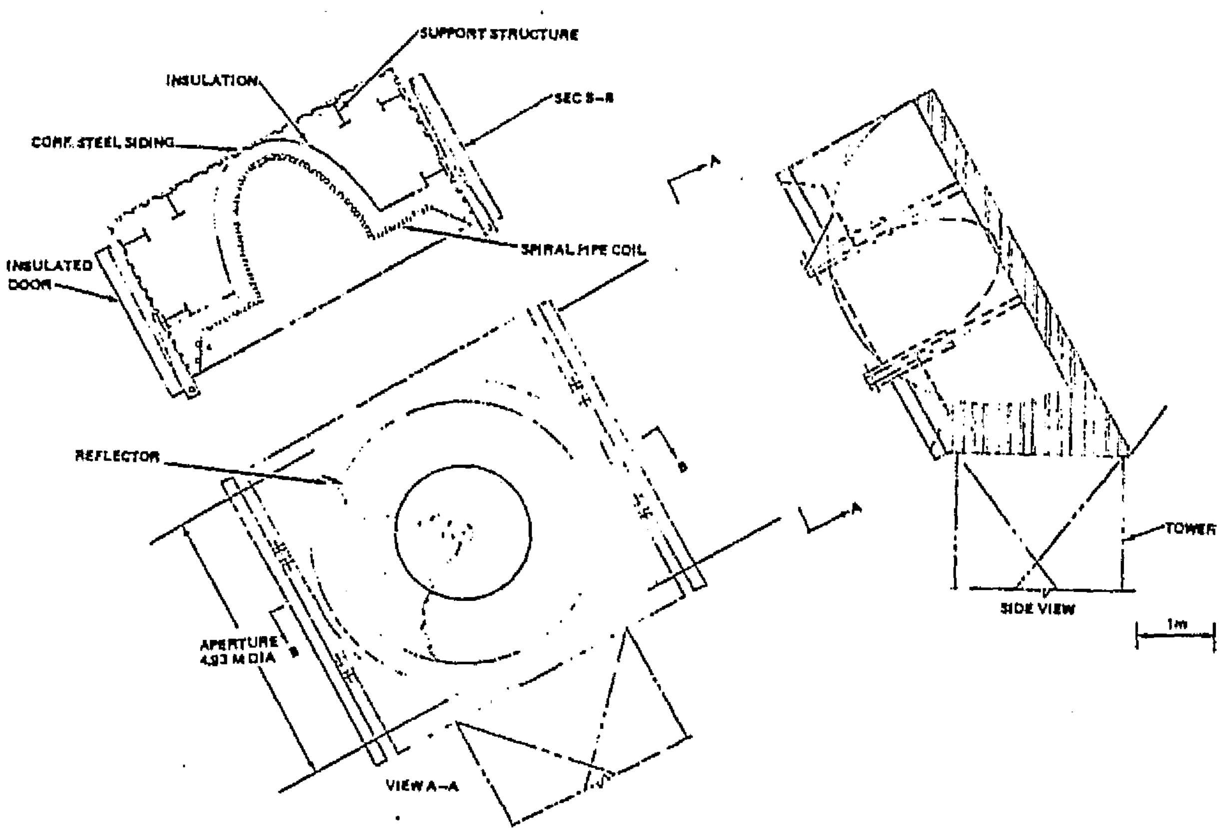


Figure 3.3-43. Partial Cavity Receiver (4.831 MWT HITEC)

3. The local heat flux in the cavity can be held at any desired value to prevent overheating of the tube materials and heat transfer fluid.
4. For all of the above reasons, the partial cavity receiver has a higher efficiency than an external receiver.
5. A full cavity receiver would have a still higher efficiency, but would also have a higher cost, weight, and aerodynamic drag.

The partial cavity receiver appears then, to present an attractive compromise between high efficiency and high costs for a reasonably effective heat transfer fluid.

There are two options for fluid flow path in the generic partial cavity receiver illustrated by Figure 3.3-43. In both options, the fluid enters in the shroud and spirals inward toward the center. The fluid may continue in an uninterrupted path to an outlet at the apex of the cavity. Alternatively, the fluid may be manifolded from the end of the external absorber to the apex of the cavity and exit where the cavity meets the external absorber. The two alternate fluid flow paths are shown schematically in Figures 3.3-44 and 3.3-45.

The partial cavity receiver generalized temperature distribution is shown on Figure 3.3-46.

Sizing parametrics for a 6.43-MWt full cavity receiver for Caloria fluid are shown in Figure 3.3-47. The corresponding receiver data are summarized in Section 3.3.4.9, as well as data for gas-cooled full cavities, two cavity/cone receivers (HTS and Syltherm fluids), and an HTS-cooled partial cavity.

3.3.4.8 Water/Steam Receiver

Figure 3.3-48 is a sketch of a natural recirculation saturated steam generator receiver. Radiation from the collector field is absorbed on a layer of close-packed, evaporator tubes located in the aperture plane. The steam

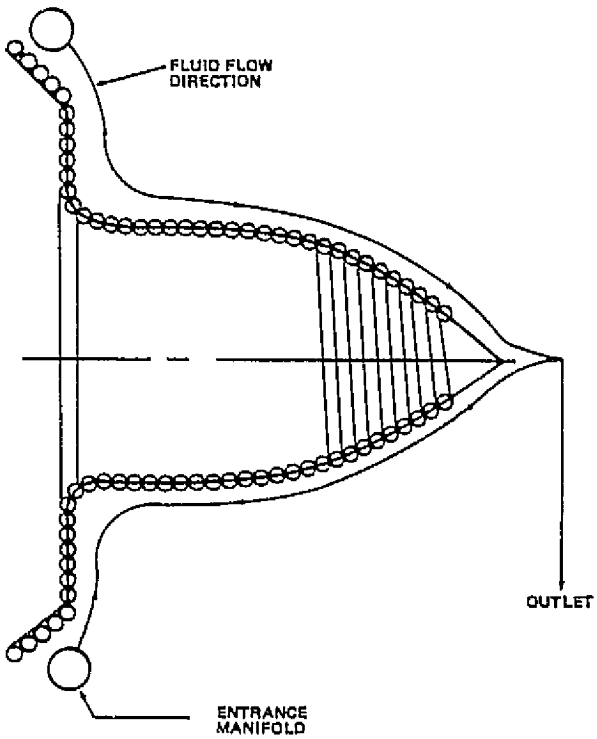


Figure 3.3-44. Partial Cavity Receiver — Inward Directed Flow

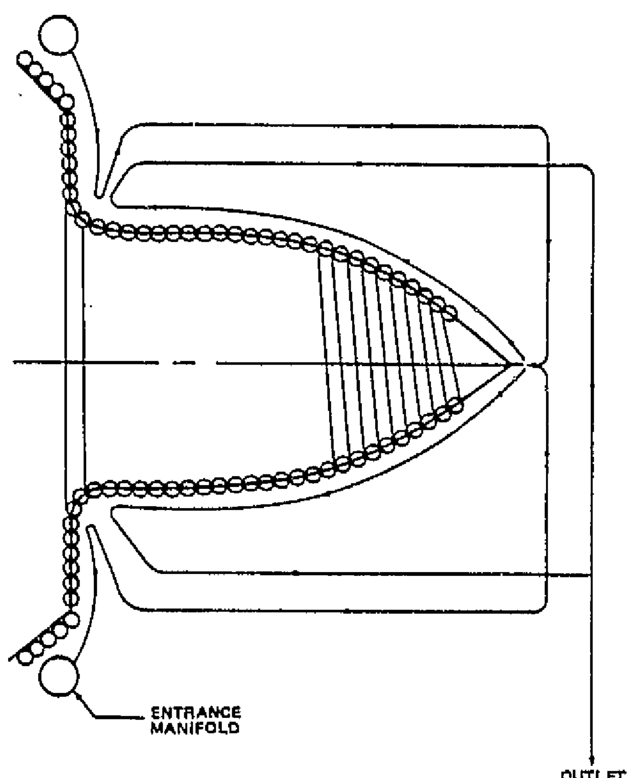


Figure 3.3-45. Partial Cavity Receiver — Split Flow

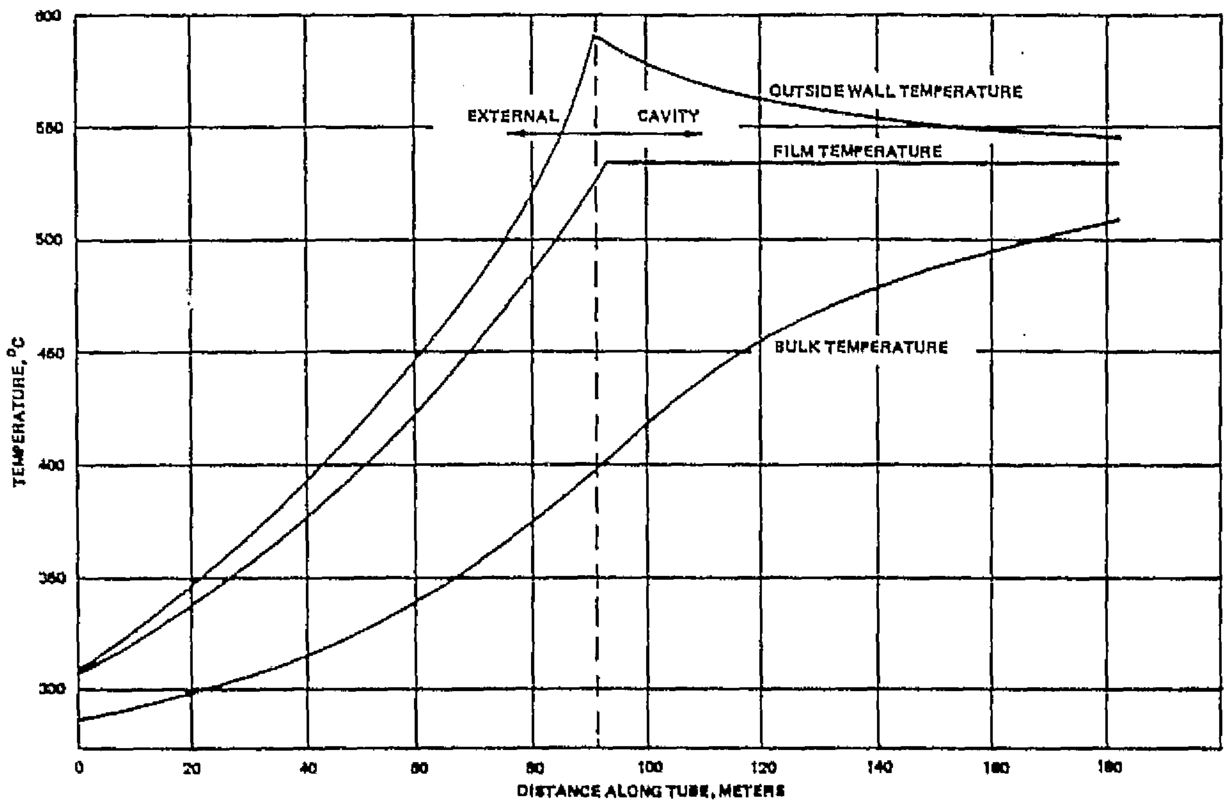


Figure 3.3-46. Partial Cavity Receiver Temperature Distributions

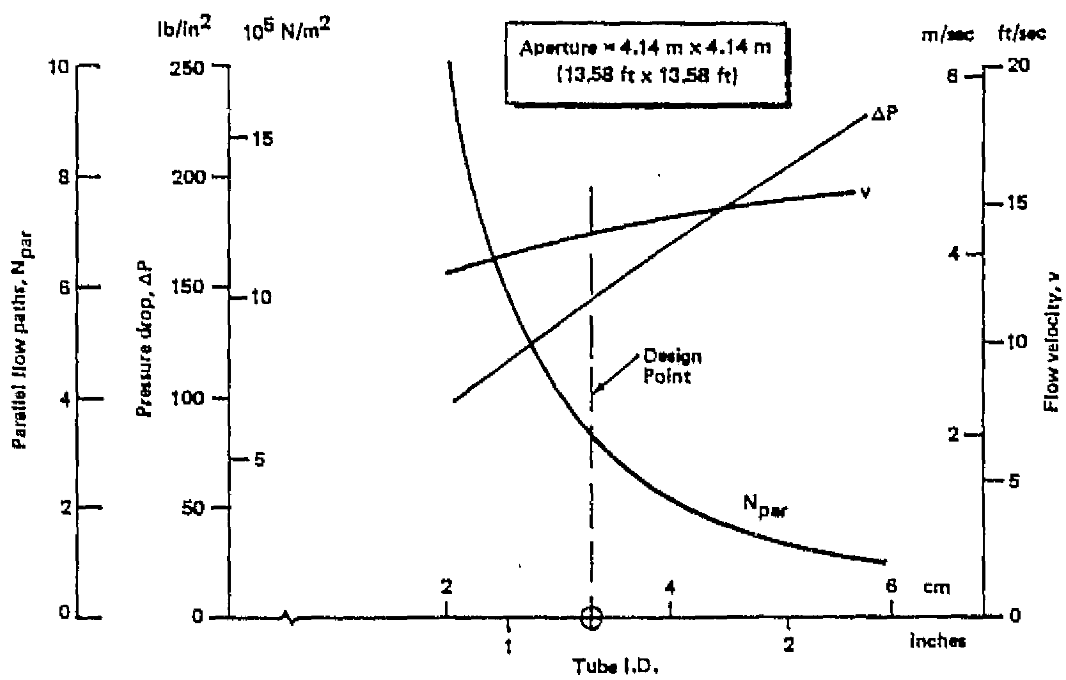


Figure 3.3-47. 6.43-MWt Receiver — Caloria HT-43 Fluid, Full Ca ity

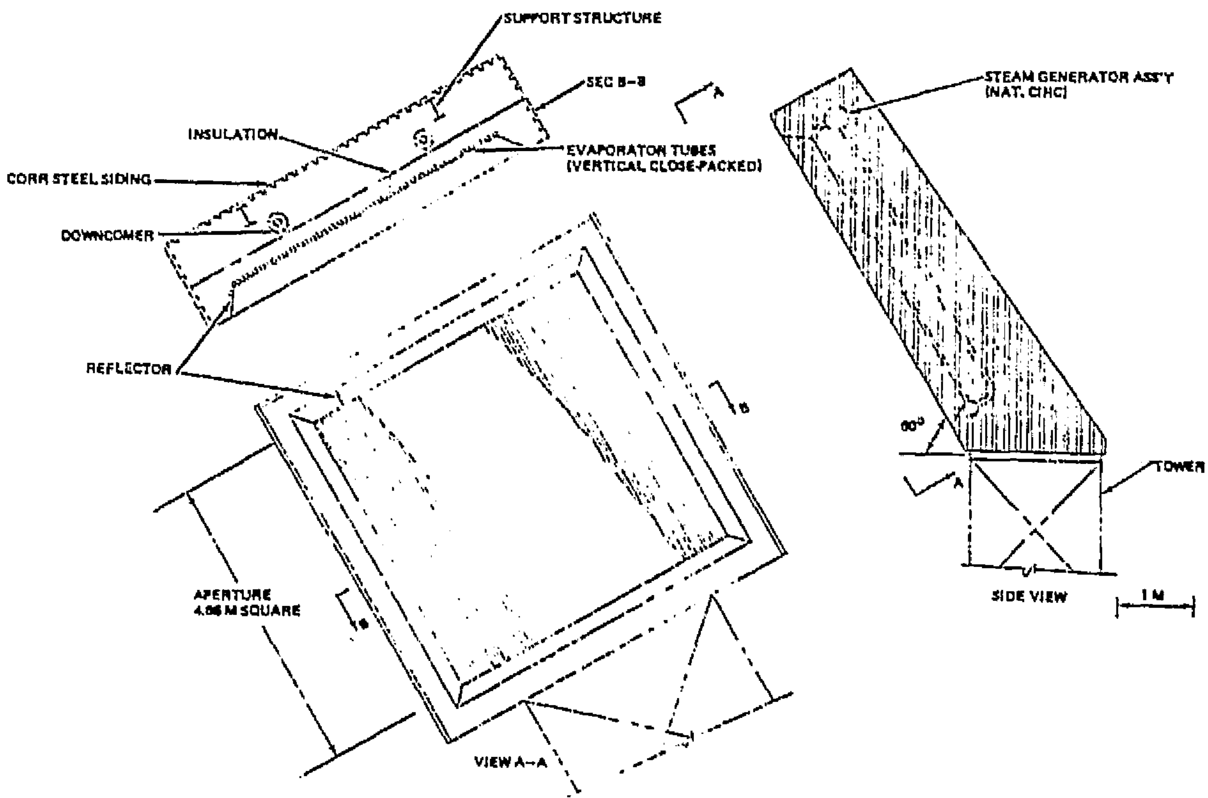


Figure 3.3-48. Steam Generator Receiver 8.36 MWT

generator loop, indicated in the side view on the figure, consists of the evaporator tubes, separator drum, downcomers and evaporator header. Saturated steam is produced in the evaporator tubes and flows as a steam/water mixture to the separator drum where it separates at the water surface. Water returns to the lower ends of the evaporator tubes via the downcomer piping (shown in Section B-B of the Figure). The circulation ratio (downcomer flow/steam flow) is about 15/1.

The aperture size, 5.15 m x 5.15 m, is determined by the system power requirement of 8.36 MwT and a peak heat flux limitation to 0.788 Mw/m^2. The field-optics-minimum aperture is about 4.65 m.

3.3.4.9 Receiver Configuration Tabulation

Data for the receiver configurations described in Sections 3.3.4.4 through 3.3.4.8 are listed in Table 3.3-12. The values for "absorbed power" and the fluid inlet and outlet temperatures are initial values set by the system requirements. The maximum film temperatures are the fluid manufacturer's suggested maximum values.

Table 3.3-12 (Page 1 of 2)

RECEIVER DESIGN SUMMARY

Section Ref. Receiver Type Fluid Path	3.3.4.4 Two-Zone Serpentine		3.3.4.4 Two-Zone Serpentine		3.3.4.5 One-Zone, Plane Serpentine		3.3.4.6 One-Zone, Plane Spiral		3.3.4.6 One-Zone, Plane Spiral		3.3.4.8 Cavity/Cone Spiral	
Absorbed Power, MWT	4.831		5.886		4.831		4.831		5.886		5.886	
Fluid	HTS		Syltherm		HTS		HTS		Syltherm		Syltherm	
T_{in}, °C (°F)	288	(550)	232	(450)	288	(550)	288	(550)	232	(450)	232	(450)
T_{out}, °C (°F)	510	(950)	399	(750)	510	(950)	510	(950)	399	(750)	399	(750)
$T_{max\ film}$, °C (°F)	538	(1,000)	443	(830)	538	(1,000)	538	(1,000)	443	(830)	443	(830)
Total Flow, kg/hr (lb/hr)	50,480	(111,300)	60,740	(133,900)	50,480	(111,300)	50,480	(111,300)	60,740	(133,900)	60,740	(133,900)
Total Flow, L/sec (GPM)	7.70	(122)	23.9	(379)	7.70	(122)	7.70	(122)	23.9	(379)	23.9	(379)
Max. Velocity, m/sec (ft/sec)	1.92	(6.29)	3.05	(10.0)	1.92	(6.29)	1.37	(4.49)	3.04	(9.98)	3.04	(9.98)
Aperture Size (Excluding Reflector), m x m (ft x ft)	3.72	(12.19)	5.75	(18.86)	4.56	(14.95)	4.20	(13.77 Diam.)	5.80	(19.05 Diam.)	4.51	(14.79 Diam.)
Peak Heat Flux, Kw/m^2 (BTU/hr ft^2)	875	(277,400)	446	(141,300)	582	(184,400)	875	(277,400)	555	(176,100)	555	(176,100)
I.D. of Flow Channel, cm (in)	3.33	(1.31)	3.33	(1.31)	3.33	(1.31)	5.79	(2.28)	5.79	(2.28)	5.79	(2.28)
Max. Film Coef., Kw/m^2 °C (BTU/hr ft^2 °F)	1.61	(922)	0.701	(400)	1.61	(922)	1.10	(629)	0.701	(400)	0.701	(400)
Radiation Loss, Kwt	245		339		331		380		418		251	
Convection Loss, Kwt	179		388		274		219		329		199	
Conduction Loss, Kwt	8.9		21		14		8.9		17		17	
Reflection Loss, Kwt	254		310		254		254		310		91	
Receiver Efficiency	0.876		0.848		0.847		0.849		0.846		0.913	
Coolant ΔP, Bars (lb/in^2)	4.78	(69.3)	3.36	(48.8)	4.02	(58.3)	1.29	(18.7)	2.05	(29.7)	2.05	(29.7)
Pumping Power, Kw Hyd.	3.67		8.06		3.09		0.99		5.01		5.01	
Heat Trans Area, m^2 (ft^2)	20.7	(223)	49.6	(534)	20.7	(223)	13.8	(149)	26.5	(285)	26.5	(285)
Pipe Size	1-1/2 OD - 13 GA		1-1/2 OD - 13 GA		1-1/2 OD - 13 GA		2-1/2 OD - 12 GA		2-1/2 OD - 12 GA		2-1/2 OD - 12 GA	
Material	Alloy Steel/ Carbon Steel		Carbon Steel		Alloy Steel		Alloy Steel		Carbon Steel		Carbon Steel	
Weight, Pipe Ass'y, kg (lb)	1,860	(4,100)	4,381	(9,658)	1,906	(4,201)	1,301	(2,869)	2,398	(5,286)	2,398	(5,286)

Table 3.3-12 (Page 2 of 2)

RECEIVER DESIGN SUMMARY

Section Ref.	3.3.4.7	3.3.4.7	3.3.4.7	3.3.4.7	3.3.4.8
Receiver Type	Cavity/Semi-Cyl	Partial Cavity	Cavity/Cone	Cavity/Semi-Cyl	Steam Generator
Fluid Path	Serpentine		Spiral	U-Tube	
Absorbed Power, Mwt	6.430	4.831	4.831	4.380	8.36
Fluid	Caloria HT-43	HTS	HTS	Air at 4 Bar	Boiling Water at 42 Bar
T_{in}, °C (°F)	218 (425)	288 (550)	288 (550)	538 (1,000)	109 (228)
T_{out}, °C (°F)	316 (600)	510 (950)	510 (950)	816 (1,500)	252 (486)
$T_{max\ film}$, °C° (°F)	360 (680)	538 (1,000)	538 (1,000)	871 (1,600)	271 (520)
Total Flow, kg/hr (lb/hr)	81,240 (179,100)	50,490 (111,300)	50,490 (111,300)	54,210 (119,500)	12,250 (27,000)
Total Flow, L/sec (GPM)	32.7 (519)	7.70 (122)	7.70 (122)	-	-
Max. Velocity, m/sec (ft/sec)	3.87 (12.7)	2.82 (9.25)	1.37 (4.49)	30.5 (100)	-
Aperture Size, m x m (ft x ft)	4.14 (13.58)	4.20 (13.77 Diam.)	4.20 (13.77 Diam.)	3.17 (10.4)	4.65 (15.25)
Peak Heat Flux, Kw/m^2 (BTU/hr ft^2)	242 (76,700)	445 (141,300)	527 (167,000)	47.3 (15,000)	789 (250,000)
I.D. of Flow Channel, cm (in)	3.33 (1.31)	5.79 (2.28)	5.79 (2.28)	2.62 (1.032)	5.26 (2.07)
Max. Film Coef., Kw/m^2 °C (BTU/hr ft^2 °F)	0.698 (400)	1.94 (1,113)	1.10 (629)	0.075 (43)	14.0 (8,000)
Radiation Loss, Kwt	149	223	286	558	101
Convection Loss, Kwt	164	219	195	227	232
Conduction Loss, Kwt	29	13	16	180	17
Reflection Loss, Kwt	85	199	74	-	440
Receiver Efficiency	0.938	0.881	0.894	0.819	0.914
Coolant ΔP, Bars (lb/in^2)	9.73 (141)	7.17 (104)	2.25 (32.6	0.239 (3.46)	-
Pumping Power, Kw Hyd	31.9	5.49	1.73	165	-
Proj. Heat Trans Area, m^2 (ft^2)	66.5 (716)	19.5 (210)	24.2 (260)	232 (2,500)	26.5 (285)
Pipe Size	1-1/2 in - 13 GA	2-1/2 in - 12 GA	2-1/2 in - 12 GA	1-1/4 in - 12 GA	2-1/2 - Sch 40
Material	Carbon Steel	Alloy Steel	Alloy Steel	Nickle Alloy	Carbon Steel
Weight, Pipe Ass'y, kg (lb)	5,579 (12,300)	1,595 (3,516)	2,117 (4,667)	18,460 (40,700)	3,856 (8,500)

The thermal losses are a function of the configuration and the operating temperature level. The convection loss is the least-certain of the listed losses. It corresponds to a film coefficient of 0.9 W/m^2 °C over the aperture area. Radiation and reflection losses assume 0.95 emissivity and absorptivity. Conduction losses assume 25 cm (12 in) of thermal insulation and conductivity of 0.001 W/cm °C.

From the data presented in Table 3.3-12, the following conclusions can be made:

- Receivers using HTS fluids have the smallest aperture area, the lowest weight, and require the least power for pumping the fluid.
- Cavity receivers nave the highest thermal efficiency and weight.
- Receivers using the oils (Caloria; Syltherm) have high efficiencies, however, the receivers are large, heavy and require high pumping power.
- Receivers using air as the heat transfer medium are extremely large and heavy.
- The spiral one zone receiver configurations require the minimum heat transfer surface.
- The spiral cavity and partial cavity configurations produce higher receiver efficiencies than the planar one or two zone configurations, at a small increase in size and weight.

3.3.5 Tower Design Concepts

Conceptual tower designs and corresponding cost estimates were prepared to facilitate system evaluation and selection. Attention was given to both free standing and guyed steel structures. Reinforced concrete towers were given a precursor evaluation and eliminated from further consideration because of:

(1) Traditionally higher costs associated with concrete structures of this size in comparison to steel structures due to extensive on-site construction activities and substantial foundation require-

ments. (Steel towers can be partially prefabricated and site-assembled in sections.)

(2) Structural stiffness which produce high receiver accelerations during seismic events which requires additional receiver structure. (Flexible steel towers absorb some of the ground motion, delivering less severe acceleration loads to the receiver.)

(3) Greater difficulty in attaching pipe supports, work platforms, and providing extensive access for maintenance.

3.3.5.1 Tower Requirements

The requirements upon which the preliminary design and costing activities were based can be divided into design and environmental factors. From a design standpoint, it was desired to develop data over a sufficient range of tower height and receiver weight to permit these results to be applicable to any of the candidate systems. As a result, three discrete combinations of tower height and receiver weight were specified for each tower type.

	Tower Height m (ft)	Receiver Weight Kg (lb)
Case 1	48 (158)	7,273 (16,000)
Case 2	48 (158)	34,090 (75,000)
Case 3	42 (138)	7,273 (16,000)

In addition, the heavier receiver, with a face dimension of 12.2 x 12.2 m (40 x 40 ft), was assumed to have its center of gravity located 4.6 m (15 ft) above the top of the tower and located along the vertical centerline of the tower. The receiver attachment points were assumed to be the corners of a square pattern 4.9 m (16 ft) on a side. The lighter receiver, with a face dimension of 5.2 x 5.2 m (17 x 17 ft), was assumed to have its center of gravity located 2.3 m (8.5 ft) above the top of the tower and displaced by 1.6 m (5.3 ft) from the tower centerline. The receiver attachment points were assumed to be the corners of a square pattern, 2.45 m (8 ft) on a side.

From an environmental standpoint, the following requirements were specified:

Operating wind speed at 10 m elevation	16.1 m/sec	(36 mph)
Operating deflection	0.15 m	(6 in.)
Survival wind speed	40.2 m/sec	(90 mph)
Seismic load	0.25 g	(horizontal ground acceleration)
Soil bearing strength	7,322 Kg/m^2	(1500 lbs/ft^2)

3.3.5.2 Guyed Steel Tower Concepts

The guyed steel tower which is shown in Figure 3.3-49, in the configuration required to support the heavier receiver load, is of a constant cross-section with four guy cables strung at 45° angles to the sides. In carrying out the analysis, it was found that the overturning moment associated with the survival wind load was a factor of 2 larger than the seismic induced moment. As a result, the towers were designed on the basis of wind loads requirements.

The principal design characteristics for each of the guyed towers are summarized in Table 3.3-13. The structural steel which forms the vertical structure and drag bracing is made up of commercial steel angles with the angle depth and thickness being selected to accommodate local load conditions. Cabling is assumed to be of commercial galvanized bridge cable type with the diameter being determined on the basis of loads associated with the maximum overturning moment condition.

The tower foundation consists of a mat design of sufficient area to distribute the compressive load at a rate less than the soil bearing strength limit of 7322 Kg/m^2 (1500 lbs/ft^2). The mat is assumed to be 0.61 m (2 ft) thick which is a sufficient depth, based on Barstow soils data, to encounter reasonably stable soil. The deadmen consist of buried concrete piers which are sized to accommodate the maximum cable loads.

3.3.5.3 Free Standing Tower Concept

The free standing steel tower of the type shown in Figure 3.3-50 is a tapered design with the base dimension approximately one-fifth (1/5) the tower height.

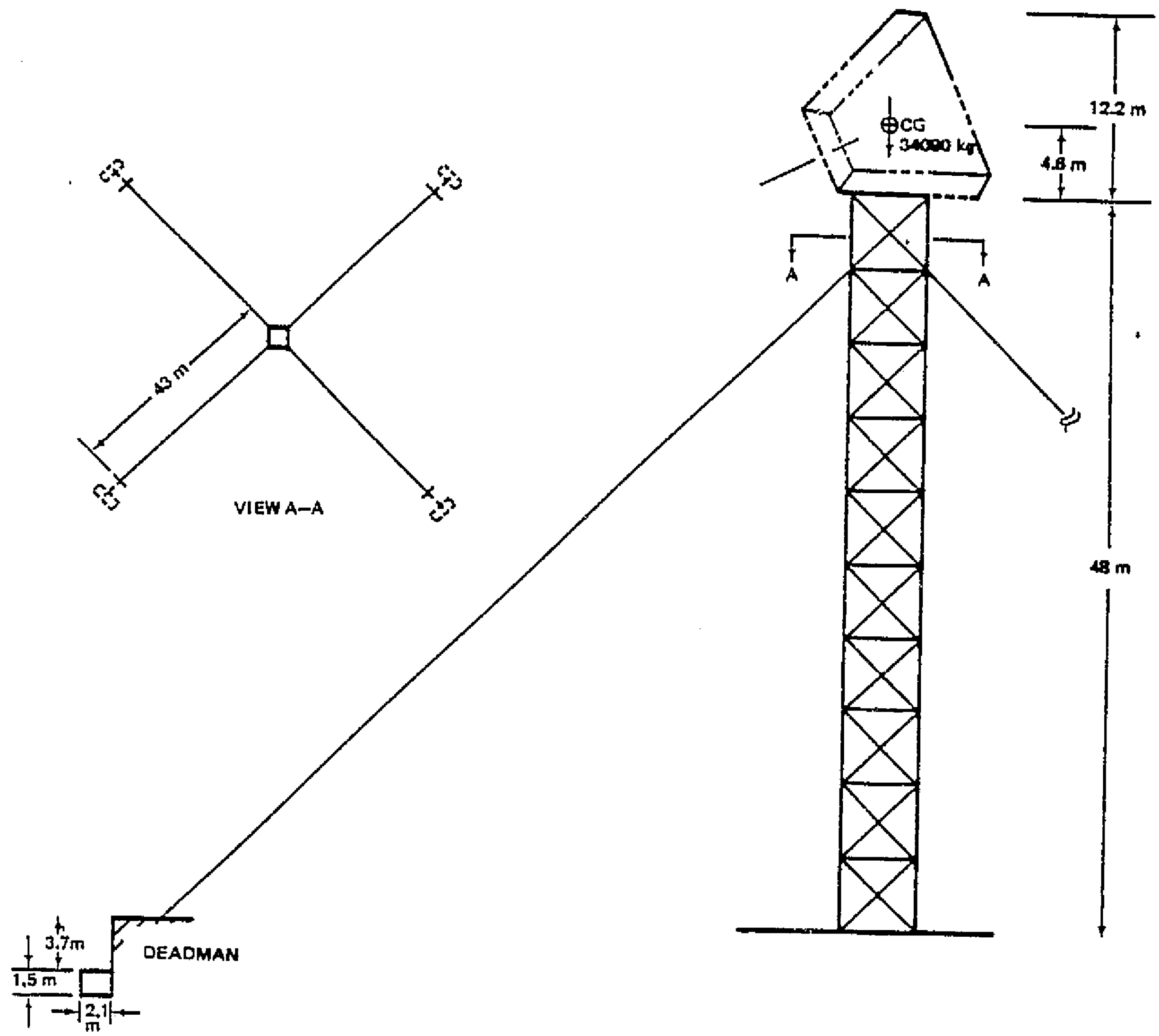


Figure 3.3-49. Guyed Tower Design (34,090 kg Receiver)

Table 3.3-13

CHARACTERISTICS OF GUYED STEEL TOWERS

Tower Height		Receiver Weight		Structural Steel		Cable Diameter		Cable Length		Concrete	
m	(ft)	Kg	(lb)	Kg	(lb)	cm	(in.)	m	(ft)	m^3	(yd^3)
48	(158)	7,273	(16,000)	17,341	(38,150)	2.06	(13/16)	305	(1,000)	23	(30)
48	(158)	34,090	(75,000)	24,091	(53,000)	4.45	(1-3/4)	305	(1,000)	64	(84)
42	(138)	7,273	(16,000)	14,841	(32,650)	1.91	(3/4)	262	(860)	23	(30)

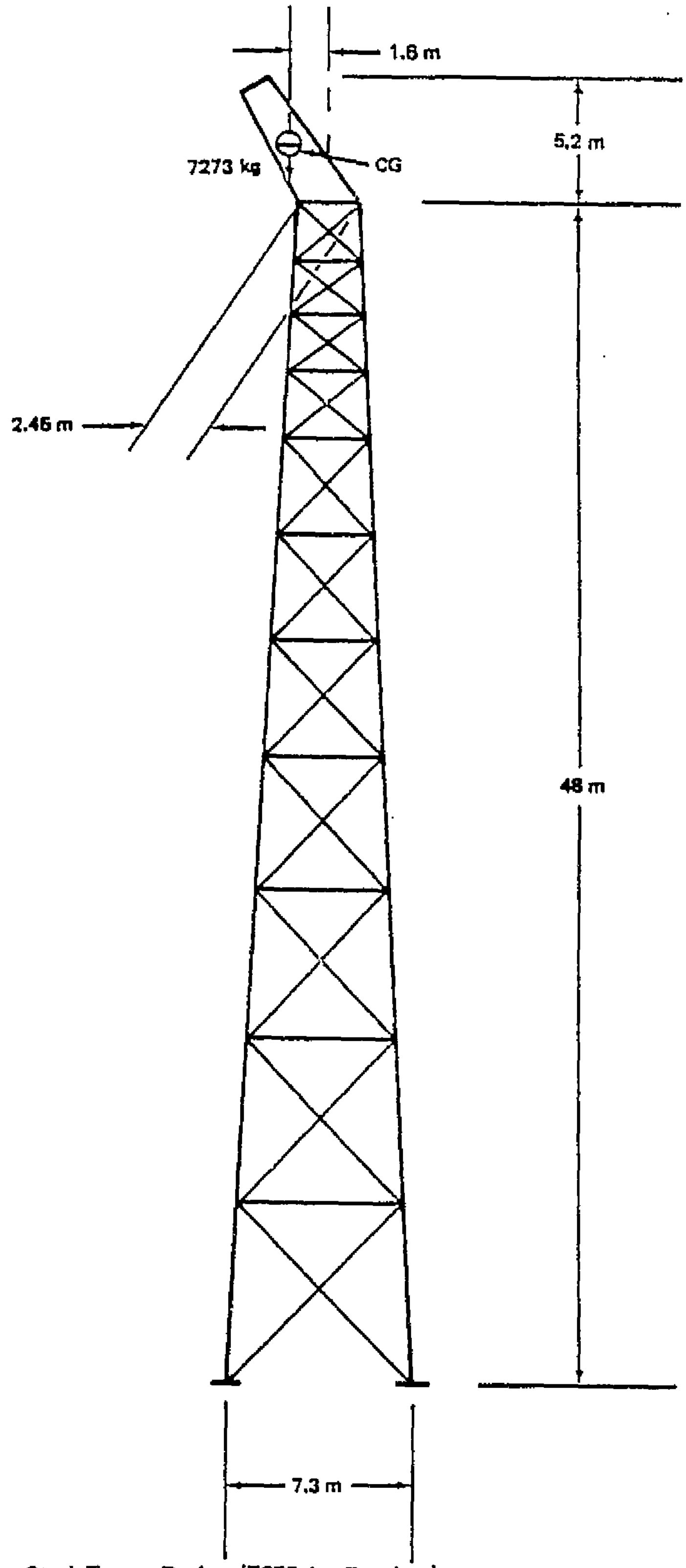


Figure 3.3-50. Free Standing Steel Tower Design (7273 kg Receiver)

As in the case of the guyed tower, the structural and foundation designs are based on the overturning moments created by the maximum wind loads.

The principal design characteristics for the free standing towers are shown in Table 3.3-14. The structural steel contained in the vertical members and drag braces is assumed to be commercial angle steel. The foundations for each of the four legs are designed to withstand the overturning moments while providing a sufficient base for the distribution of the compressive loads consistent with soil loading limitations.

Table 3.3-14
CHARACTERISTICS OF FREE STANDING STEEL TOWERS

Tower Height		Receiver Weight		Structural Steel		Concrete		Dimensions (Square) Top		Base	
m	(ft)	Kg	(lb)	Kg	(lb)	m^3	(yd^3)	m	(ft)	m	(ft)
48	(158)	7,273	(16,000)	28,545	(62,800)	142	(186)	2.4	(8)	7.3	(24)
48	(158)	34,090	(75,000)	45,113	(99,250)	153	(200)	4.9	(16)	9.7	(31.8)
42	(138)	7,273	(16,000)	24,364	(53,600)	126	(165)	2.4	(8)	7.3	(24)

3.3.5.4 Tower Concept Evaluation

Figure 3.3-51 presents tower cost data as a function of tower height and receiver weight. The results indicate the consistent superiority of the guyed tower over the height and weight ranges of interest in this study. Figure 3.3-52 presents a comparison of the cost breakdowns for two towers designed to satisfy an identical set of design requirements. It is seen that each of the cost increments for the free standing tower exceeds the corresponding value shown for the guyed tower (except the electrical value) with the biggest discrepancy occurring for the concrete required for foundations and supports. The indirect entries include construction equipment and supplies, temporary facilities, labor benefits, and other field expenses. The miscellaneous category includes engineering, contingency, and fees.

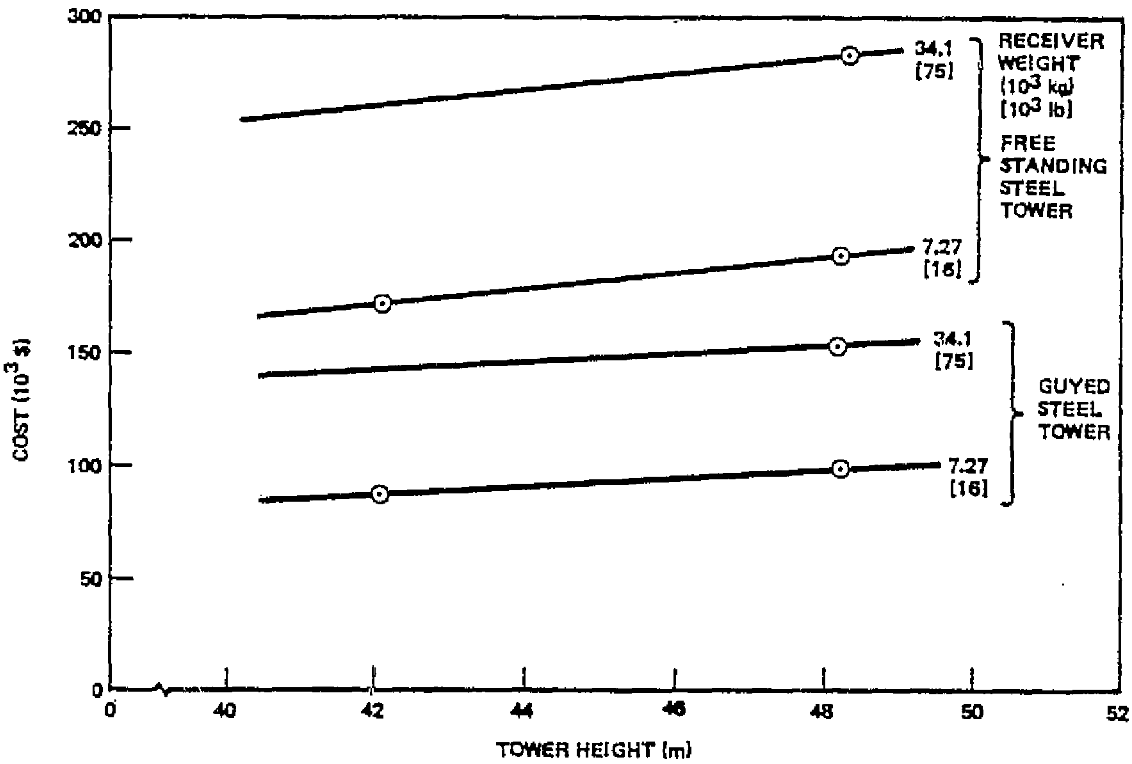


Figure 3.3-51. Cost Comparison Between Free Standing and Guyed Steel Towers

Based on these cost data, the guyed tower is a superior choice for the present application and will be retained as the baseline tower configuration. In addition, the high cost increment associated with concrete for the free standing steel tower also supports the earlier decision to eliminate the free standing concrete tower from further consideration.

3.3.6 Energy Storage

The purpose of this task was to optimize the selected storage concepts for each of the candidate transport fluids so that costs can be determined for each subsystem as a function of storage capacity. Once the storage requirement (capacity) has been established for any candidate system, the storage subsystem could then be determined for comparisons of total system costs.

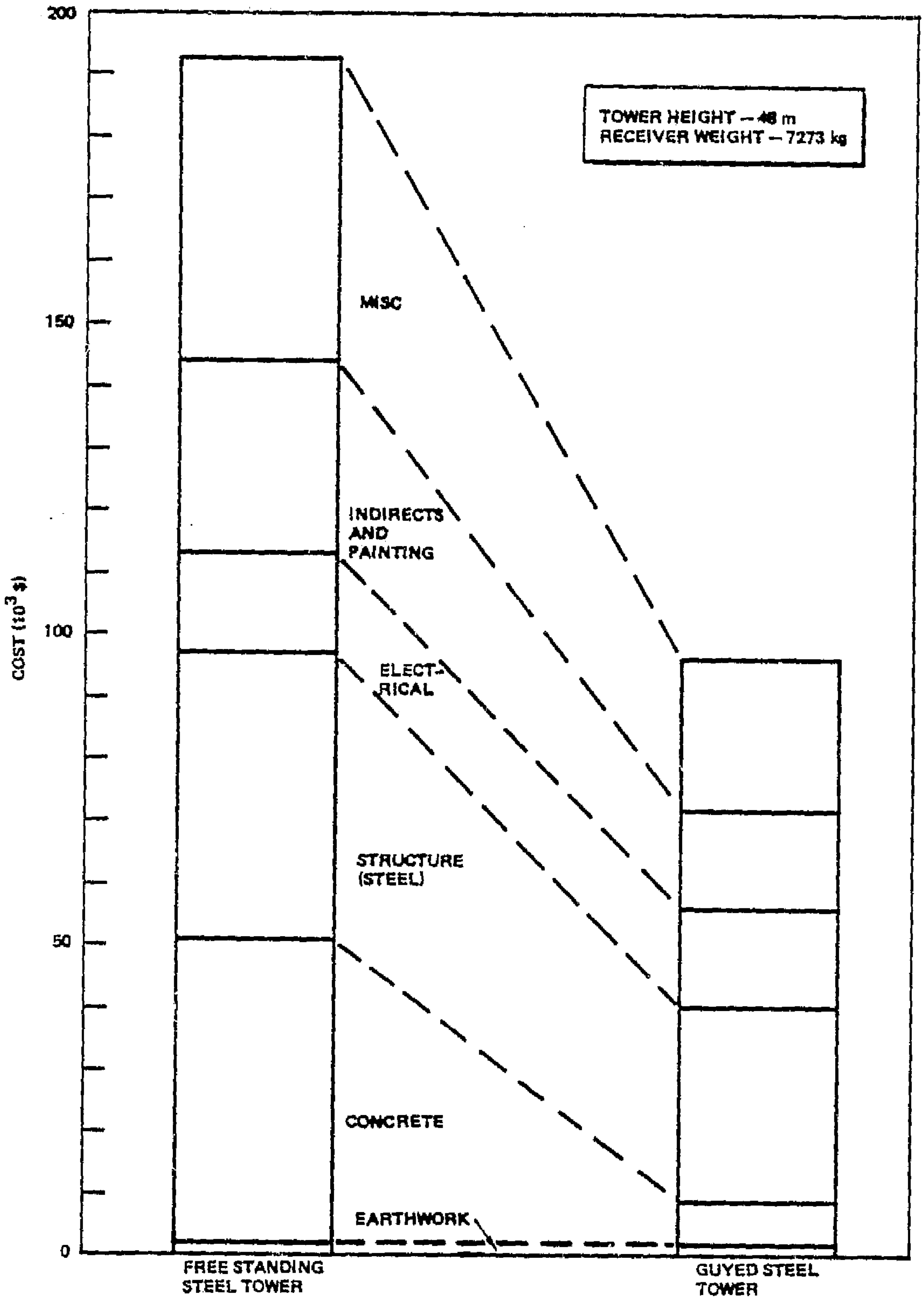


Figure 3.3-112. Comparative Tower Cost Breakdown

3.3.6.1 Requirements

Subsystem optimization for thermal storage was based on a system capacity factor of 0.4. This defined the turbine demand requirement relative to the average daily power collection curve derived from Barstow insolation data. From this relationship, the minimum storage capacity was determined to be equivalent to 2.75 hours of rated turbine operation. The required thermal storage capacity will therefore be a function of the power conversion subsystem and can be calculated from the cycle efficiency.

$$Q_s = \frac{T_s P_r}{\eta_c}$$

Q_s = Storage capacity, MWHt
T_s = Minimum storage at rated output, Hr
P_r = Plant rated output, MWe
η_c = Gross cycle efficiency

Design point storage capacity requirements as related to power conversion system variables are shown in Table 3.3-15 for the three startup times being considered. The thermal storage matrix, for which detailed cost estimates will be generated for system comparisons, is shown in Table 3.3-16.

3.3.6.2 Thermal Storage Subsystem Configurations

Schematics with basic subsystem components are shown in Figures 3.3-53 through 3.3-56 respectively for the following candidate thermal storage subsystems: dual media thermocline, two-tank, trickle charge, and pressurized water.

Pumps (shown for the two-tank configuration) were considered as energy transport components and were charged to that subsystem for cost evaluation. External piping was also incorporated in the energy transport subsystem.

Table 3.3-15

SYSTEM STORAGE REQUIREMENTS

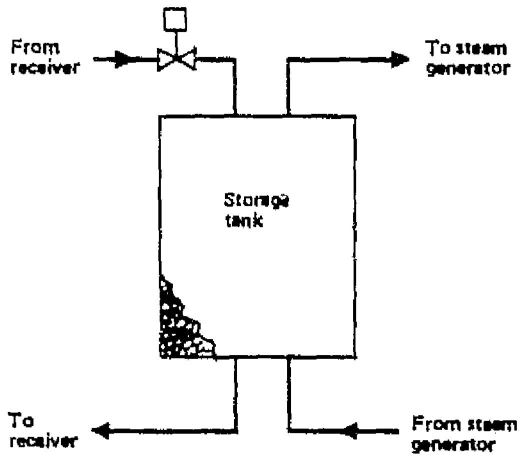
System Variables			Storage Capacity, MWHt		
			Program Duration		
Power Conversion Cycle	Cycle Efficiency	Temperature Range (°C)	3-1/2 Yr	4-1/2 Yr	6-1/2 Yr
Axial (Steam)	0.260	288-510	11.64		
	0.269	288-510		11.25	
	0.246	232-400	12.31		
	0.256	232-400		11.83	
	0.253	260-454	11.96		
Radial (Steam)	0.346	288-510			8.74
	0.280	232-400			10.81
	0.238	218-316			12.72
	0.20	109-252			16.0
Supercritical (Toluene)	0.284	232-400		10.66	
Subcritical (Toluene)	0.242	232-400	12.51		
	0.260	218-316	11.64		

Brayton - External storage requirement - 2.75 MWH_e

Table 3.3-16

THERMAL STORAGE SUBSYSTEM REQUIREMENTS

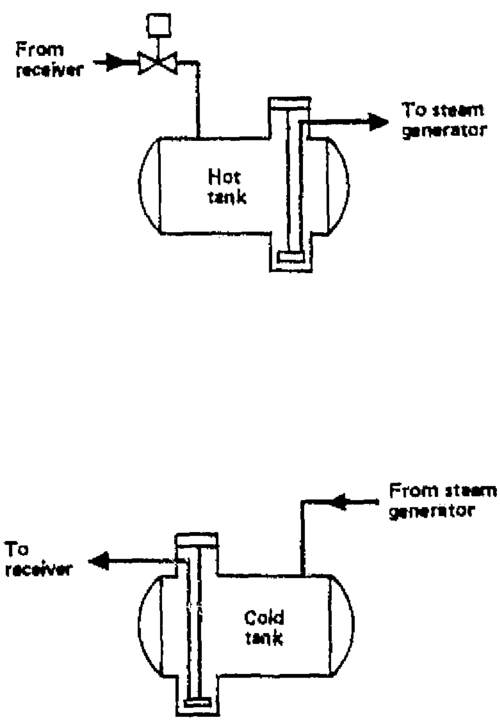
Storage Concept	Fluid	Temperature Limit (°C)	Storage Capacity (MWHt)
Dual Media Thermocline	Hitec	510	8.74-11.25
	Syltherm	400	10.66-12.31
	Caloria	316	11.64-12.72
Two Tank	Hitec	510	11.25-11.96
	Caloria	316	11.64-12.51
Trickle Charge	Syltherm	400	10.66-12.31
Pressurized Water	Water	315	16



Basic Components

1. Storage tank and manifold
2. Tank insulation
3. Solid media
4. Heat transfer fluid
5. Remote valve
6. Foundation

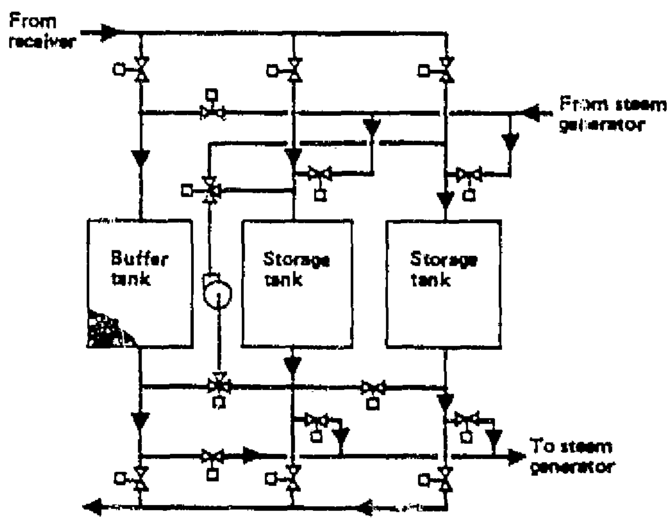
Figure 3.3-53. Dual Media Thermocline Energy Storage Subsystem



Basic Components

1. Hot tank
2. Cold tank
3. Tank insulation
4. Fluid
5. Remote valve
6. Foundation

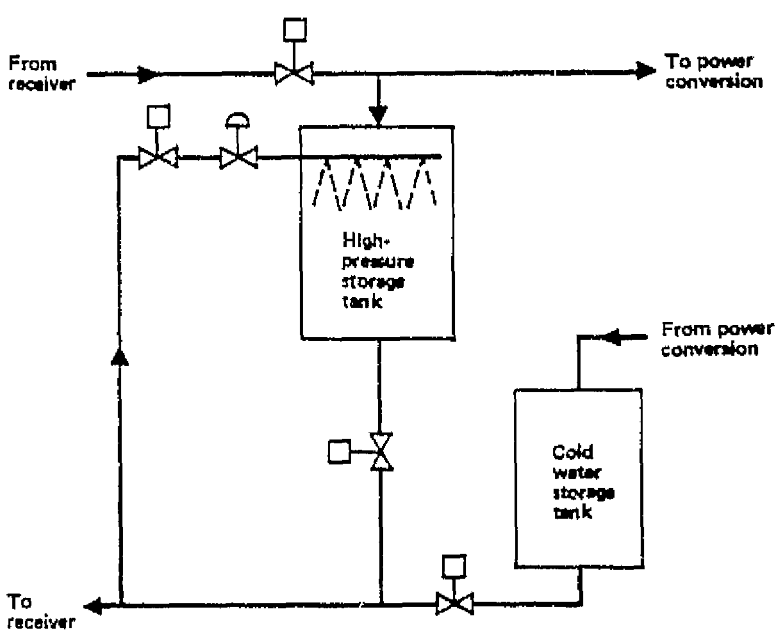
Figure 3.3-54. Two-Tank Energy Storage Subsystem



Basic Components

1. Buffer tank and manifolds
2. Storage tanks (2) and manifolds
3. Solid media
4. Heat transfer fluid
5. Tank insulation
6. Transfer pump
7. Remote two-way valves (13)
8. Remote three-way valves (2)
9. Foundation

Figure 3.3-55. Trickle Charge Energy Storage Subsystem



Basic Components

1. High-pressure storage tank
2. Cold water storage tank
3. Tank insulation
4. Remote valves (4)
5. Control valve

Figure 3.3-56. Pressurized Water Energy Storage Subsystem

3.3.6.3 HTS Thermal Storage Subsystems

The two storage concepts applicable for heat transfer salt were dual media thermocline and two-tank. Design descriptions for these cases are shown in Tables 3.3-17 and 3.3-18. Typical costs for two reference subsystems are shown in Table 3.3-19. Capital costs as a function of thermal storage capacity are illustrated in Figure 3.3-57.

The dual media storage tanks were sized two percent larger than required to allow for manifold and ullage space. An additional 10 percent capacity was also included because of the thermocline thickness. The void volume of the rock was assumed to be 25 percent. In the two-tank system, five percent excess fluid was added to allow for operations.

3.3.6.4 Syltherm Thermal Storage Subsystems

Trickle charge and dual media storage subsystems were analyzed for use with Syltherm. Design descriptions are shown in Tables 3.3-20 and 3.3-21. An example of reference costs is included in Table 3.3-22. Capital costs are illustrated in Figure 3.3-58 as a function of thermal storage capacity.

Trickle charge storage tanks were oversized by two percent to allow for distribution manifolds and five percent excess iron ore was assumed. A 45 percent void volume was assumed for the iron ore.

3.3.6.5 Caloria HT 43 Thermal Storage Subsystems

The two storage concepts evaluated for use with Caloria were the dual media thermocline (rock/fluid) and two-tank concepts. Design descriptions for the two systems are given in Tables 3.3-23 and 3.3-24. Reference costs are shown in Table 3.3-25. Capital costs as a function of thermal storage capacity are illustrated in Figure 3.3-59.

Table 3.3-17

DESIGN DESCRIPTION - DUAL MEDIA THERMOCLINE STORAGE SUBSYSTEM FOR HITEC ABOVE 450°C

Component	Material	Size	Description	Quantity
Storage Tank	Stainless 304 *[Carbon Steel]	Thickness 1.3 cm (0.5 in.)	Vertical, Shop Fabricated	1
Insulation and Cover	Fiberglass	Thickness 31 cm (12 in.) *[28 cm (11 in.)]	High Temperature	Varies
Solid Media	Granite	-	Rock/Sand 2:1 Mixture	Varies
Fluid	Hitec	-	Heat Transfer Salt	Varies
Valve	Stainless *[Carbon Steel]	6.4 cm (2.5 in.)	Remote Off/On	1
Foundation	Concrete			

*Below 450°C

Table 3.3-18

DESIGN DESCRIPTION - TWO TANK STORAGE SUBSYSTEM FOR HITEC BELOW 450°C

Component	Material	Size	Description	Quantity
Storage Tanks	Carbon Steel	Thickness 1.3 cm (0.5 in.)	Horizontal, Shop Fabricated	2
Insulation and Cover	Fiberglass	Thickness 28 cm (11 in.)	High Temperature	Varies
Fluid	Hitec	-	Heat Transfer Salt	Varies
Valve	Carbon Steel	6.4 cm (2.5 in.)	Remote Off/On	1
Foundation	Concrete			

Table 3.3-19

REFERENCE COSTS FOR HITEC STORAGE SUBSYSTEMS

Requirements		
Storage Technique	Dual Media	Two Tank
Power Conversion	Radial (Steam)	Axial (Steam)
Turbine Efficiency	0.80	0.67
Storage Capacity, MWHt	8.74	11.96
Temperature Range, °C	288-510	260-454
Costs		
Tanks	50,000	36,300
Insulation (Installed)	12,500	29,700
Rock/Sand	2,100	-
Hitec	20,700	116,539
Valves (+ Inst.)	1,700	1,300
Installation, Site Prep, Foundation, Miscellaneous	27,300	49,300
Total Cost	$114,300	$233,100

3.3.6.6 Pressurized Water Storage

A design description of the pressurized water storage system is given in Table 3.3-26 with reference cost data shown in Table 3.3-27 based on storage requirements for a radial turbine operating at an inlet pressure of 6.2×10^5 N/M^2 (90 psi); this is the highest inlet pressure possible with the condensing turbine due to moisture limitations unless reheat is used. The pressure vessel was sized to operate at 4.1×10^6 N/M^2 (600 psi).

3.3.6.7 Battery Storage

Batteries considered as feasible candidates are discussed below with respect to performance, development status, and cost projections. Pertinent data are shown in Table 3.3-28.

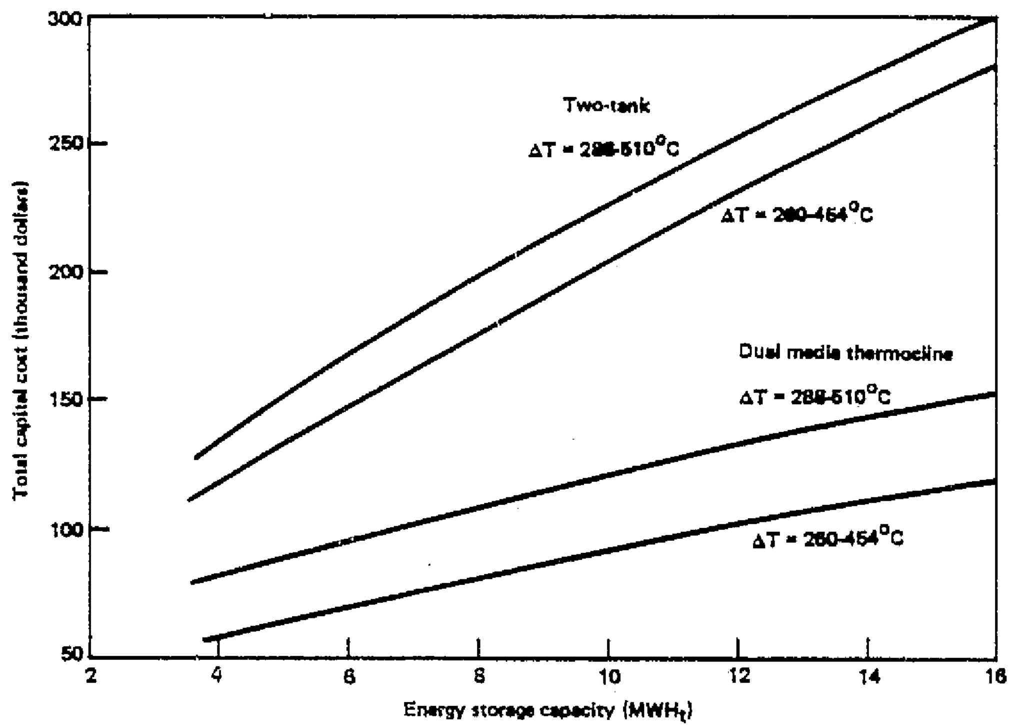


Figure 3.3-57. Capital Cost of HTS Energy Storage Subsystem

Lead-Acid

Lead Acid batteries have been in use for nearly a century. They require cooling to prevent life shortening caused by high rate charging and discharging. The industrial battery is estimated to cost \$70-90/KWH (Reference 40) and provide 1500 discharge cycles over its life depending on the level of discharge (Reference 41).

Sodium-Sulfur

Research has been conducted with this battery system at General Electric since 1967 (Reference 42). Molten sodium is used as the anode and a mixture of molten sulfur and sodium polysulfide as the cathode. The cell operates at approximately 300°C and a cooling system is required. The major problems encountered in the development of this system are corrosion of the sulfur container and failure of the seal between the solid electrode (Beta-Alumina) and the insulating header. Costs for these systems have been estimated at \$37.50/KWHe with development expected by 1985 (Reference 43).

Table 3.3-20

DESIGN DESCRIPTION - DUAL MEDIA THERMOCLINE STORAGE SUBSYSTEM FOR SYLTHERM (232-400°C)

Component	Material	Size	Description	Quantity
Storage Tank	Carbon Steel	Thickness 1.3 cm (0.5 in.)	Vertical, Shop Fabricated	1
Insulation and Cover	Fiberglass	25 cm (10 in.)	High Temperature	Varies
Solid Media	Iron Ore	Pellets	63% Fe	Varies
Fluid	Syltherm	-	Silicone Fluid	Varies
Valve	Carbon Steel	10.2 cm (4 in.)	Remote Off/On	1

Table 3.3-21

DESIGN DESCRIPTION - TRICKLE CHARGE STORAGE SUBSYSTEM FOR SYLTHERM (288-510°C)

Component	Material	Size	Description	Quantity
Storage Tank	Carbon Steel	Thickness 1.3 cm (0.5 in.)	Vertical, Shop Fabricated	3
Insulation and Cover	Fiberglass	25 cm (10 in.)	High Temperature	Varies
Solid Media	Iron Ore	Pellets	63% Fe	Varies
Fluid	Syltherm	-	Silicone Fluid	Varies
Valves	Carbon Steel	10.2 cm (4 in.)	Remote, Two-Way	13
Valves	Carbon Steel	10.2 cm (4 in.)	Remote, Three-Way	2
Pump	Carbon Steel	10.2 cm (4 in.)	Centrifugal, Inline	1
Foundation	Concrete			

Table 3.3-22

REFERENCE COSTS FOR SYLTHERM STORAGE SUBSYSTEMS

Requirements		
Storage Technique	Trickle Charge	Dual Media
Power Conversion	Supercritical (Toluene)	Radial Steam
Turbine Efficiency		0.8
Storage Capacity, MWHt	10.66	10.88
Temperature Range, °C	232-400	232-400
Costs		
Tanks	42,300	17,900
Insulation	23,200	12,700
Iron Ore	12,800	11,000
Syltherm	20,100	65,500
Valves (+ Inst.)	29,700	1,900
Installation, Site Prep, Foundation, Miscellaneous	56,200	29,700
Total Cost	$184,300	$138,700

Lithium - Iron Sulfide

The principal development work on this battery system has been conducted at Argonne National Laboratory (Reference 44) and Atomics International (Reference 45). Operating at approximately 400°C, Molten $LiC\ell$-$KC\ell$ salt is used as an electrolyte. The positive electrode is composed of FeS or FeS, while a Lithium-Silicon or Lithium-Aluminum alloy is utilized as the negative electrode. Development of this battery may require 3-5 years and will cost approximately $50/KWH (Reference 46). The major problem is the scale up from laboratory cells to modules of practical size.

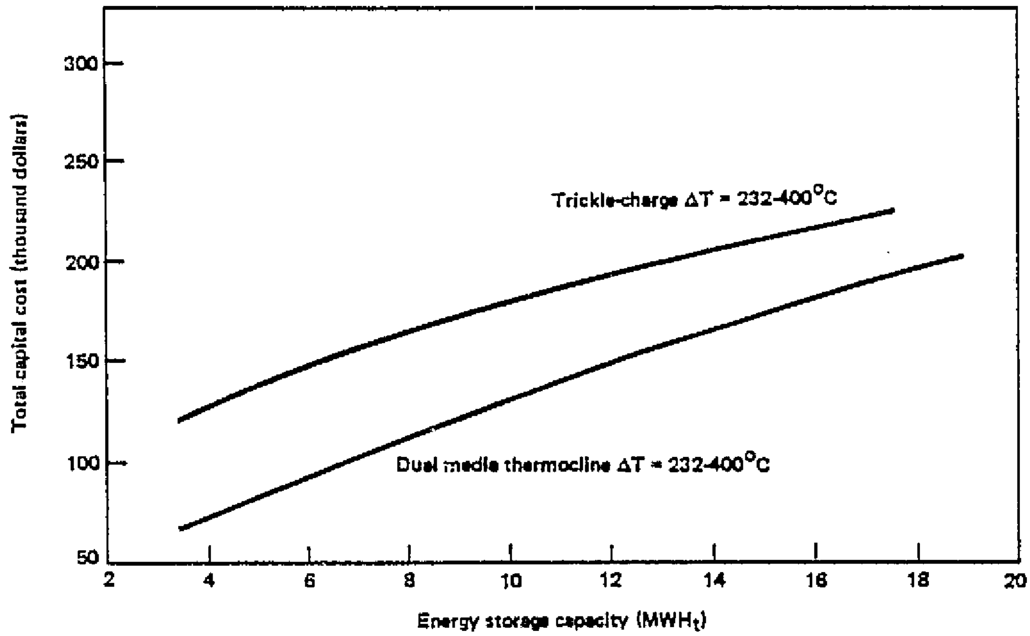


Figure 3.3-58. Capital Cost of Syltherm Energy Storage Subsystem

Table 3.3-23

DESIGN DESCRIPTION - DUAL MEDIA THERMOCLINE STORAGE SUBSYSTEM FOR CALORIA (218-316°C)

Component	Material	Size	Description	Quantity
Storage Tank	Carbon Steel	Thickness 1.3 cm (0.5 in.)	Vertical, Shop Fabricated	1
Insulation and Cover	Fiberglass	Thickness 20 cm (8 in.)	High Temperature	Varies
Solid Media	Granite	-	Rock/Sand 2:1 Mixture	Varies
Fluid	Caloria	-	Heat Transfer Oil	Varies
Valve	Carbon Steel	12.7 cm (5 in.)	Remote Off/On	1
Foundation	Concrete			

Table 3.3-24

DESIGN DESCRIPTION - TWO TANK STORAGE SUBSYSTEM
FOR CALORIA (218-316°C)

Component	Material	Size	Description	Quantity
Storage Tanks	Carbon Steel	Thickness 1.3 cm (0.5 in.)	Horizontal, Shop Fabricated	2
Insulation	Fiberglass	Thickness 20 cm (8 in.)	High Temperature	Varies
Fluid	Caloria	-	Heat Transfer Oil	Varies
Valve	Carbon Steel	12.7 cm (5 in.)	Remote Off/On	1
Foundation	Concrete			

Table 3.3-25

REFERENCE COSTS FOR CALORIA STORAGE SUBSYSTEMS

Requirements		
Storage Technique	Dual Media	Two-Tank
Power Conversion	Subcritical (Toluene)	Subcritical (Toluene)
Cycle Efficiency	0.260	0.242
Storage Capacity, MWHt	11.64	12.51
Temperature Range, °C	218-316	218-316
Costs		
Tanks	24,000	54,600
Insulation (+ Inst.)	18,400	47,200
Rock/Sand	7,500	-
Caloria	10,300	54,400
Valves (+ Inst.)	2,500	1,900
Installation, Site Prep, Foundation, Miscellaneous	35,900	67,500
Total Cost	$98,600	$225,600

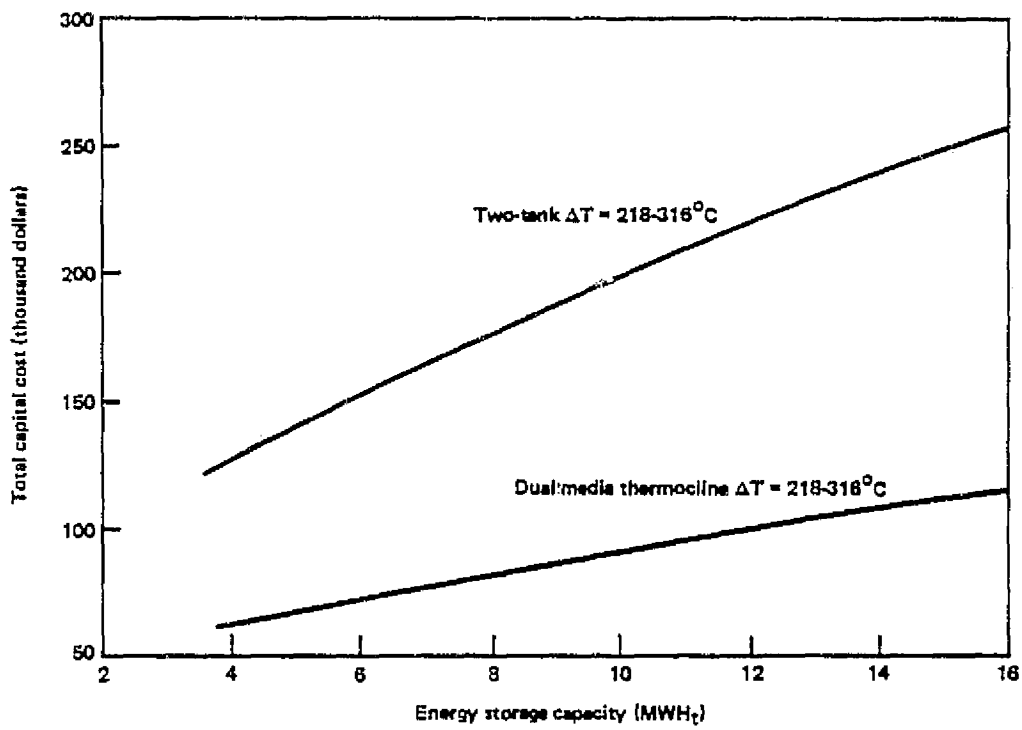


Figure 3.3-59. Capital Cost of Caloria Energy Storage Subsystem

Table 3.3-26

DESIGN DESCRIPTION - PRESSURIZED WATER STORAGE SUBSYSTEM

Component	Material	Size	Description	Quantity
Pressure Vessel	Carbon Steel	Thickness 5 cm (2 in.) 2.6 M I.D.	Cylindrical, Horizontal	1
Storage Tank	Carbon Steel	Thickness 0.8 cm (0.31 in.)	Vertical, Standard Tank	1
Valves	Carbon Steel	5 cm (2 in.)	Remote Off/On	4
Valve	Carbon Steel	5 cm (2 in.)	Flow Control	1
Foundation	Concrete			

Table 3.3-27

REFERENCE COSTS FOR PRESSURIZED WATER STORAGE SUBSYSTEM

Requirements	
Power Conversion	Radial
Turbine Efficiency	0.8
Storage Capacity, MWHt	15.4
Costs	
Pressure Tank	100,300
Storage Tank	7,500
Insulation	40,900
Valves	5,600
Installation, Foundation, Site Prep, Miscellaneous	21,800
Total Cost	$176,100

Table 3.3-28

BATTERY PERFORMANCE AND COST ESTIMATES

Battery System	Lead/ Acid	Sodium/ Sulfur	Lithium/ Iron Sulfide	Zinc/ Chlorine
Projected Development Time, Years	Current	7	3-5	3
Life Cycles Goal	1500	2500	2500	1000-2000
Efficiency Goal, %	87	75	80	65
Projected Costs				
Battery and Cooling, $/KWHe	70-90	38	50	62*
Building Installation Foundations, $/KWHe	20-35	5-20	5-20	--
Converter and Transformer, $/KWe	70-80	60-70	60-70	75*

*Cost estimated for a 100 MHW plant.

Zinc-Chlorine

This battery system is based on the use of Aqueous Zinc-Chloride as the electrolyte operating at 10-50°C (Reference 47). The chlorine evolved is externally stored as a solid chlorine hydrate. The electrolyte is recirculated during discharge to the cells. The low cost of reactants makes this an attractive concept but costs can only be kept within reason with large scale storage (100 MWH) and mass production. Current concepts require auxiliary equipment in each battery module such as, hydrate container gas, pumps and motors, electrolyte pumps, $H_2/C\ell_2$ reactor, and 3 heat exchangers. In addition, refrigeration equipment, and power conditioning equipment are required. Porous graphite for electrodes is also a major cost item. Assuming a production rate of one 100 MWH plant per year, costs are estimated at $64/KWH.

It appears that only lead/acid batteries will be available for the 3-1/2 and 4-1/2 year development programs. The Sodium/Sulfur system seems to be the best prospect for use in the commercial (10 year) plant with Lithium/Iron Sulfide as a possible alternative.

3.3.5.8 Life Cycle Costs

Life cycle costs were calculated for sensible heat storage subsystems and compared to life cycle costs for a typical battery storage subsystem. The life cycle costs are comprised of the following three components:

(1) Capital cost
(2) Annual fluid makeup cost or battery replacement cost
(3) Annual maintenance costs
(4) Efficiency cost penalty

The maintenance costs (excluding makeup and replacement) were assumed to be comparable for alternative storage concepts.

The replacement costs result from the finite life time associated with batteries and will be incurred on a regular basis. The efficiency of a battery, on the other hand, will result in a direct capital cost penalty because the field size and related cost must be increased to makeup for the loss of energy routed through storage. The assumption for battery cost

C-3

estimates are essentially those projected for Sodium-Sulfur systems. The larger value of installation costs is used because of the moderate size facility.

- Life cycles 2500
- Efficiency 75%
- Capital Cost 58 $/KWHe
  +65 $/KW

Separate cost entries were calculated for initial capital cost, fluid makeup, battery replacement, and reduced efficiency through storage. Additional costs resulting from the battery recovery efficiency utilized a ratio of energy sent directly to the user compared to energy directed through storage of three to one. This is consistent with an annual load factor of 0.4. A severe penalty is associated with low battery efficiencies and replacement requirements at 7.5-year intervals. On this basis the cost of battery storage is high even if the projected cost and performance goals are met.

3.3.7 Energy Transport

The energy transport subsystems were configured for each candidate to allow for system evaluation. The energy transport subsystem includes all the receiver fluid circulation and flow control equipment necessary to (1) provide coolant flow to the receiver in a controlled manner to maintain a constant receiver outlet temperature, (2) provide a steady state source of steam at design conditions for the turbine, and (3) charge or discharge the thermal storage subsystem as required. The energy transport subsystem must accommodate the design and operational characteristics of the collector, energy storage, and power conversion subsystems. In addition it must be compatible with all system operating modes.

3.3.7.1 Requirements

The requirements used to define the energy transport subsystem are shown in Table 3.3-29. A range of design conditions are shown to envelop the candidate systems and project startup times. In some cases, alternative power conversion cycles are being considered for the long term systems. However, the range of system requirements shown in the table should be representative even through final configurations may differ. With the definition of the system requirements, energy transport subsystem requirements can be determined. The principal design requirements of the energy transport subsystems are summarized in Table 3.3-30. Operating requirements for the energy transport subsystem include: (1) receiver startup, (2) steam generator startup, (3) receiver fluid at design temperature and flow rate range (normal operation), (4) operation from storage, and (5) system draining.

3.3.7.2 Energy Transport Subsystem Configurations

Energy transport schematics have been prepared for each of the candidate thermal storage concepts which survived the initial screening in Task 1A2. Schematics are presented in Figures 3.3-60 through 3.3-63 for the following thermal storage candidates: (1) two-tank, (2) dual media thermocline, (3) trickle charge, and (4) pressurized water. Receiver, thermal storage, and power conversion subsystem interfaces are shown in the schematics. Heat transfer salt, Syltherm, and Caloria fluids all require an inert cover gas. The gaseous nitrogen equipment which is intended to supply the inert environment has not been shown in the schematics for clarity.

Schematics depicting the operating modes are presented in Figure 3.3-64 through 3.3-68. The operating schematics are shown for the two tank thermal storage concept and are typical for all concepts. The operating modes include the following:

<u>Receiver Startup (Figure 3.3-64)</u> — During receiver startup, when the fluid leaving the receiver is below the acceptable temperature band, the receiver flow is controlled by control valve CV-1 with motorized valves RV-2 and RV-1 open and closed, respectively, to divert flow back to the cold tank. Once the design receiver outlet temperature is reached, valve RV-1 is opened and valve RV-2 is closed to allow flow into the hot tank.

<u>Steam Generator Startup (Figure 3.3-65)</u> — Steam generator warmup is accomplished by opening motorized valves RV-3 and RV-5 and metering the hot and cold fluid by control valves CV-2 and CV-3, respectively. Once the steam generator has reached acceptable temperature levels, valve RV-5 is closed and steam generator design flow rate is metered by control valve CV-2.

Table 3.3-29

RANGE OF DESIGN CONDITIONS FOR CANDIDATE SYSTEMS

Receiver Fluid	HTS		Syltherm 800		Caloria HT 43		Saturated Steam
Project Startup	Near Term	Long Term	Near Term	Long Term	Near Term	Long Term	Long Term
Power Conversion Cycle	Axial Steam Turbine	Radial Steam Turbine	Axial Steam Turbine	Radial Steam Turbine	Subcritical Toluene	Radial Steam Turbine	Radial Steam Turbine
Receiver Temperature, °C							
Outlet	450	600	400	400	315	315	252
Inlet	260	288	232	232	218	218	109
Steam Generator Power, MWt	4.35	3.03	4.47	3.93	4.55	4.62	5.50
Peak Receiver Power, MWt	6.61	4.61	6.80	5.97	6.91	7.02	8.36
Storage Capacity, MWHt	12.0	8.3	12.3	10.8	12.5	12.7	16.0

Table 3.3-30

ENERGY TRANSPORT DESIGN CONDITIONS

Receiver Fluid	HTS	Syltherm 800	Caloria HT 43	Saturated Steam
Peak Receiver Flowrate, Kg/Hr	34,800-80,400	61,300-69,800	86,800-88,300	12,200
Receiver Supply and Return Line Size, cm	5.1-7.6	10.1	12.7	4.1
Receiver Pump Head Rise, Bar	12.6-15.2	7.3	13.1	47
Steam Generator Feed Flow Rate, Kg/Hr	23,100-53,200	43,300-49,200	58,400-59,400	8,500
Steam Generator Supply and Return Line Size, cm	3.8-6.4	10.1	10.1	3.8
Steam Generator Feed Pump Head Rise, Bar	3.0	3.0	3.0	-
Storage Concepts	2-Tank Thermocline	Trickle Charge Thermocline	2-Tank Thermocline	Pressurized Water

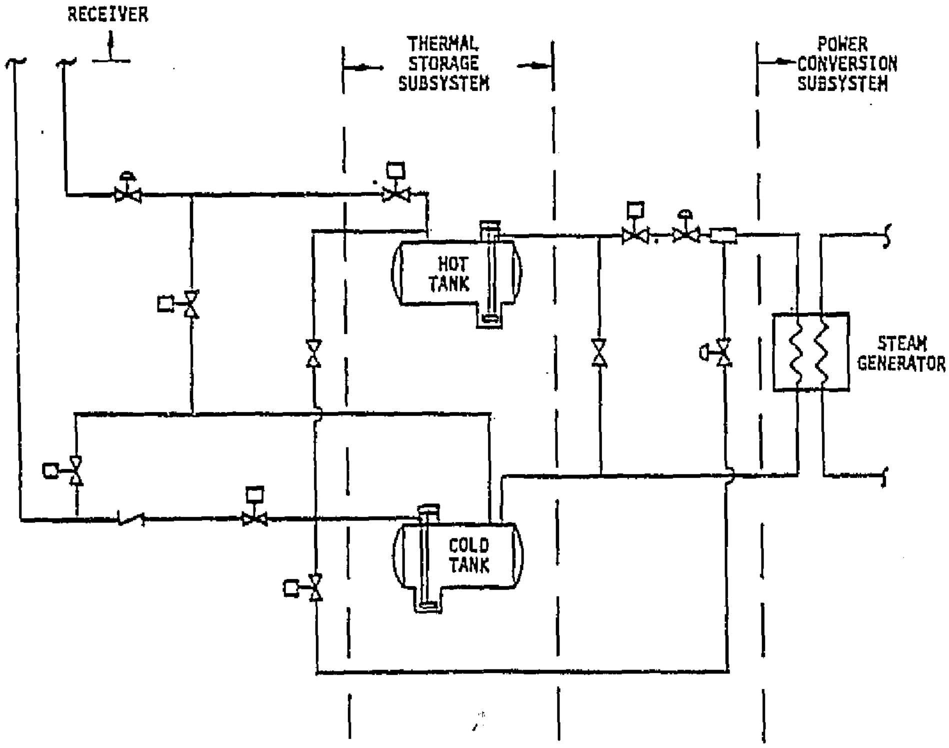


Figure 3.3-60. Two-Tank Energy Transport Configuration

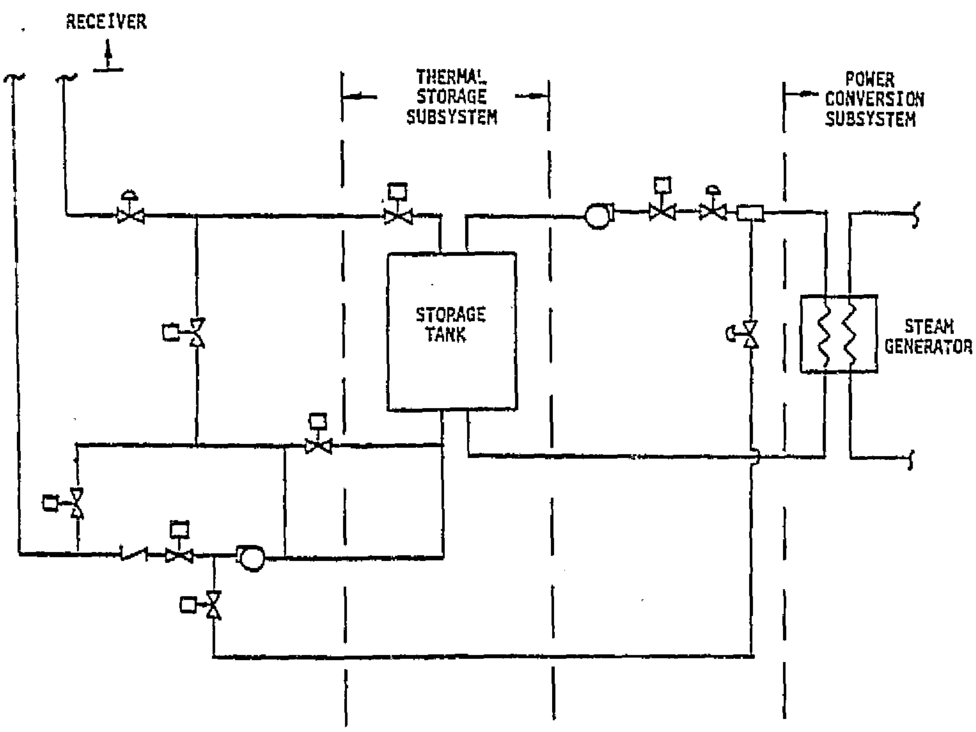


Figure 3.3-61. Dual Media Thermocline Energy Transport Configuration

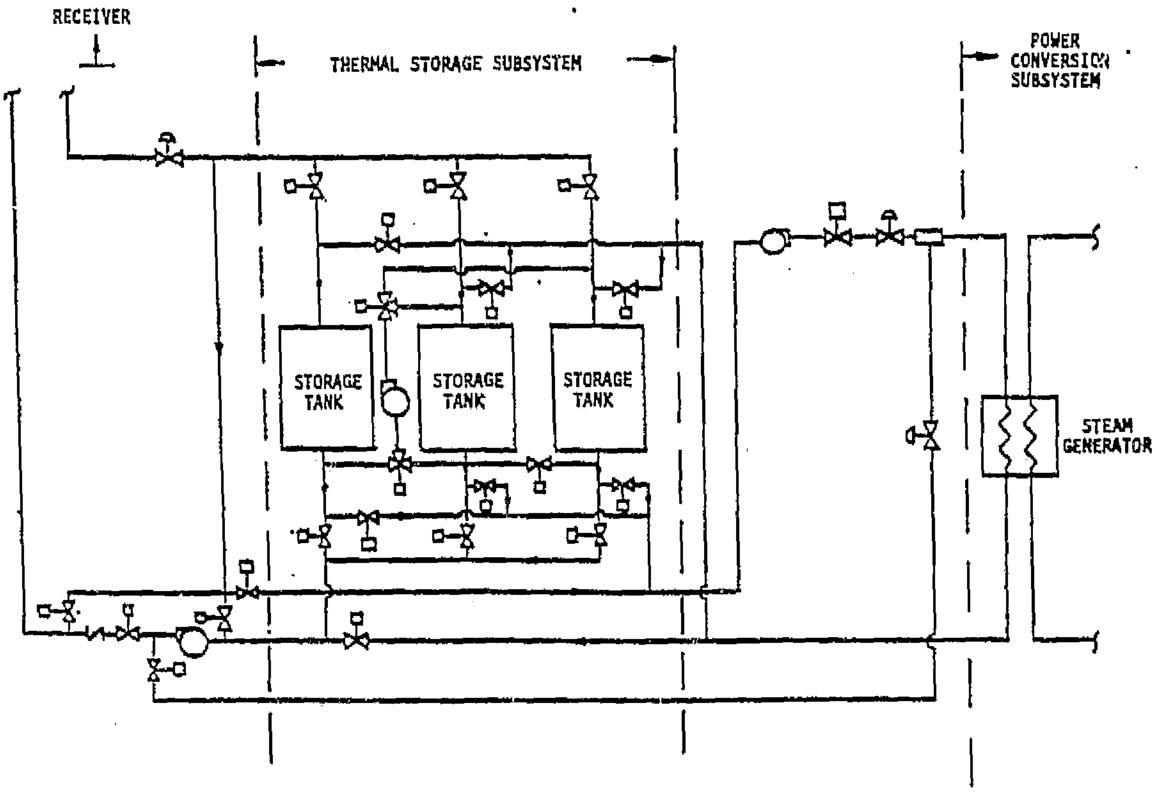


Figure 3.3-62. Trickle Charge Energy Transport Configuration

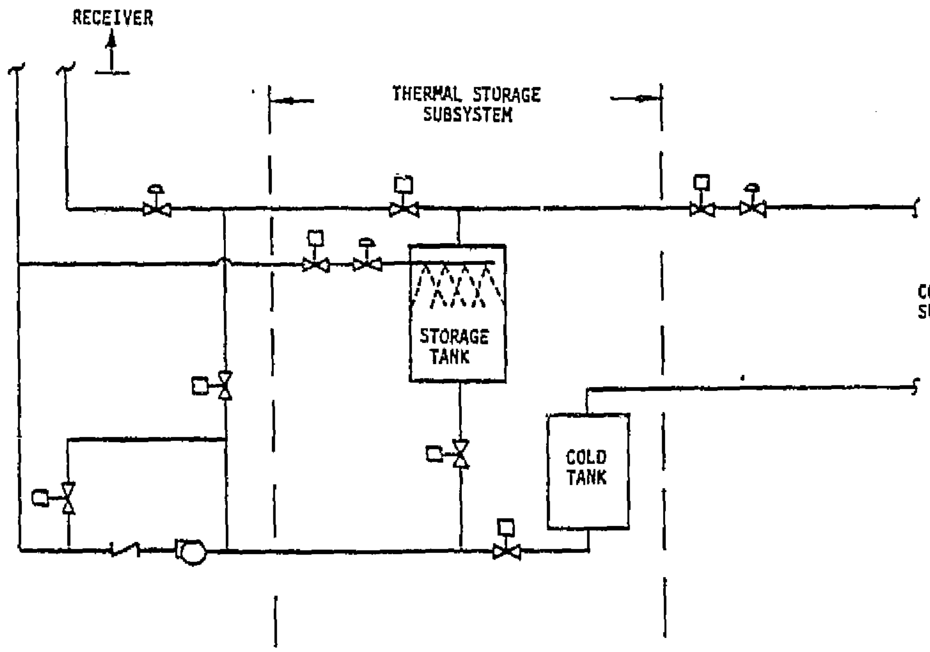


Figure 3.3-63. Pressurized Water Energy Transport Configuration

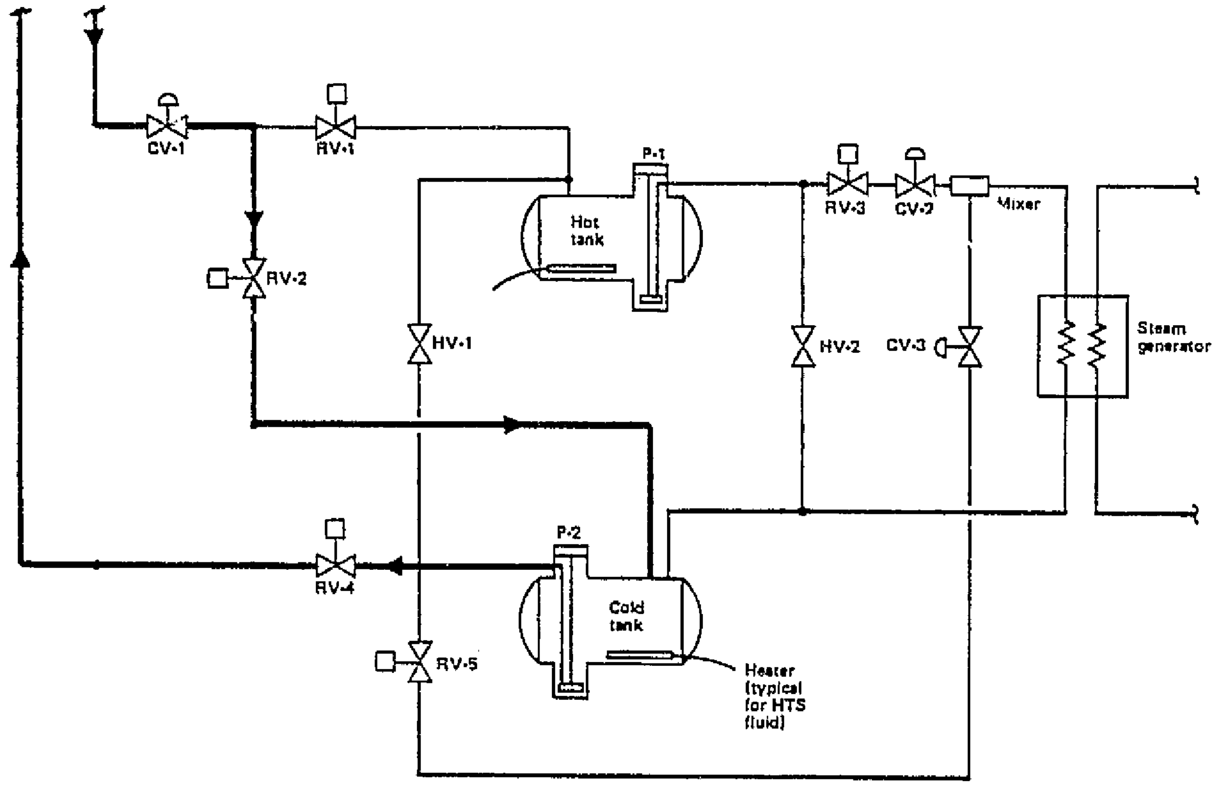


Figure 3.3-64. Energy Transport Operational Schematic — Receiver Startup

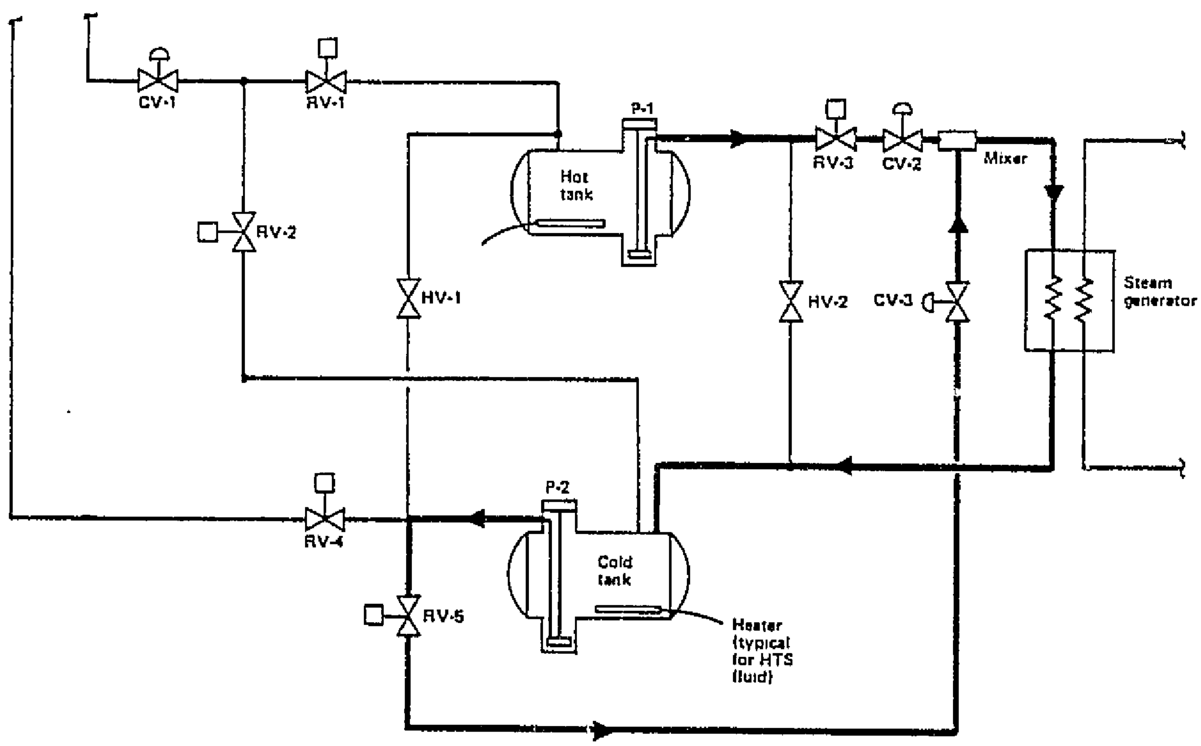


Figure 3.3-65. Energy Transport Operational Schematic — Steam Generator Startup

Normal Operation (Figure 3.3-66) — As the fluid comes to the required temperature, the proper flow rate is maintained by control valve CV-1. In this operational mode, the fluid travels from the cold storage tank to the hot storage tank. Simultaneously, when required, hot fluid is pumped to the steam generator and returned to the cold tank. All bypass lines are closed.

Operation from Storage (Figure 3.3-67) — After receiver collection has stopped, the energy in the thermal storage unit is used to operate the system. Fluid is pumped from the hot tank, metered by control valve CV-2, passes through the heat exchanger, and returned to the cold tank.

System Drain (Figure 3.3-68) — Motorized valve RV-2 is activated to drain the fluid loop and receiver. Manual valves are also included in the energy transport loop to allow direct transfer of fluid from one storage tank to the other storage tank.

3.3.7.3 HTS Energy Transport Loop

Thermal storage candidates for the HTS are: (1) two-tank, and (2) dual media thermocline. The two-tank concept is the near term candidate with the receiver outlet temperature limited to a maximum of 450°C. Thermocline systems will be considered at temperatures over 450°C and up to 600°C.

The energy transport subsystem component design descriptions for the two-tank loop are summarized in Table 3.3-31. Thermocline energy transport component descriptions for the under 450°C case are the same as the two-tank candidate with the deletion of the manual valves. Table 3.3-32 presents the component descriptions for the high temperature (over 450°C) thermocline loop. Energy transport subsystem costs are summarized in Table 3.3-33.

3.3.7.4 Syltherm Energy Transport Loop

Thermal storage candidates for the Syltherm fluid are (1) trickle charge (near term), and (2) dual media thermocline (long term). Maximum temperature

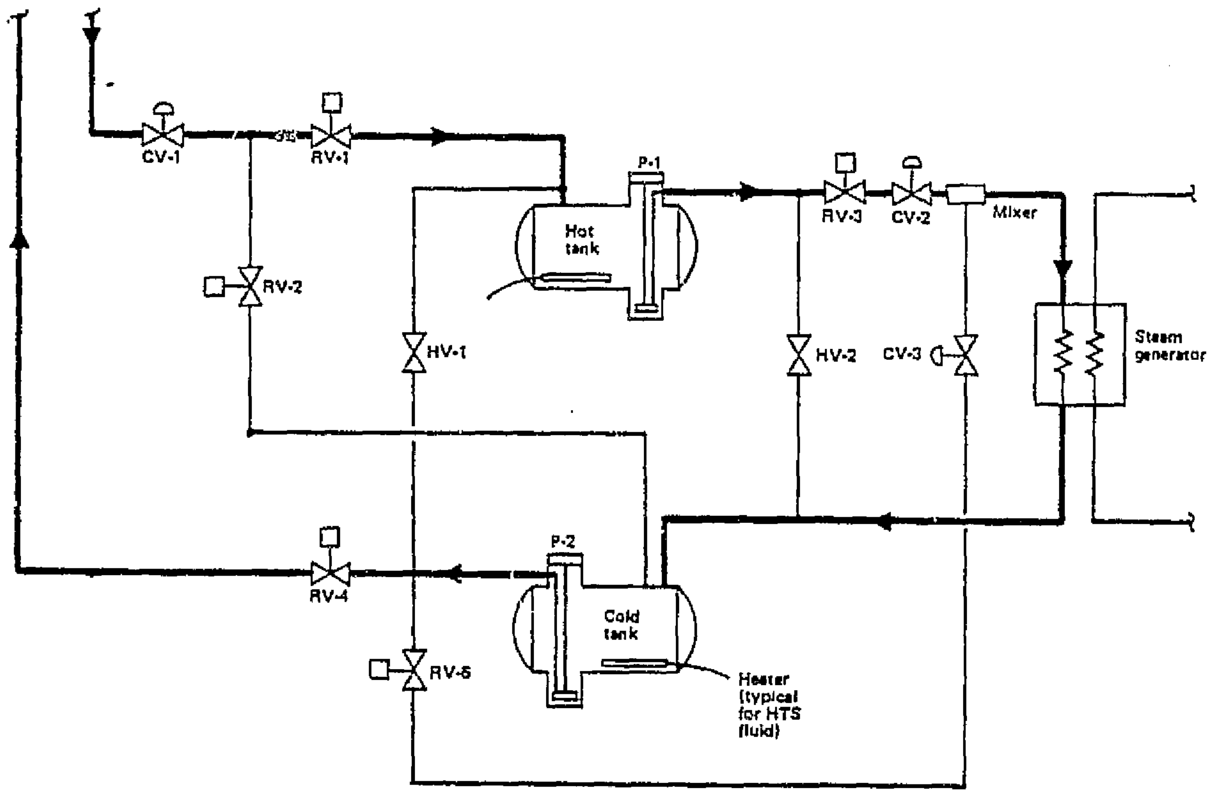


Figure 3.3-66. Energy Transport Operational Schematic — Normal Operation

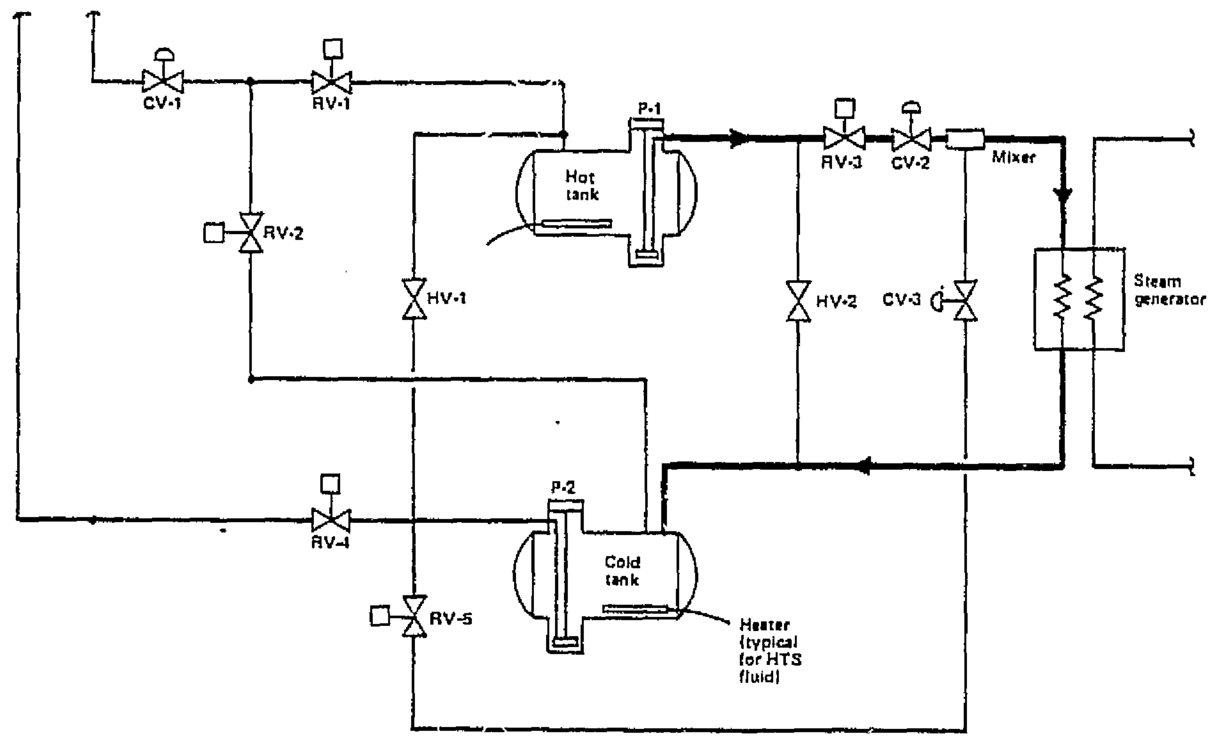


Figure 3.3-67. Energy Transport Operational Schematic — Operation from Storage

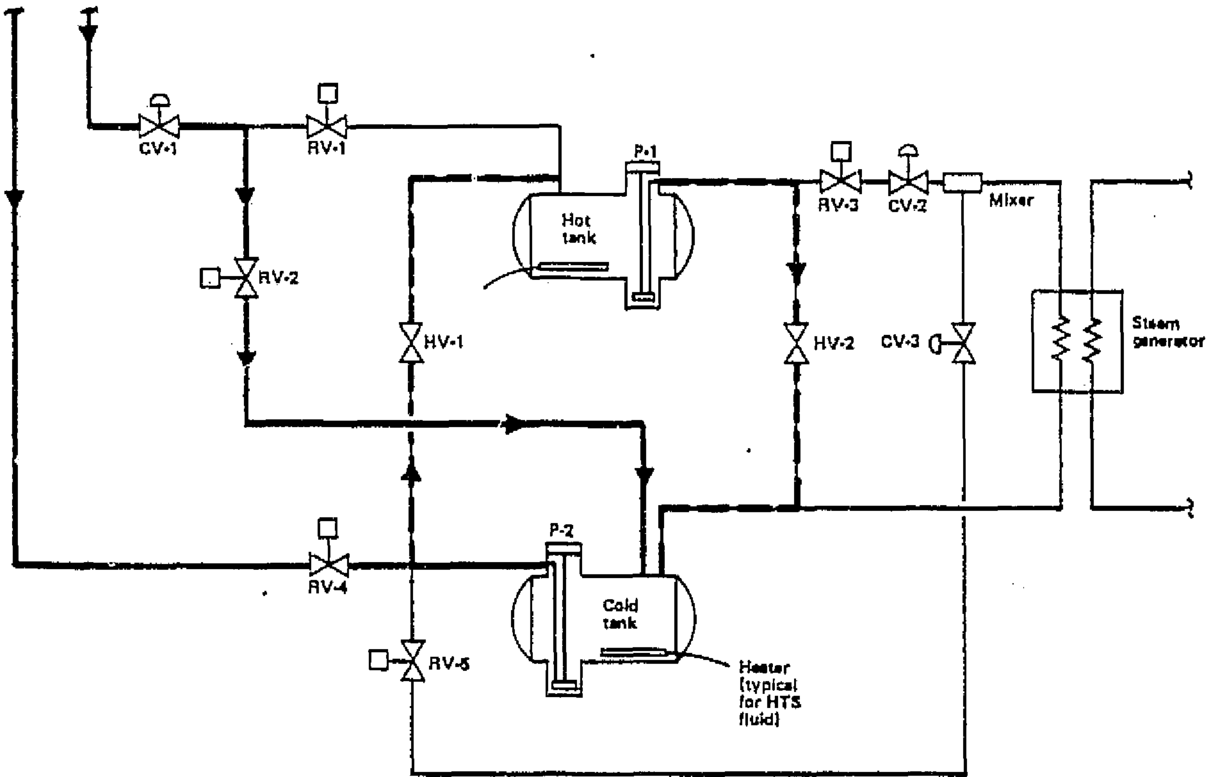


Figure 3.3-68. Energy Transport Operational Schematic — System Drain

of 400°C is assumed for both cases. The Syltherm energy transport design descriptions are nearly identical for both candidates and can be assumed to be the same for costing purposes. Component design descriptions and costs are summarized in Table 3.3-34 and Table 3.3-35, respectively.

3.3.7.5 Caloria HT 43 Energy Transport Loop

The near term thermal storage cnadidate for the Caloria HT 43 fluid is the two-tank concept. Component design descriptions for this transport loop are summarized in Table 3.3-36. Energy transport design conditions for the thermocline candidate (long term) are not significantly different than the two-tank loop. Caloria HT 43 energy transport costs are included in Table 3.3-35.

3.3.7.6 Water/Steam Energy Transport Loop

The only thermal storage candidate for the water/steam fluid is a pressurized water concept. Saturated steam conditions at the receiver outlet are 252°C and 41 bar. Component design descriptions and costs are provided in Tables 3.3-37 and 3.3-35, respectively.

Table 3.3-31

TWO-TANK HTS ENERGY TRANSPORT COMPONENT DESCRIPTION
(T_{max} < 450°C)

Component		Description
Receiver Feed Pump	Type	Centrifugal
	Head Rise	15.2 Bar
	Design Flow Rate	80,400 Kg/Hr
	Inlet/Outlet Size	7.6 cm
	Material	Carbon Steel
Steam Generator Feed Pump	Type	Centrifugal
	Head Rise	3.0 Bar
	Design Flow Rate	53,200 Kg/Hr
	Inlet/Outlet Size	6.4 cm
	Material	Carbon Steel
Valve, Remote	Type	Shutoff, Flow Control
	Size	7.6 cm - Receiver Circuit
		6.4 cm - Steam Generator Circuit
	Pressure Rating	21 Bar - Receiver Circuit
		10.5 Bar - Steam Generator Circuit
	Material	Carbon Steel
Valve	Type	Check
	Size	7.6 cm
	Pressure Rating	21 Bar
	Material	Carbon Steel
Valve, Manual	Type	Shutoff
	Size	3.8 cm
	Pressure Rating	10.5 Bar
	Material	Carbon Steel
Piping	Size	7.6 cm - Receiver Circuit
		6.4 cm - Steam Generator Circuit
	Schedule	40
	Material	Carbon Steel
Insulation	Thickness	10 cm
	Material	Calcium Silicate

Table 3.3-32

THERMOCLINE HTS ENERGY TRANSPORT COMPONENT DESCRIPTION
(T_{max} > 450°C)

Component		Description
Receiver Feed Pump	Type Head Rise Design Flow Rate Inlet/Outlet Size Material	Centrifugal 12.6 Bar 34,800 Kg/Hr 5.1 cm Carbon Steel
Steam Generator Feed Pump	Type Head Rise Design Flow Rate Inlet/Outlet Size Material	Centrifugal 3.0 Bar 23,100 Kg/Hr 3.8 cm Stainless Steel
Valve, Remote	Type Size Pressure Rating Material	Shutoff, Flow Control 5.1 cm - Receiver Circuit 3.8 cm - Steam Generator Circuit 21 Bar - Receiver Circuit 10.5 Bar - Steam Generator Circuit Stainless Steel - Receiver Outlet Steam Generator Inlet Carbon Steel - All Other Valves
Valve	Type Size Pressure Rating Material	Check 5.1 cm 21 Bar Carbon Steel
Piping	Size Material	5.1 cm - Receiver Circuit 3.8 cm - Steam Generator Circuit Stainless Steel - Receiver Outlet Steam Generator Inlet Carbon Steel - All Other Lines
Insulation	Thickness Material	10 cm Calcium Silicate

Table 3.3-33
HTS ENERGY TRANSPORT COST SUMMARY

Item	Two-Tank Storage T < 450°C	Thermocline Storage T < 450°C	Thermocline Storage T > 450°C
Pumps	21,600	18,000	22,200
Valves	15,400	13,300	13,600
Piping	5,300	5,300	4,900
Insulation	20,600	20,600	17,800
Electric Trace Heating	20,700	20,700	15,000
Total (1978 Dollars)	83,600	77,900	73,500

3.3.8 Power Conversion Subsystems

3.3.8.1 Power Conversion Requirements

The final selection of appropriate power generation subsystem candidates took into consideration the performance, operational, and environmental requirements discussed in Section 3.2.1.

Performance requirements emphasize high efficiency at design point operation and good part-load performance. Desirable operational characteristics include quick start-up and shut-down, minimal operating personnel, low maintenance costs, and high availability.

Environmental requirements limit the amount of noise and air pollution created by the plant, and esthetic considerations were also considered in the final design.

3.3.8.2 Prime Movers

The prime movers under consideration may be categorized as axial steam turbines, radial steam turbines, organic, vapor turbines, and gas turbines. The following sections present the advantages and disadvantages of each, performance and operational data, and equipment requirements and description.

Table 3.3-34

SYLTHERM ENERGY TRANSPORT COMPONENT DESCRIPTION
(T_{max} = 400°C)

Component		Description
Receiver Feed Pump	Type	Centrifugal
	Head Rise	7.3 Bar
	Design Flow Rate	69,800 Kg/Hr
	Inlet/Outlet Size	10.1 cm
	Material	Carbon Steel
Steam Generator Feed Pump	Type	Centrifugal
	Head Rise	3.0 Bar
	Design Flow Rate	49,200 Kg/Hr
	Inlet/Outlet Size	10.1 cm
	Material	Carbon Steel
Valve, Remote	Type	Shutoff, Flow Control
	Size	10.1 cm
	Pressure Rating	10.5 Bar
	Material	Carbon Steel
Valve	Type	Check
	Size	10.1 cm
	Pressure Rating	10.5 Bar
	Material	Carbon Steel
Piping	Size	10.1 cm
	Material	Carbon Steel
Insulation	Thickness	10 cm
	Material	Calcium Silicate

Table 3.3-35

ENERGY TRANSPORT COST SUMMARY

	Syltherm 800	Caloria HT 43	Water/Steam
Pumps	12,500	13,500	12,000
Valves	22,500	26,800	7,200
Piping	8,200	10,600	2,500
Insulation	22,300	23,600	18,500
Total (1978 Dollars)	65,500	74,500	40,200

Table 3.3-36

CALORIA HT 43 ENERGY TRANSPORT COMPONENT DESCRIPTION

Component		Description
Receiver Feed Pump	Type	Centrifugal
	Head Rise	13.1 Bar
	Design Flow Rate	88,300 Kg/Hr
	Inlet/Outlet Size	12.7 cm
	Material	Carbon Steel
Steam Generator Feed Pump	Type	Centrifugal
	Head Rise	3.0 Bar
	Design Flow Rate	59,400 Kg/Hr
	Inlet/Outlet Size	10.1 cm
	Material	Carbon Steel
Valve, Remote	Type	Shutoff, Flow Control
	Size	12.7 cm - Receiver Circuit 10.1 cm - Steam Generator Circuit
	Pressure Rating	21 Bar - Receiver Circuit 10.5 Bar - Steam Generator Circuit
	Material	Carbon Steel
Valve	Type	Check
	Size	12.7 cm
	Pressure Rating	21 Bar
	Material	Carbon Steel
Piping	Size	5.1 cm - Receiver Circuit 3.8 cm - Steam Generator Circuit
	Material	Carbon Steel
Insulation	Thickness	10 cm
	Material	Calcium Silicate

Axial Steam Turbine - Axial steam turbines offer the advantage of having many years of development to their credit. Performances and reliability can be accurately predicted and there are several suppliers of this equipment. The principal drawback in using small steam turbines is the low volume flow that is experienced at even moderate temperatures and pressures. This results in a poor turbine performance because of small steam path components with corresponding high frictional and leakage losses as measured on a percentage basis relative to the maximum potential shaft power. Steam also possesses a large enthalpy change per pound during expansion, requiring either many turbine stages or extremely high turbine speeds for efficient operation.

Table 3.3-37

PRESSURIZED WATER ENERGY TRANSPORT
COMPONENT DESCRIPTION

Component		Description
Receiver Feed Pump	Type	Centrifugal
	Head Rise	47 Bar
	Design Flow Rate	12,200 Kg/Hr
	Inlet/Outlet Size	5.1 cm
	Material	Carbon Steel
Valve, Remote	Type	Shutoff, Flow Control
	Size	5.1 cm - Steam Lines
		3.8 cm - Water Lines
	Pressure Rating	43 Bar - Steam Circuit
		10.5 Bar - Water Circuit
	Material	Carbon Steel
Valve	Type	Check
	Size	5.1 cm
	Pressure Rating	43 Bar
	Material	Carbon Steel
Piping	Size	5.1 cm - Receiver Circuit
		3.8 cm - Steam Generator Circuit
	Material	Carbon Steel
Insulation	Thickness	10 cm
	Material	Calcium Silicate

A survey of domestic and foreign manufacturers of multi-stage steam turbines was made. Results of this survey show that casing material requirements are set by inlet steam temperature and pressure. For conditions to 48 bar (700 psia) and 370°C (700°F), carbon steel is sufficient. For temperatures and pressures up to 510°C (950°F) and 62 bars (900 psia), a casing of chrome-molybdenum steel is required. Efficiencies for 48 bar (700 psia), 370°C (700°F) inlet range from 0.61 to 0.68 and for 510°C (950°F), and 62 bars (900 psia) efficiencies of 0.70 are obtainable.

The limits placed on operating pressure for turbines of this size is a major limiting factor in performance.

A turbine that is pressure limited will not gain much in cycle performance when operated at temperatures higher than that fixed by the pressure limitations.

Figure 3.3-69 is a schematic diagram of the axial steam turbine subsystems. A maximum of two feedwater heaters will be used, dependent on the manufacturer supplying the turbine. The feedwater heaters, cooling water loop, gearbox and generators, water treatment, and steam generator are discussed in subsequent sections.

Radial Outflow Steam Turbine - The radial outflow design offers several features which result in high expansion efficiencies, compared to axial machines with the same power output. Since the steam is introduced at the center and expands radially outward, the low volumetric flow stages have a small diameter and the higher volumetric flow stages are at a larger diameter.

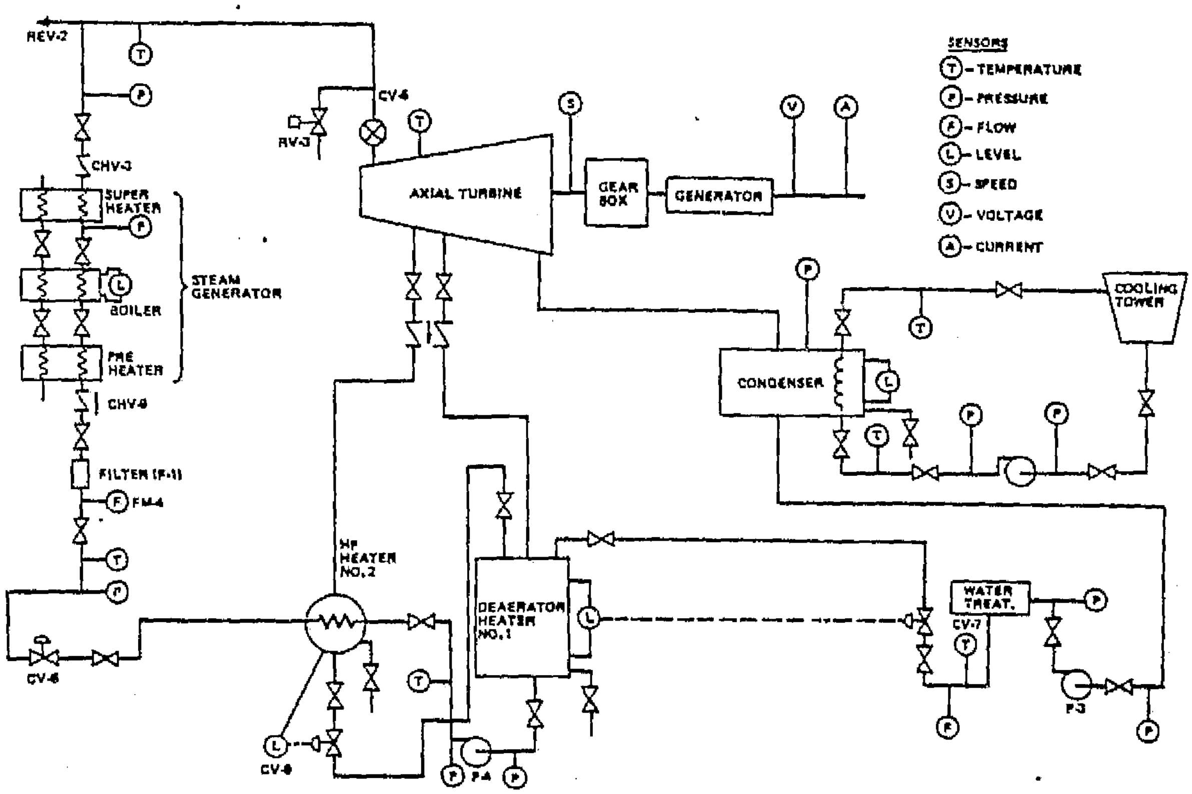


Figure 3.3-69. Power Conversion Subsystem Schematic (Water/Steam — Axial Turbine)

The single rotor disc can have a large number of stages resulting in subsonic steam velocities. This results in a high efficiency which is relatively insensitive to load and maintains good efficiency at off-design speeds.

Interstage steam leakage is reduced by the elimination of axial shaft seals necessary in axial machines and the use of fully shrouded blade rows with multiple labyrinth interstage seals. Provisions can be made for multiple extraction ports to provide for regenerative feedwater heating.

In addition to the above performance advantages, the radial outflow turbine has the potential for significant manufacturing and cost advantages in comparison with axial machines. The single rotor disc is mounted on the shaft in an overhung arrangement, leading to reduced housing and sealing requirements and a much more easily balanced shaft than with axial machines. Blade manufacturing costs are greatly reduced, since the blades are untwisted in a radial flow design.

Preliminary work performed by ETI has resulted in design, performance and cost estimates given in Table 3.3-38.

Table 3.3-38

DESIGN/PERFORMANCE/COST ESTIMATES FOR RADIAL OUTFLOW TURBINES

Inlet Conditions	No. of Stages	Turbine Efficiency	Extraction Pressures (Bars)	Second Unit Costs (Thousands)	100 Unit Cost (Thousands)
482°C/102 Bars (900°F/ 1500 psia)	10	0.85	26.5,6.8, 1.7,0.9,0.45	150	99
371°C/48 Bars (700°F/ 700 psia)	9	0.85	26.5,6.8, 1.7,0.9,0.45	143	95

A typical schematic diagram for the radial outflow steam turbine is given in Figure 3.3-70. This schematic shows provisions for five feedwater heaters. The steam generator, feedwater heaters, water treatment and generator and gearbox are discussed in subsequent sections.

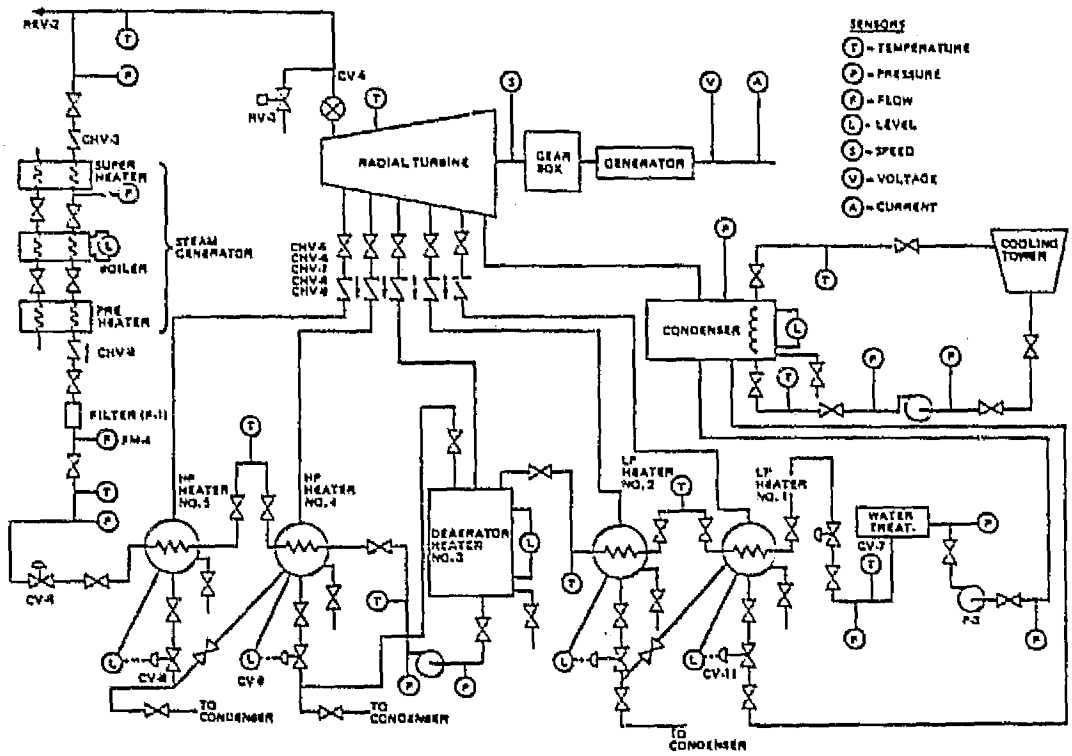


Figure 3.3-70. Power Conversion Subsystem Schematic (Water/Steam — Radial Turbine)

Organic Rankine Cycles — In recent years, there has developed considerable interest in Rankine cycles utilizing organic fluids. One of the major advantages of organic fluids is the relatively small enthalpy change upon expansion resulting in fewer turbine stages for efficient expansion and higher mass flow rates than with water, minimizing leakage losses. Also, many of the organic fluids have a positive saturation line and the vapor expands into the superheat region, thus eliminating any problem of blade erosion or efficiency loss from moisture formation. Many fluids have been considered for use in the past, including the freons, pyridine, benzene, chlorobenzene, Dowtherm A and Flutec PP3. The current choices for use in the moderate temperature range of 250 - 400°C (480-750°F) are toluene and Fluorinol -85. Fluorinol -85 is a non-toxic, non-flammable fluid with a temperature limit of 300°C (572°F) and a present cost of $17 per kilogram. Toluene offers a 400°C (750°F) upper limit and costs $2 per kilogram, but is toxic and flammable. Both fluids have positive saturation lines and are capable of supercritical operation.

Three manufacturers have organic turbines either under development or available for purchase that operate in the temperature range of interest. Sundstrand has a 600 kW organic turbine using toluene as the working fluid immediately available. The turbine is a single stage axial flow impulse turbine that runs at 10,100 RPM and is directly connected to the feed pump. The condenser and regenerator are located within a common housing and are of the tube and fin configuration. Turbine inlet conditions are 288°C (550°F) and 21 bars (305 psia) and with a condenser temperature of 60°C (140°F). The conversion efficiency to electricity is 20.2%. Thermoelectron is presently building a 500 KW organic turbine that utilizes Fluorinol-85. The turbine is a six stage axial machine designed to run at 7950 RPM. Inlet temperature of 288°C (550°F) and condenser temperature of 24°C (75°F) result in an expected cycle efficiency of 26%. Garrett/AiResearch has designed a toluene system utilizing a two stage axial turbine arrangement which has variable admission geometry. The generator runs at variable speed with a power conditioning unit converting the output to 60 cycle AC. Inlet temperatures of 288°C (550°F) should result in a 23.2% cycle efficiency. Part load performance of this unit is excellent due to the unique turbine-generator design. A schematic diagram of an organic turbine cycle is shown in Figure 3.3-71. The diagram is typical of all three manufacturers with minor modifications.

Gas Turbines — Many of the advantages and disadvantages of gas turbines have been discussed earlier in Section 3.2.4. The conclusions reached in that section were that battery storage was necessary. This requires the power generation subsystem to follow the thermal load throughout the day. A gas turbine acceptable for use in this mode must have good part-load performance. The logical candidates include open regenerative two-shaft machines and closed regenerative cycles. A schematic of these subsystems is shown in Figure 3.3-72. The turbines used in plants of this size consist of five to seven stages and obtain expansion efficiencies of 0.85 to 0.89. Compressors consist of 12 to 20 stages with an efficiency of 0.80 to 0.85.

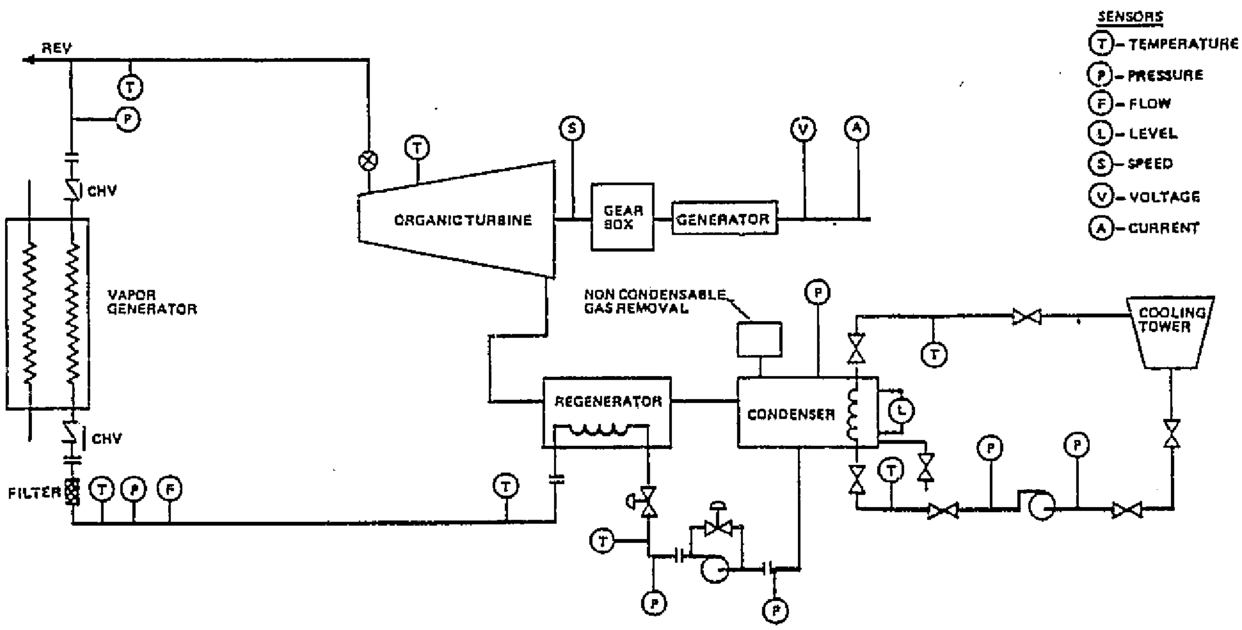


Figure 3.3-71. Power Conversion Subsystem Schematic (Organic Rankine Cycle)

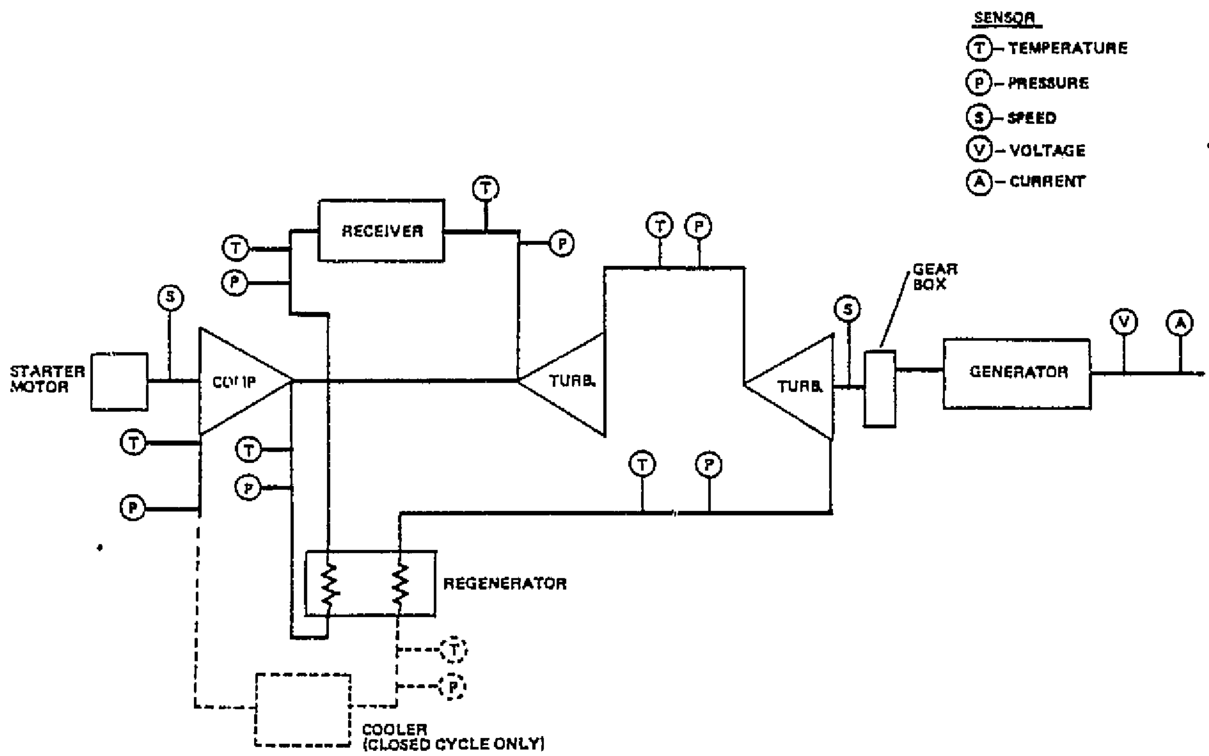


Figure 3.3-72. Power Conversion Subsystem Schematic (Brayton Cycle)

One of the most expensive components of a gas turbine plant is the regenerator, which uses the waste heat of the low pressure gas expanded in the turbine for preheating the high pressure gas and thus has the same function as the feedwater heaters in a steam Rankine cycle. However, while in a steam turbine the heat transferred via condensation is a small fraction of that added at the boiler, the heat transferred at the regenerator is nearly equal to the heat transfer at the heater and is a gas-to-gas transfer mechanism. The result is an expensive device whose performance is closely related to cost.

The performance of a regenerative Centaur gas turbine manufactured and modified by Solar Turbines International (Reference 48) is shown in Figure 3.3-73. This machine has a turbine inlet of 871°C (1600°F) and compressor inlet of 38°C and an estimated regenerator efficiency of 0.70-0.75. Improvement of the regenerator to 0.90 effectiveness and reduction of turbine inlet to 815°C (1500°F) should result in a cycle efficiency of 0.31 to 0.33.

Summary — A list of the Power Generation Subsystem candidates is given in Table 3.3-39. Steam axial and radial turbines were evaluated with both optimistic and conservative estimates of expansion efficiencies for operating temperatures compatible with Hitec, Syltherm and Caloria HT 43 heat transfer fluids. Axial turbines with the efficiencies assumed are immediately available. Radial turbines are development items that could be available for 4-1/2 and 6-1/2 year start-up times. Estimates of the cycle efficiency obtainable from organic Rankine cycles using toluene and Fluorinol-85 were provided by Sundstrand and ThermoElectron Corporations.

Performance estimates for a two-shaft regenerative gas turbine were obtained from Solar Turbine International. This same machine has been selected by Boeing Engineering and Construction and Black and Veatch Consuling Engineers for use in the EPRI Central Receiver Brayton Pilot Plant Design.

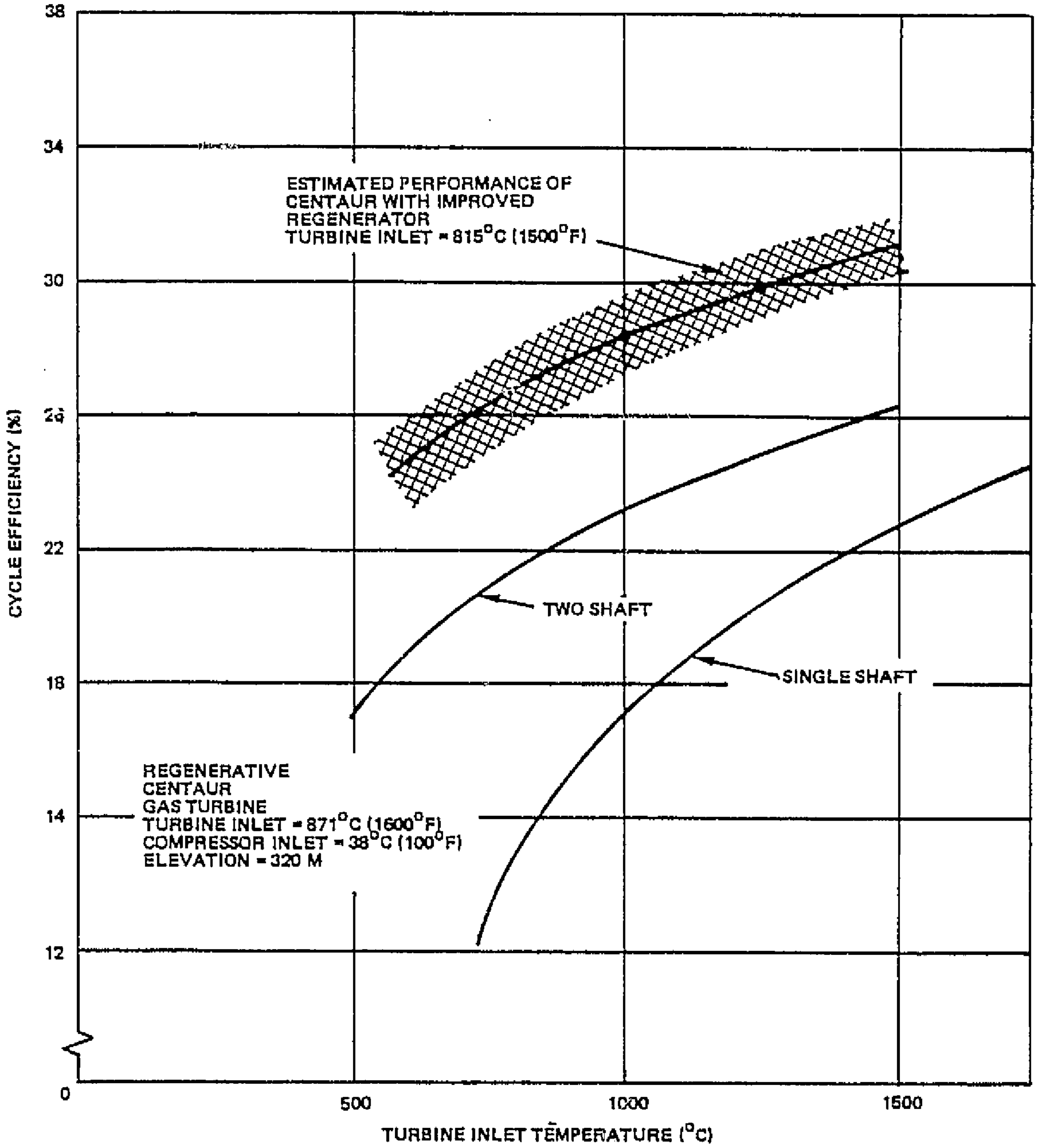


Figure 3.3-73. Gas Turbine Performance

Table 3.3-39

SUMMARY OF CANDIDATE POWER GENERATION SUBSYSTEMS

Cycle	Turbine Efficiency	Turbine Inlet	Vapor Generator Inlet	Cycle Efficiency	H.T. Fluid
Axial	0.67	480°C 62 Bars	163°C 62 Bars	0.260	Hitec
Axial	0.70	480°C 62 Bars	163°C 62 Bars	0.269	Hitec
Axial	0.67	371°C 48 Bars	152°C 48 Bars	0.246	Syltherm
Axial	0.70	371°C 48 Bars	152°C 48 Bars	0.256	Syltherm
Radial	0.80	480°C 102 Bars	218°C 102 Bars	0.346	Hitec
Radial	0.85	480°C 102 Bars	218°C 102 Bars	0.361	Hitec
Radial	0.80	371°C 31 Bars	199°C 31 Bars	0.280	Syltherm
Radial	0.85	371°C 48 Bars	199°C 48 Bars	0.292	Syltherm
Radial	0.80	288°C 14 Bars	162°C 14 Bars	0.238	Caloria HT 43
Radial	0.85	288°C 20 Bars	162°C 20 Bars	0.248	Caloria HT 43
Organic (Toluene)		385°C 62 Bars	180°C 62 Bars	0.284	Syltherm
Organic (Toluene)		288°C 21 Bars	150°C	0.242	Caloria HT 43
Organic (Fluorinol-85)	0.85	288°C 40 Bars	150°C	0.260	Caloria HT 43
Radial	0.85	160°C 6.2 Bars		0.200	Saturated Steam
Brayton		815°C 5 Bars		0.310	Air

3.3.8.3 Electrical Power Generation — This equipment converts mechanical shaft power to electricity. Shaft speeds range from 5,000 to 12,000 rpm for the steam, organic and gas turbines considered. The function of the gearbox is to reduce these speeds to the 1800 rpm shaft speed of a 4-pole generator. The gearbox usually consists of a single reduction, double helical, involute tooth gear with a steel casing and transmits power with an efficiency of 0.98 to 0.985. Gearbox accessories include a main oil pump driven by the gear shaft to provide lubrication for gears and turbine where needed. The lubrication system consists of a reservoir, tubular type oil cooler with by-pass piping and valves, strainer, oil level indicator and all necessary valves, gauges, thermometers and interconnecting piping. A motor operated auxiliary oil pump is provided for starting and operation during a failure of the main oil pump. Gear box assemblies of this type are available from a variety of suppliers.

The generator used is typically a 4-pole unit operating at 1800 rpm with open, drip-proof construction, class "F" insulation, static type voltage regulation and direct connected, brushless exciter. Electrical conversion efficiency is 0.96 and remains nearly constant at half-load. The generator control unit will include the necessary voltmeter and ammeters, transformers, voltage regulators, synchronizing switch gear and circuit breakers to allow operation on the utility grid.

3.3.8.4 Steam (Vapor) Generators

The steam or vapor generating equipment is required to transfer the thermal energy from the heat transfer fluid to the turbine working fluid. This heat transfer process is governed by the local temperature difference which can be maintained between the two fluids and depends heavily on the thermodynamic properties of each fluid. Figure 3.3-74 illustrates a typical steam generation process using Hitec as the heat transfer fluid. It is seen that the two critical regions in the heat transfer process occur at the point where saturated water is reached and at the superheated steam outlet. For a successful heat transfer process, it is necessary, at all points along the curve, that the heat transfer fluid have a higher temperature than the water/steam

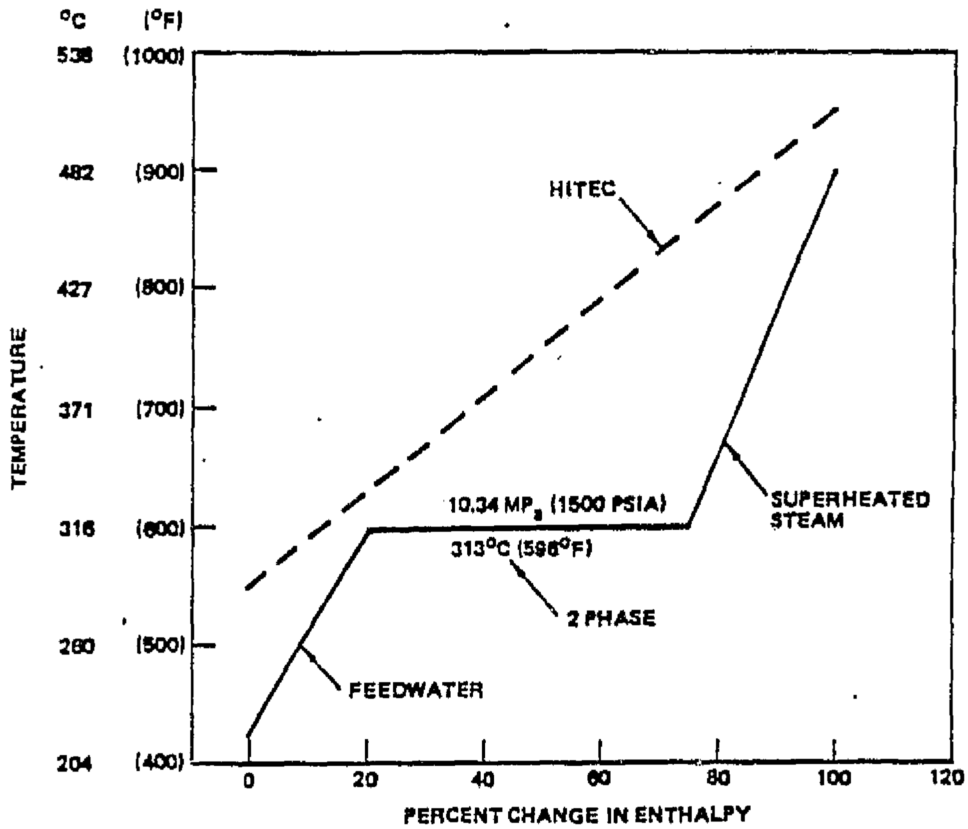


Figure 3.3-74. Steam Generation Process

or the organic fluid being heated. The local temperature differences combined with the total heat transfer requirement serve as the principal design requirements for the heat transfer equipment.

In reviewing candidate steam or vapor generation equipment three types of equipment can be considered: (1) once-thru boiling units; (2) kettle-boiler designs; and (3) recirculating drum-type equipment. Each of these have certain features and limitations which must be considered relative to the particular application under consideration.

The once-thru boiling equipment utilizes one or more continuous passes to convert inlet water or organic fluid to dry saturated or superheated vapor. No liquid separation equipment is contained in the design to ensure that no moisture is contained in the final exit vapor.

The principal advantage to this equipment is low heat exchanger cost due to overall design simplicity. From an operational standpoint, however, the once thru design requires close and moderately sophisticated control equipment to ensure that liquid does not pass through the equipment and on to the turbine. This is of particular concern during non-steady state periods of operation when local flow mismatches may occur between the heat transfer and turbine working fluids. This added control equipment partially offsets the cost advantage derived from the heat exchanger equipment itself.

Other adverse factors associated with the once-thru design include high water or fluid quality requirements and the need for an ancillary loop to be used during heat exchanger startup. From a fluid quality standpoint, once-thru equipment requires water to have less than 50 ppb dissolved solids with corresponding purity levels in the organic fluid systems. This is in contrast to the typical drum design requirement for inlet feedwater with dissolved solids less than 150 ppb and drum water quality of 750-1000 ppm. The reason for the high water quality for the once-thru equipment is to prevent solid deposition on the heat transfer surfaces near the dryout point. Since drum type equipment does not experience local dryout in the tubes, the problem of solid deposition is much less severe with this equipment.

For startup, the once-thru equipment requires a bypass circulation loop since the startup is initiated with all liquid flow. As the startup process is completed, the boiling region is established in the boiling zone and the resulting steam or vapor is then made available to the turbine. This additional equipment also adds to the overall cost of the steam or vapor generator which is an important factor for small installations such as the type currently under investigation because of the loss of certain economies of scale associated with the equipment.

Based on the adverse factors associated with operation, control, ancillary equipment cost, and high water quality, once-thru equipment has been eliminated from further consideration for this application. This choice also relaxes somewhat the requirement to have full-time, skilled operators available at all times to handle the operational complexity introduced by the once-thru equipment.

The kettle boiler and drum-type recirculating equipment represent the more traditional approaches to vapor or steam generation. From a cost standpoint, the kettle-boiler approach is clearly superior at low pressures, less than 1.38 MPa (200 psia), with the superiority falling off as pressure increases. This is because the kettle boiler has the high-pressure water/steam on the shell side of the heat-exchanging surfaces. Thus, as pressure and capacity increase, shell thicknesses also increase.

The drum-type boiler has the advantage that the high-pressure water/steam is contained in the tubes and the shell can therefore be maintained at the minimum allowable gage. The drum-type boiler, however, requires the use of an elevated high-pressure drum for the separation of steam from water. Thus, a pressure/cost relationship is also incurred with the drum configuration. In addition, the tubes must be located in a vertical or near vertical orientation to insure proper circulation, particularly if a natural circulation approach were used. If forced-circulation drum concepts were used, an additional recirculating pump would be required. For either recirculating concept a return line or downcomer would be required in the design to complete the recirculation pass, thus further complicating the configuration.

From a design standpoint, it should be noted that separate preheater and superheater devices are required to supplement the kettle-boiler operation since the kettle boiler relies on high-boiling heat-transfer coefficients in a "stagnant" pool of water. This approach would be very inefficient in either preheating the feedwater to the saturation temperature or superheating the resulting steam. Therefore, a complete steam generator unit or "train" consists of a preheater, a boiler, and a superheater joined in a series configuration.

The costs for the kettle-boiler and drum type steam generation equipment depend on heat transfer duty, log-mean temperature differences between the fluid steams, fluid pressures, and materials. Data was gathered on kettle boiler trains which reflect standard off-the-shelf designs. Cost estimates for the drum-type equipment were based on costing algorithms developed for basic heat transfer equipment since off-the-shelf designs do not readily exist over the range of parameters of interest. In all cases, the material of construction was assumed to be carbon steel except for those components in which temperatures in excess of 455°C (850°F) are experienced. In these cases, stainless steel was selected.

3.3.8.5 Feedwater Loop and Heat Rejection

This category includes the definition of the equipment required for the power conversion subsystem exclusive of the turbine-generator set, electrical power conditioning equipment and the steam (vapor) generator.

The approach adopted was to expand the definition of the feedwater loop components associated with a radial turbine installation in terms of a description, cost, and potential hardware supplier. In all cases, the components considered were of off-the-shelf design and available from many suppliers.

The principal elements of the feedwater loop receiving design attention include:

- Heat exchangers
- Pumps
- Condenser and ancillary equipment
- Water treatment
- Piping and valves
- Instrumentation

The heat exchangers include both tube and shell feedwater heaters and direct-contact deaerator equipment. Table 3.3-40 presents the principal design characteristics for the tube and shell heat exchangers for the system presented in the proposal. The surface areas were determined assuming an overall heat transfer coefficient of 10.2 kW/°C-m^2 (500 Btu/Hr-ft^2-°F) which would be representative of clean tube surfaces. The tube material selected for the low pressure heat exchanger was 90-10 Cu-Ni while carbon steel was selected for the high pressure heat exchangers. The Cu-Ni alloy was selected because of improved heat transfer over stainless steel. This choice prevents the use of ammonia for water pH control.

Table 3.3-40
FEEDWATER HEATERS SPECIFICATIONS

	LP Htr No. 1	HP Htr No. 3	HP Htr No. 4	HP Htr No. 5
Duty, KJ/S (Btu/h)	167.7 (571,700)	222.0 (756,800)	208.4 (710,400)	203.5 (693,600)
Feedwater Outlet Temp, °C (°F)	82.2 (180)	149 (300)	185 (365)	216 (424)
Extraction Temp, °C (°F)	85 (185)	177 (351)	253 (487)	318 (604)
Extraction Pressure, MPa (psia)	0.058 (8.4)	0.497 (72.1)	1.19 (173)	2.37 (344)
Design Shell Pressure, MPa (psia)	0.17 (25)	0.690 (100)	1.38 (200)	2.76 (400)
Heater Drain Temp, °C (°F)	48.3 (119)	116 (241)	154 (309)	191 (376)
Terminal Difference, °C (°F)	2.8 (5)	2.8 (5)	2.8 (5)	2.8 (5)
Drain Cooler Approach, °C (°F)	5.6 (10)	5.6 (10)	5.6 (10)	5.6 (10)
Design Tube Pressure, MPa (psia)	0.69 (100)	11 (1600)	11 (1600)	11 (1600)
Tube Area, m^2 (ft^2)	5.06 (54.5)	6.79 (73.1)	5.83 (62.7)	4.78 (51.4)
Dimensions, LX D, m (ft)	1.78 x 0.03 (5.83 x 1)	2.29 x 0.3 (7.5 x 1)	1.98 x 0.3 (6.5 x 1)	1.68 x 0.3 (5.5 x 1)
Mass, Dry, Kg (lbs)	191 (420)	227 (500)	195 (430)	177 (390)
Mass, Flooded, Kg (lbs)	323 (710)	395 (870)	341 (750)	300 (660)

Cost estimates for this equipment are contained in Appendix A. For other feedwater heater requirements, the costs are approximately proportional to heater duty assuming the same log-mean temperature difference. Changes in shell side pressure have little effect on cost for small sized units since the walls could be constructed of standard weight carbon steel pipe.

The requirements and characteristics of the deaerator are summarized in Table 3.3-41. This unit is designed to reduce dissolved oxygen to less than

Table 3.3-41

DEAERATOR REQUIREMENTS AND CHARACTERISTICS

Feedwater in	3660 Kg/hr (8052 lb/hr) 82.2°C (180°F) 344 J/g (148 Btu/lb)
HP Htr drains in	939 Kg/hr (2066 lb/hr) 116°C (241°F) 484 J/g (208 Btu/lb)
Steam in	186 Kg/hr (409 lb/hr) 110°C (230°F) 2627 J/g (1130 Btu/lb)
Feedwater out	4784 Kg/hr (10,515 lb/hr) 110°C (230°F) 460 J/g (198 Btu/lb)
Shell Operating Pressure	145 kPa (21 psia)
Shell Design Pressure	207 kPa (30 psia)
Storage Capacity	10 minutes of full flow
Weight, Dry	345 Kg (760 lbs)
Weight, Flooded	909 Kg (2000 lbs)
Dimensions	0.61m (2 ft) dia x 2.44m (8 ft) high
Type	Vertical, tray type
Material	Stainless steel trays, carbon steel shell

0.005 cc/l. It is sized to provide for a 10 minute full flow feedwater supply and is elevated to provide sufficient inlet head for the steam generator feed pump. Cost estimates are presented in Appendix A. As in the case of the feedwater heaters, the costs scale approximately proportional to duty for small sizes.

The proposed system contains four pumping stations within the power conversion subsystem. These are: (1) steam generator feed pump; (2) condensate pump; (3) circulation water pump for cooling tower heat rejection; and (4) condenser exhauster vacuum pump. The design requirements and characteristics of each of these pumps are defined in Table 3.3-42. In each case, a single full capacity pump is assumed. Costs for each pump are presented in Appendix A. The final choice concerning the need for pump redundancy was made on the basis of overall system costs and plant availability.

The condenser and cooling towers are designed to reject waste heat from the overall system. The cost of the equipment depends most strongly on the heat load and the condenser back pressure.

Table 3.3-42
MAJOR PUMP REQUIREMENTS AND CHARACTERISTICS

	Steam Generator Feed Pump	Condensate Pump	Circulating Water Pump	Condenser Exhauster Vacuum Pump
Number Required	1	1	1	1
Type	Triplex Plunger (Pos. Displacement)	Vertical Cent	Vertical Cent	2 Stage Liquid Ring
Manufacturers	Aldrich (I-R)	B-J	B-J	Nash
Model No.	1½ x 3HS3	100 VLT	GGL-1	AT404E
Capacity, Kg/hr (gpm)	4800 (22)	3670 (20)	341,000 (1500)	---- 3 scfm @ 1 in. Hga
Temp, °C (°F)	110 (230)	42.8 (110)	29 (85)	---- ----
Specific Gravity	0.95	0.989	0.993	----
Approx TDH, m (ft)	1219 (4000)	70 (200)	10.7 (35)	---- ----
Pump Speed, rpm	530	1750	1150	1150
Driver Speed, rpm	1750 (gear drive)	1750	1150	1150
Driver Rating, kW (bhp)	22.4 (30)	2.2 (3)	15 (20)	18.6 (25)
Efficiency, %	88	67	82	----
Est Operating Req kW (bhp)	17.9 (24)	1.1 (1.5)	11.9 (16)	14.2 (19)

Three designs were developed in order to determine cost sensitivity to heat load and condenser back pressure. The requirements, design characteristics, and cost estimates are presented in Table 3.3-43. The costs for the condenser are essentially proportional to tube surface area (and circulating water flow rate). The costs for the cooling tower equipment reflects the heat load and commercially available size increments. It is seen in comparing the "baseline" with "Alternate #1," identical hardware is required. The fan power for the alternate is significantly lower which directly influences plant parasitic loads.

Table 3.3-43

ALTERNATE CONDENSER AND COOLING TOWER DESIGN AND COST DATA

Condenser	Base Case	Alternate #1	Alternate #2
Condenser Duty	2.0 MWth @ .5 in. Hga	2.0 MWth @ 5 in. Hga	3.0 MWth @ 5 in. Hga
Condenser Type	2 Pass	2 Pass	2 Pass
Cir Water Flow, gpm	1532	745	1117
Tube Type	18 BWG Admiralty	18 BWG Admiralty	18 BWG Admiralty
Tube Length, ft	20	20	20
Tube O.D., inches	1	1	1
Total Tube Area, sq ft	290	142	212
Condenser Cost, $	4000	2000	3000
Cooling Tower			
Approach, °F	11	11	11
Rise, °F	8.91	18.33	18.33
Dimensions, LXWXH, ft	12 x 18 x 23	12 x 18 x 23	18 x 18 x 23
Pump Head, ft	21	21	21
Fan Driver Rating, hp	20	15	20
Cost, $	25,000	25,000	35,000

The water treatment equipment includes:

- One single train makeup demineralizer
- One duplex mixed bed condensate polisher
- One boiler chemical feed system
- One cooling tower chemical feed system
- One cooling tower control system
- One boiler water monitoring panel

Makeup Demineralizer

The makeup demineralizer system will consist of a strongly acidic cation exchange vessel followed by a strongly basic anion exchange vessel followed by a mixed bed exchange vessel. A single train demineralizer feeding a storage tank will provide adequate boiler makeup water since the steam generator makeup requirements are quite modest--between 0.38 and 2.65 m^3 (100 and 700 gallons) per day--and the smallest automatically regenerated package demineralizer of most manufacturers is much larger than required. A condensate storage tank will be provided to store several days makeup requirements so that the demineralizer can be regenerated during operating periods and to allow for equipment down time for repairs.

Hydrochloric acid will be used to regenerate the cation vessel and the cation exchange resin in the mixed bed vessel because it simplifies the design and operation of small package demineralizers. When these units are used, the above considerations usually outweigh the higher cost of hydrochloric acid, as compared to sulfuric acid.

Condensate Polisher

The condensate polisher will be a nonregenerative powdered resin unit. The piping will be arranged to provide a bypass around the polisher.

Steam Generator Chemical Feed Systems

Chemical feed systems will be provided to feed chemicals to the boiler or preboiler system in order to control the chemistry in the boiler and condensate system. The chemicals to be fed would normally be selected by the plant operations personnel with the advice of a water treatment consultant. Chemical feed systems have been selected on the assumption that three chemical activities will be fed--hydrazine, an amine, and a boiler chemical.

Cooling Tower Chemical Feed System

Chemical feed systems will be provided to feed chemicals to the circulating water system to control pH and the tendency of the water to be corrosive and/or scale forming.

Two systems will be provided, one to feed sulfuric acid and one to feed a scale inhibitor. The acid feed system will be controlled by the pH of the circulating water.

Cooling Tower Control System

A control system will be provided to monitor the conductivity and pH of the circulating water. The control system will blow down the cooling tower in order to maintain a preset conductivity. pH will also be monitored and the feed rate of sulfuric acid will be controlled to maintain a preset pH.

Steam Generator Water Monitoring Panel

A control panel complete with instruments, sample cocks, recorders, and necessary accessories would be provided to monitor the following:

- Demineralizer
  - (Conductivity will be monitored at the demineralizer equipment)
- Condensate Pump Discharge
  - pH
  - Cation conductivity
  - Sodium
- Condensate Polisher Outlet
  - pH
  - Specific Conductivity
  - Sodium (Analyzer will be shared with condensate pump discharge sample)

Deaerator

Dissolved oxygen

Steam Generator

Silica

pH

Conductivity

Steam

Sodium

A temperature control system is not included. Pressure reducing devices and sample coolers are provided where required. This represents a minimal "no frills" system considering the design temperatures and pressures involved in the water/steam cycle.

Cost estimates for each of the water treatment elements defined above are presented in Appendix A. These costs reflect the requirements as they have been developed for the water/steam cycle.

Instrumentation required for the power conversion subsystem includes temperature, pressure, flowrate, and level sensors, as well as speed, voltage, and current sensors. It is assumed that minimum cost sensors will be used which are consistent with the overall accuracy and sensor reliability objectives. Temperatures will be monitored with commercial thermocouples while the other sensing functions will be carried out by commercial equipment which is available from a variety of suppliers. In developing final subsystem design detail, care was exercised to minimize the number of instrumentation locations specified since pressure and flow sensors can add significantly to the cost if used on an indiscriminate basis. The use of low cost thermocouples, however, is not nearly as sensitive.

3.3.8.6 Power Conversion Subsystem Cost Summary

A summary of preliminary costs of the power conversion subsystems is given in Table 3.3-44. These costs include the steam/vapor generator, all piping and major power generation components, structure to house equipment, cooling

Table 3.3-44

POWER CONVERSION SUBSYSTEM COST SUMMARY

Axial Turbine - Carbon Steel	$960K
Axial Turbine - Chrome-Moly	$1080K
Radial Turbine - 482°C (900°F) Inlet	$890K
Radial Turbine - 371°C (700°F) Inlet	$880K
Organic Turbine - Subcritical or Supercritical	$800K
Gas Turbine -	$900-1000K

towers, electrical switchgear and controls and installation. Costs of organic Rankine cycle systems are based on manufacturers estimates of downstream "Nth" unit packages assembled at the factory on skids and shipped ready for immediate use. The costs for the axial and radial steam turbine systems reflect detailed cost estimates of components and assembly in the field. A cost reduction in these steam turbine subsystems could be expected if a skid-mounted, mass production approach was adopted.

3.3.9 Plant Control Concept

The Plant Control Subsystem (PCS) for the Small Power System Experiment provides the facilities for the control and monitoring of the operating plant. These facilities, through automatic, semi-automatic or manual operating methods, perform the following functions:

- Sensing of all subsystem operations to assess safe and proper operation of the plant.
- Display and/or recording of the sensed parameters in a form pertinent to the evaluation of plant performance, operation and safety.
- Control and command of the subsystems components (i.e. valves, motors, blowers, etc.) to establish and/or maintain plant stability through all phases of operation.

To accomplish these functions the objectives of high reliability, cost effectiveness and simplicity must be recognized in the Plant Control Subsystem implementation. These objectives take on an added importance for small power plant applications since the efficiency of plant operation can have a significant impact on the cost of energy produced. These objectives are further reinforced considering that attractive uses of this type of power generation plant are in rural and remote localities where the plant operations will not be consistent with general utility practices.

The effort for the selection of the plant control concept included the following scope:

- Establish the operating and interface requirements for the Plant Control Subsystem.
- Develop several potential Plant Control System concepts that fit the requirements.
- Perform a plant control concept analyses of the potential candidates.
- Select a preferred plant control concept from the technical, and cost analyses.

3.3.9.1 Plant Control System Requirements

The general top level control system design requirements philosophy included the following basic points:

- PCS shall provide sensing, detection, monitor and control of all system and subsystem parameters necessary to ensure safe and proper operation of the Plant.
- Data display and/or recording shall be provided for those parameters pertinent to evaluation of Plant performance, operation and safety.
- Plant operation shall be essentially automatic with operator override capability provided.
- Provide independent subsystem control for manual operation should the operator desire.
- Provide for single console control with easily read displays during both automatic and manual operation.
- Provide standard, proven, off-the-shelf control practices and simple, well-defined interfaces between PCS and subsystem controls.
- Eliminate single point failure effects where cost effective.

- Provide supervisory subsystem control; i.e., set points, transfer functions, constants, biases, timing, sequencing.

In a single phase system of the type under investigation, the steam conditions are essentially decoupled from the solar conditions by the buffering action of the storage. This creates a reasonably benign set of control requirements in comparison to systems in which the receiver is directly tied to the turbine or steam generating equipment.

There are two basic PCS duties that are needed for operation: 1) management of energy input to the system, and 2) management of load demand. Energy input to the system, for a given solar condition, can only be controlled by controlling the number of heliostats that are active. PCS must accomplish this by communicating with the Collector Subsystem control system. Load demand can be controlled most effectively by giving the network controller the visibility of available plant operating time, established by PCS, at the energy level called for.

Integration of the independent subsystems for the small power plant imposes the following PCS operational requirements:

- Power Conversion Subsystem
  - Turbine startup and shutdown in accordance with an optimum life algorithm, automatically and/or operator guidance menu.
  - Coordinate grid demand signals with turbine throttle and steam generator
  - Monitor, display and alarm appropriate data.
- Receiver/Storage
  - Startup and shutdown sequencing of valves, set points, motors, etc.
  - Emergency sensing and shutdown coordination with the collectors.
  - Controller transfer function and set point adjustment as required.
  - Monitor, display and alarm appropriate data.
- Collector
  - Startup and shutdown sequencing.
  - Tracking management for modes, time, beam safety and alignment.

- Power modulation by partial field tracking.
- Emergency sensing, slew and stow.
- Monitor, display and alarm appropriate data.

The PCS requirements also cover the acquisition, manipulation and recording of plant data used to evaluate plant performance. These functions are to be accomplished in a timely manner, correlated with events of the plant operation, retained and logged in an interpretable form. Centralizing these duties for all subsystems within the PCS provides a single point within the plant for evaluating the plant operation performance.

The general plant control requirements provide the basis from which the plant operating requirements and subsystem interface requirements are defined. The Plant Control Operating Requirements are separated into two categories defining: 1) plant operations to develop PCS operating modes and operations integrity and 2) plant support that describes the operator interface and facility support requirements.

The operating modes requirements describe the power generated, startup, shutdown, emergency, maintenance and safing modes that PCS must be capable of controlling and coordinating. Complementing the operating mode requirements are the operations integrity requirements stating the reliability, safety and environments within which the plant must operate.

The Plant Operating Support Requirements define the operator interface and the facility that the PCS operates within. Operator interface requirements include the monitor, command, and plant evaluation aspects of the Plant Control System. Facility support requirements define the power, environments, safety/hazards and maintenance elements associated with PCS at the plant site.

The second category of Plant Control Requirements defines the interface requirements between PCS and the subsystems of the Plant. These requirements describe the interfaces and the characteristics of the interfaces to PCS. Both physical and functional requirements are defined. Physical requirements include the controls, measurements, alarms, safing and installation references to PCS. Functional requirements relate to interface compatibility and environments between PCS, the operator and the other subsystems.

3.3.9.2 Plant Control Subsystem Concepts

The design of the Plant Control Subsystem for the Small Power System must address the objectives of high reliability, cost effectiveness and simplicity. To achieve these objectives the design must incorporate proven concepts; low cost hardware, software and interfaces; and a simple plant operational approach.

Several conventional control system approaches were analyzed for the PCS application. These approaches were:

- Analog control using a centralized control center.
- Direct digital control using a centralized control center.
- Distributed digital control using a central control center.

The analyses and evaluations of these concepts, in addition to providing comparative technical and cost data, were used in the optimization of control system approach of the commercial plants for the 3.5 year program, the 4.5 year program and the 6.5 year program.

Recent literature searches and discussions with the utilities and petroleum chemical process industries indicate a shift from the total centralized analog control systems philosophies to a variety of forms of distributed and direct digital control methods. The distributed digital control and direct digital control approaches appear to be the most promising Plant Control System concepts for solar power plants. However, the analog approach for small plants where simple control techniques can be employed cannot be overlooked.

Looking ahead in the 1980 time-frame when the Small Power System Experiment would be consumated into a working plant, several opportunities will be available to the power plant control system designers that have a distinct advantage over present power plant control hardware techniques. These advantages include: 1) lower cost electronic products of all kinds, 2) high speed, very reliable information transmission techniques, 3) low power consuming electronic devices, and 4) high density electronic packaging. These opportunities are becoming prominent in all industries today and will see significant improvements and developments in the years ahead.

The serial digital data transmission bus has been growing in popularity in the process industry because of: 1) the reduced wiring costs, 2) high immunity to external noise sources, and 3) the increased use of digital computers for process monitor and control applications. Fiber optic techniques are gradually replacing the coaxial and twisted pair serial data transmission busses. This technique retains the attributes of the conventional serial digital information transmission bus but has the capacity to handle transmission speeds approaching the speed of light. With the extremely wide frequency bandwidth of fiber optics (over 200 megaHertz) many individual signal paths can be accommodated on a single strand.

All of these devices and techniques mentioned heretofore utilize solid state integrated circuit technologies almost exclusively. This technology continues to show mean time between failure rates for components of greater than fifty thousand hours (approximately 5.5 years). Furthermore, the low power requirements to operate these devices coupled with the materials and packaging techniques used have extended the environmental limits of temperature, humidity and shock within which these components will operate. Consequently, sequence programmers, microprocessors, and digital converters do not have to be placed in stringently controlled environments. These devices will operate in many field environments.

All of these considerations provide effective design tools for the Plant Control System concept that will minimize the operator role in the control of the commercial plant. A significant cost benefit can be obtained for a small solar power plant if there is not a need for a full time operator and when operator intervention is required, that role is simple, providing a minimum of decision alternates. Considerable design emphasis will be placed to design automatic and semi-automatic PCS capabilities for the commercial plant, whereby on-site maintenance personnel can be trained to provide intermittent operator functions.

Generalized design concepts for the Plant Control Subsystem using analog, distributed digital and direct digital approaches are presented in Figures 3.3-75 through 3.3-77. The analog control concept and the direct digital control concept illustrate central control methods for PCS. All of the logic to control the elements of the Plant is located in the control room. The distributed digital control method divides the plant control logic between the central control room and the field controller. From these sketches, it is easy to see that there are substantial savings in field wiring using either the directed digital or distributed digital approaches. Either of these two methods utilize a single coaxial cable or twisted pair wire to transmit signals to and from the central control equipment and the field controllers. However, fancy communication schemes and system security methods (error detections and corrections to information transfers and redundancies to prevent line outages that bring the system down) have to be incorporated in these concepts to obtain system reliability

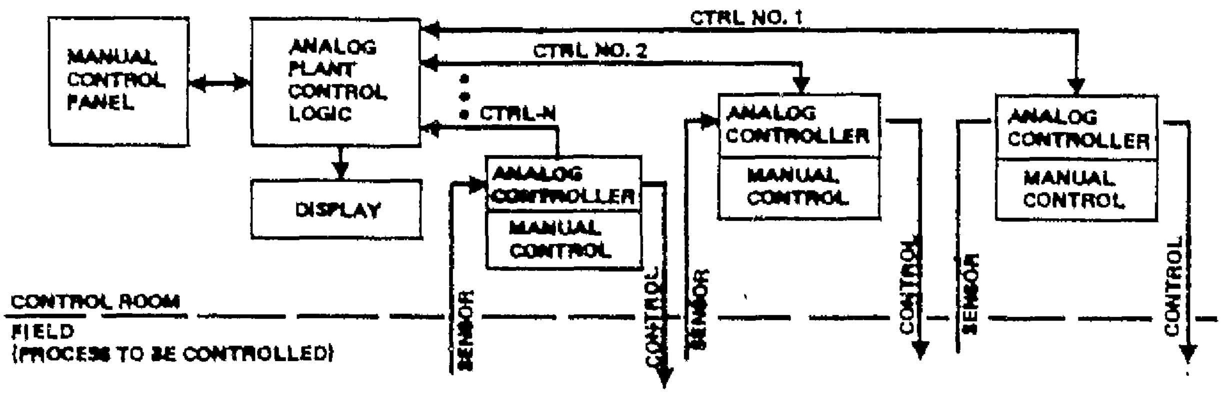


Figure 3.3-75. Generalized Analog Control System — Block Diagram

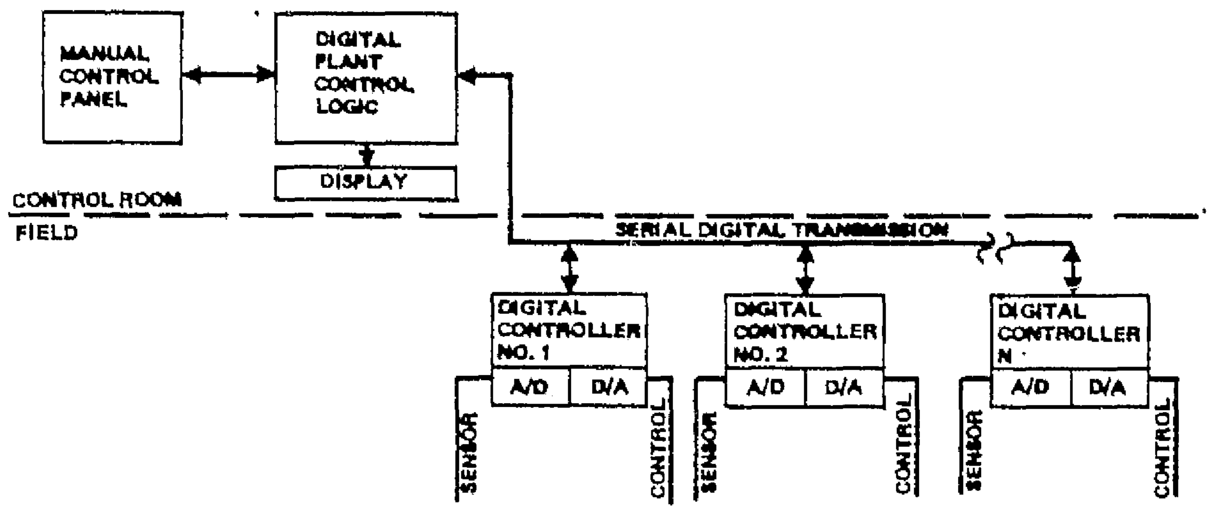


Figure 3.3-76. Generalized Distributed Digital Control System — Block Diagram

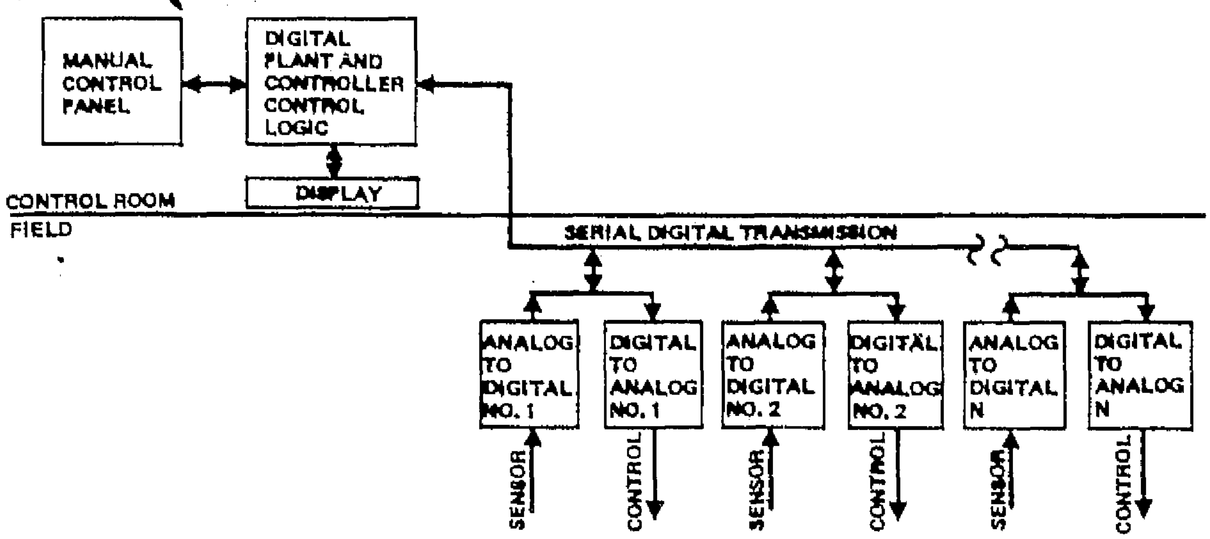


Figure 3.3-77. Generalized Direct Digital Control System — Block Diagram

3.4 SELECTION OF PREFERRED SYSTEM CONCEPTS

The primary objective of this evaluation was to select the preferred design for Engineering Experiment Number 1 (EE-1) for each of the three program start-up times (3.5, 4.5, and 6.5 years). The basic task approach was to use the selection criteria developed in Section 3.1 to compare the candidate concepts selected in Section 3.2 and optimized in Section 3.3.

3.4.1 System Candidates

Final system configurations were synthesized from the initial candidate concepts for each of the 3.5, 4.5, and 6.5 year startup programs. These system candidates are summarized in Table 3.4-1. Boxes with X's represent concepts eliminated in the screening process described in Section 3.2.

As indicated on Table 3.4-1, seven basic system types were synthesized for each start-up period according to receiver coolant and power conversion working fluid. These systems include: (1) HTS/steam, (2) Syltherm/steam, (3) Syltherm/organic, (4) Caloria/steam, (5) Caloria/organic, (6) Saturated steam, and (7) Gas/air Brayton cycle. Operating temperature ranges were reduced for the shorter duration programs to accommodate the relatively short development periods available and to minimize development/startup risks and costs. Similarly, thermal storage concepts were selected to be relatively simple for the shorter programs as compared to the longer programs. Likewise, more standard/existing type turbines (i.e., axial) were selected for the shorter programs as compared to the more advanced turbine types (i.e., radial) for the longer programs. As shown on the table, alternative subsystem approaches have also been identified for some of the system candidates.

For some candidates, the short (3.5 year) program is not feasible due to the limited (8 month) Phase II development period specified. For this reason, no candidates are proposed for the Syltherm/organic, Caloria/steam, Saturated steam and Gas/air concepts for the 3.5 year program.

After the identification of the system candidates shown on Table 3.4-1, a process of elimination was then initiated resulting in the selection of one preferred concept for each of the three startup programs. This process is described in the following section.

Table 3.4-1

SYSTEM CANDIDATES

	1	2	3	4	5	6	7
STARTUP PROGRAMS	HTS STEAM	SYLTHERM		CALORIA		SATUR. STEAM	GAS/AIR
		STEAM	ORC	STEAM	ORC		
3.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	430-510°C 2 TANK AXIAL	400-454°C TRICKLE AXIAL			300-316°C 2T/DMT ORGANIC (SUBCR.)		
4.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	510°C 2T/DMT AXIAL/RAD	450-480°C TRICKLE/DMT AXIAL/RAD	450-480°C TRICKLE/DMT ORGANIC (SUPERCR.)	316°C 2T/DMT RADIAL	316°C 2T/DMT ORGANIC (SUBCR.)	500-600°C PRESS. WATER RADIAL (W/REHEAT)	680-820°C BATTERY GAS
6.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	510-580°C DMT RADIAL	450-480°C TRICKLE/DMT RADIAL	450-480°C TRICKLE/DMT ORGANIC (SUPERCR.)	316°C DMT RADIAL	316°C DMT ORGANIC (SUBCR.)	500-600°C PRESS. WATER RADIAL (W/REHEAT)	680-820°C BATTERY GAS

2T ∿ TWO TANK STORAGE
DMT ∿ DUAL MEDIA THERMOCLINE STORAGE

3.4.2 Selection Criteria and Methodology

In order to make meaningful comparisons between the various power systems proposed for the Small Power System Experiment, JPL established four system selection criteria to be used in the assessment of each concept. This section addresses the method in which the selection criteria were applied to the system candidates for each of the three programs.

3.4.2.1 Overall Program Approach

From the overall program standpoint, MDAC has taken the approach that the experiments must lead towards the best commercial system that is expected to be operational in the mid to late 1980's. Thus the experimental unit is not necessarily the precise design for the commercial unit envisioned, but it must establish the fundamental technology and operational approach to be used in the ultimate commercial power plant. In other words, system integrity must be maintained between EE-1 and the commercial system. For instance, the basic heliostat/tower/receiver approach must be maintained, however, design variations or improvements for individual heliostats, tower structure or receivers are considered to be permissible. The receiver fluid should not be varied -- the fluid selected for the commercial plant should also be used for EE-1. The overall design approach for the receiver fluid transport, storage, and steam generator loop should be maintained. In the power conversion subsystem, the general type of prime mover (i.e., gas, steam, organic) should remain constant, however, turbine design may vary.

Thus, the overall MDAC approach to system synthesis was to determine the best commercial plant configuration for each candidate system type and then identify the maximum simulation of this unit attainable within the three program durations allowed for EE-1 consistent with specific system selection criteria. For each of these three program durations, subsystems and components were selected that could satisfy the development schedule requirement with a high degree of confidence, yet permit improvements leading towards an ultimate commercial application without requiring significant changes in basic design approach, operations, or technology development.

The specified program durations for the experiments and their interface with the commercial unit were summarized on Figure 3.4-1. As shown, the current Phase I conceptual design activities are scheduled for 10 months for all of the three startup programs. Phase II system preliminary design and subsystem component development testing periods vary (8, 18, or 42 months) for the three programs. Phase III final experimental system design, fabrication, installation and operation are all approximately 3 years in duration with on-line capability established 22-24 months after go-ahead. One year test operation of the experimental plant is scheduled.

It has been assumed that the commercial version of these experiments should be available for commercial sales by at least mid-1988 (10 years after Phase I go-ahead). To accomplish these sales, it will be necessary to demonstrate the capabilities and operational features of the actual commercial system over a

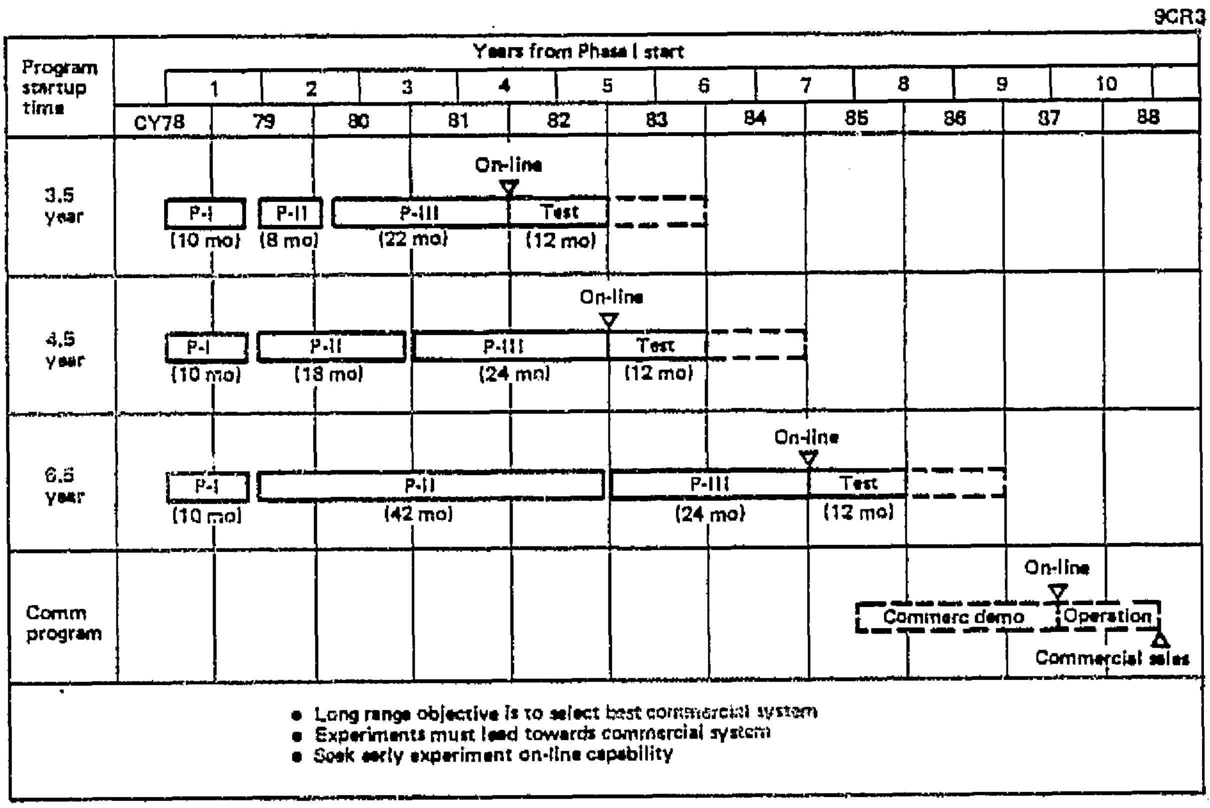


Figure 3.4-1. Overall Program Approach to Concept Selection

reasonable time period. A one-year operational period has been selected for this demonstration period. Thus, the commercial demonstration unit should be "on line" in mid-1987. To achieve this date, the design, fabrication and installation of the demonstration unit must be initiated by mid-1985. The 3.5- and 4.5-year programs allow this sequential development. However, the 6.5 year program does not have this flexibility for iteration between EE No. 1 and the commercial plant. In fact, experimental testing for the 6.5 year program does not begin until early 1985. Therefore, the 6.5 year version of EE No. 1 must be very similar to the final commercial version.

3.4.2.2 Selection Criteria and Application

In Section 3.1, the system selection criteria established by JPL were further developed and used to support subsystem selection and optimization (Sections 3.2 and 3.3). These criteria were then applied to select the preferred system candidates. The system selection criteria are: (1) high operational reliability, (2) minimum risk of failure, (3) commercialization potential, and (4) low program costs. They are summarized in their order of importance on Table 3.4-2.

The application of these criteria to the experimental and commercial systems is summarized on Table 3.4-3. Some of the selection criteria apply to only the commercial unit, whereas other criteria apply only to the experimental units. For instance, the requirement that the commercial plant operate with a high reliability during its life time applies to the commercial version of the plant. In this case, the experimental units must involve subsystems and operations that can lead towards establishing this commercial reliability capability. On the other hand, the requirement that the experimental plant start-up satisfactorily and operate with high reliability applies only to the experimental units, but is reflected in the commercial version.

As indicated on the table, the selection of the commercial plant was to be based primarily on reliability/availability and commercialization which relates to costs, performance, flexibility and institutional interface aspect The selection of the specific design concepts for EE No. 1 was to be based primarily on startup/operational reliability, program risk, and program cost.

Table 3.4-2

SELECTION CRITERIA (IN ORDER OF IMPORTANCE)

1) HIGH OPERATIONAL RELIABILITY - SELECTED SYSTEM CONCEPTS SHOULD LEAD TO:

- A COMMERCIAL PLANT THAT OPERATES WITH A HIGH RELIABILITY DURING ITS LIFETIME (TYPICALLY 30 YEARS)

- AN EXPERIMENTAL PLANT WHICH WILL START UP SATISFACTORILY AND OPERATE RELIABLY FOR AT LEAST TWO YEARS AFTER STARTUP WITH MINIMUM FORCED OUTAGES ATTRIBUTABLE TO DESIGN DEFICIENCIES AND HARDWARE FAILURES

(ENHANCEMENT OF RELIABILITY THROUGH MODULARITY/REDUNDANCY SHOULD BE CONSIDERED)

2) MINIMUM RISK OF FAILURE - SELECTED CONCEPTS SHOULD MINIMIZE DEVELOPMENT RISK AND THEREBY PROVIDE HIGH CONFIDENCE THAT SUBSYSTEM DEVELOPMENT CAN BE ACHIEVED WITHIN PHASE II TIMES AND THAT THE EXPERIMENT CAN BE BROUGHT ON-LINE AT THE SPECIFIED STARTUP TIMES

3) COMMERCIALIZATION POTENTIAL - SELECTED CONCEPTS SHOULD USE OR CONTRIBUTE DIRECTLY TO THE EVENTUAL SYSTEMS THAT ARE LIKELY TO ACHIEVE COMMERCIAL SUCCESS IN THE LATE 1980'S

- COSTS/PERFORMANCE

- FLEXIBILITY (MODULARITY SHOULD BE ONE OF PRIMARY CONSIDERATIONS)

- INSTITUTIONAL INTERFACE ASPECTS

4) LOW PROGRAM COSTS - CONCEPTS SHOULD BE SELECTED TO MINIMIZE THE ESTIMATED DEVELOPMENT AND CAPITAL COSTS OF PHASE II AND PHASE III

For these reasons, heavy emphasis was placed on the selection of subsystems and components compatible with these criteria as well as the selection of the best overall system approach. The specific methodology for system and subsystem selection is presented in the next section.

3.4.2.3 Methodology of Evaluation

An evaluation methodology was developed for the selection of the three preferred system candidates that progressively tested or screened each candidate with respect to the selection criteria described above. These steps are summarized below:

1. Form concept candidates into seven families defined by collection fluid and prime mover

Table 3.4-3

APPLICATION OF SELECTION CRITERIA

SELECTION CRITERIA	ENGINEERING EXPERIMENT NO. 1			COMMERCIAL SYSTEM
	3.5 YR	4.5 YR	6.5 YR	
• RELIABILITY/AVAILABILITY				
- COMMERCIAL SYSTEM	➡	➡	➡	✓
- EXPERIMENT	✓	✓	✓	*
• PROGRAM RISK	✓	✓	✓	*
• COMMERCIALIZATION	➡	➡	✓	✓
• PROGRAM COSTS	✓	✓	✓	*

✓ ∿ APPLICABLE CRITERIA

➡ ∿ MUST LEAD TOWARDS

* ∿ REFLECTED BY EE-1

2. Apply program risk as an absolute screening criterion within each of the three program durations
   - Screen subsystems/components within each family according to available development time
   - Screen concepts themselves according to program risk criterion
3. Evaluate system reliability/availability
   - Provide quantitative estimate of commercial system availability
   - Rate EE No. 1 candidates for each program duration according to departures from existing equipment and extent of qualification testing
4. Evaluate candidate systems' commercialization potential
   - Synthesize preferred commercial system configuration for each candidate
   - Calculate relative energy cost for each candidate
     - Capital costs
     - Relative maintenance and replenishment costs
   - Rate according to flexibility and institutional aspects
5. Estimate relative program costs for each candidate system for each program duration
6. Overall evaluation and selection of the preferred system for each program

The initial step was to form the candidate concepts into seven families defined by the collection fluid and prime mover. These seven families were listed on Table 3.4-1.

The second step was to apply program risk as an absolute screnning criteria within each of the three program durations. Each subsystem or component within each family was screened according to development required and available development time. Then each system concept was screened according to the program risk criterion. Results of this screening for program risk are reviewed in Section 3.4.3.1.

The third step was to evaluate system reliability/availability by estimating commercial system availability and rating each candidate system according to departures from existing equipment and the extent of qualification testing required. Results of this assessment are reviewed in Section 3.4.3.2.

The fourth step was to evaluate the remaining candidate systems with respect to their commercialization potential. Commercial system configurations were synthesized for each candidate system. Capital and energy costs were then estimated for each commercial candidate and ratings established relative to system flexibility and institutional aspects. Results of this assessment are reviewed in Section 3.4.3.3.

The fifth step was to estimate Phase II and III program cost differences for the remaining candidate systems. These relative cost estimates were preliminary and exclude all common element costs. Results of this assessment are reviewed in Section 3.4.3.4.

The last step was to summarize all of the above evaluations for the candidate systems surviving the program risk screening, compare the systems, and select the final preferred system candidates. The results of the comparison and selection are covered under Section 3.4.4 and a general identification of the preferred commercial system is given in Section 3.4.5

3.4.3 System Evaluation

3.4.3.1 Program Risk

The program risk selection criteria was applied as an absolute screen of alternative concepts. That is, candidates that did not meet the risk criterion were eliminated from further consideration. Consequently, this criterion was applied first in system evaluation. To accomplish this risk screening, a preliminary assessment was made of the development status of each major subsystem or component as appropriate. From this assessment, it was determined if the equipment was existing or state-of-the-art. For those subsystems/components requiring development, the primary means of technical verification was identified--either by (1) correlation with or testing in other on-going programs, (2) by Phase II analyses, or (3) by Phase II development testing. If Phase II development testing was required, an estimate of the time required for this development was made. In some cases, this time was influenced by the operating temperature range to be used. This assessment is summarized on Table 3.4-4.

Table 3.4-4

SUBSYSTEMS REQUIRING DEVELOPMENT

SUBSYSTEM COMPONENT/FEATURE	EXISTING EQUIPMENT OR STATE-OF-THE-ART	PRIMARY MEANS OF TECHNICAL VERIFICATION				COMMENTS
		OTHER ON-GOING PROGRAMS	PHASE II			
			ANALYSIS	DEVELOPMENT/ TEST	EST. DURATION (MO.)	
COLLECTOR SUBSYSTEM						
• HELIOSTATS		X				BARSTOW, PROTOTYPE HELIOSTAT DEVEL. PROGRAM
• TOWER	X					
• RECEIVER						
HTS (DRAWSALT)				X	30	T > 510°C
HITEC				X	12-18	T ≤ 510°C
HITEC	X		X	X	0-8	T = 450°C
SYLTHERM			X	X	12-18	T = 400°C
CALORIA HT43	X		X	X	12-18	T = 300°C
WATER/STEAM	X		X	X	0-8	T = 252°C
AIR				X	36	T = 830°C
ENERGY TRANSPORT SUBSYSTEM						
LOOP COMPONENTS						
DRAW SALT				X	18	
HITEC/OTHER FLUIDS	X		X			
ENERGY STORAGE SUBSYSTEM						
• TWO-TANK (ALL FLUIDS)	X		X			
• DUAL MEDIA/TRICKLE CHG.						
HTS				X	8-18	MATERIAL
SYLTHERM				X	18+	COMPATIBILITY
CALORIA HT43		X				BARSTOW
• PRESSURIZED WATER	X		X			
• BATTERY						
LEAD-ACID	X					
ADVANCED		X			36-60	Li/FeS
POWER CONVERSION SUBSYSTEM						
• AXIAL STEAM TURBINE	X					
RADIAL STEAM TURBINE				X	18	
• ORGANIC-SUBCRITICAL	X					2-600 KWe UNITS
ORGANIC-SUPERCRITICAL				X	18	
AIR-OPEN LOOP		X			18	EPRI PROGRAM
AIR-CLOSED LOOP				X	36	
BALANCE-OF-PLANT	X					

The subsystems/components were then screened within each family according to available development time, and each system concept was screened according to program risk criterion. A subsystem or component was accepted for each program duration only if the estimated development duration was approximately two-third of allowable Phase II duration times and the probability of successful development was high (over 95 percent). Subsystems or components that were rejected by application of this risk criterion are summarized on Table 3.4-5, together with the corresponding Phase II program durations and reasons for rejection.

The use of HTS above 510°C (950°F) for the 3.5 and 4.5 year program was rejected because of the development time required for receiver and loop component design for this temperature range. Likewise, the use of HTS operating in the 450°C-510°C range for the 3.5-year program was rejected because of receiver development requirements exceeding allowable development time.

Review of fluid testing results limited the maximum temperature of Syltherm to 400°C rather than the 450°C-480°C shown in Table 3.4-1. Further, the use of Syltherm at 400°C was rejected for the 3.5 and 4.5-year programs and considered marginal for the 6.5 year program due to fluid degradation and compatibility with solids. MDAC's design and analyses for Syltherm concepts have been based on manufacturer's data which indicated the degradation of Syltherm-800 alone is not severe (i.e., approximately 4 percent weight loss at 1,000 hours at 371°C [700°F]). However, preliminary data from other sources have indicated that the weight loss of the fluid is much higher at 400°C (i.e., approximately 15 percent weight loss at 1,000 hours), and that the degradation with solids such as Taconite is clearly unacceptable (i.e., approximately 37 percent weight loss at 1,000 hours at 400°C). Use of the fluid at lower temperatures (downrating) was not considered to be an economically viable approach. Therefore, the probability of a successful program is low, and the program risk is high. More extensive compatibility testing with solids is needed before the use of Syltherm at 400°C is considered feasible.

The use of Caloria above 302°C (576°F) was rejected for the 3.5-year program because extensive qualification at this temperature was required which the

Table 3.4-5

SUBSYSTEMS/COMPONENTS REJECTED BY RISK OF FAILURE CRITERION

SUBSYSTEM/COMPONENT	PROGRAM DURATIONS (YRS)	REASONS
HTS ABOVE 510°C	3-1/2, 4-1/2	• DEVELOPMENT DURATION FOR LOOP COMPONENTS AND RECEIVER
HTS FROM 450°C-510°C	3-1/2	• RECEIVER DEVELOPMENT
SYLTHERM (400°C)	3-1/2, 4-1/2,	• FLUID DEGRADATION
	(6-1/2?)	• COMPATIBILITY WITH SOLIDS
		• PROBABILITY OF SUCCESSFUL PROGRAM (?)
		• FURTHER DOWNRATING NOT VIABLE COMMERCIALLY
CALORIA ABOVE 302°C	3-1/2	• EXTENSIVE QUALIFICATION AT THIS TEMPERATURE
AIR RECEIVER	3-1/2, 4-1/2	• DEVELOPMENT DURATION
HTS DUAL MEDIA THERMOCLINE STORAGE	3-1/2, 4-1/2	• CONSTRAINTS ON LOOP COMPONENTS
ADVANCED BATTERIES	ALL	• DEVELOPMENT TIME
RADIAL STEAM TURBINE	4-1/2	• DEVELOPMENT TIME
ORGANIC SUPERCRITICAL TURBINE	4-1/2	• DEVELOPMENT TIME
		• NO BACKUP
OPEN LOOP BRAYTON	4-1/2	• TURBINE MODIFICATION TIME
CLOSED LOOP BRAYTON	ALL	• DEVELOPMENT TIME FOR FULL LOOP

program duration does not permit. Likewise, the use of an air receiver for the 3.5 and 4.5 year programs was rejected because of development duration time requirements.

The use of a HTS dual-media thermocline thermal storage system was also rejected for the 3.5 year program because of design constraints imposed on loop components and the corresponding times required for subsystem development and testing. The development requirements for the HTS dual-media storage system was considered marginal for the 4.5-year program, and therefore was also rejected.

The lack of suitable development time was also the reason for rejection of several other components. For these reasons, the use of a radial steam turbine or an organic supercritical turbine were rejected for the 4.5-year program. The use of advanced batteries and the closed-loop Brayton approach was rejected for all three program durations. The open loop Brayton concept was rejected for the 4.5-year program because of the excessive time required to modify and test existing turbines.

The system candidates that remained following this risk screening are summarized on Table 3.4-6. As can be noted, five system combinations were rejected completely and two were classified as marginal. Also shown on the table are the specific operating temperature ranges, storage type and turbine types that are compatible with the various startup programs. For instance, for the HTS/steam concept the 3.5-year program must be limited to 450°C (842°F), a two-tank thermal storage system, and an existing axial steam turbine. The corresponding 4.5-year program can operate over a higher temperature range (450-510°C) with a two-tank storage system. The 6.5-year program can operate in the 510-580°C range using a dual-media thermocline tank and a radial turbine. Remaining combinations for other candidate systems are noted on the table.

Table 3.4-6

SYSTEM CANDIDATES FOLLOWING SCREENING FOR RISK

	1	2	3	4	5	6	7
STARTUP PROGRAMS	HTS STEAM	SYLTHERM		CALORIA		SATUR. STEAM	GAS/AIR
		STEAM	ORC	STEAM	ORC		
3.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	450°C TWO-TANK AXIAL	(REJECTED)			302°C TWO-TANK/ DMT/ORGANIC (SUBCR.)		
4.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	450-510°C TWO-TANK AXIAL	(REJECTED)	(REJECTED)	316°C DMT AXIAL	316°C DMT ORGANIC (SUBCR.)	(REJECTED)	(REJECTED)
6.5 YR PROGRAM • TEMP. LIMIT • STORAGE • TURBINE	510-580°C DMT RADIAL	(MARGINAL) 400°C TRICKLE/DMT RADIAL	(MARGINAL) 400°C TRICKLE/DMT ORGANIC	316°C DMT RADIAL	316°C DMT (SINGLE UNIT)	300°C PRESS. WATER RADIAL (W/REHEAT)	820°C ACID BATTERY OPEN LOOP BRAYTON

DMT ~ DUAL MEDIA THERMOCLINE STORAGE

3.4.3.2 Reliability/Availability

System reliability/availability was the next selection crieeria to be applied. As reviewed in Section 3.4.2.2, this criteria is to be applied to the experimental units as well as the commercial units. The approach taken to apply this selection criteria for the commercial and experimental units is summarized below:

- Estimate availability for commercial unit - focus on differences between concepts
- Rate EE No. 1 reliability for each program duration
  - Concentrate on energy conversion subsystem
    - 0 - Rating if solely using present state-of-the-art and/or existing proven hardware
    - 1 - Rating if new equipment is employed following successful prototype operation and extensive qualification testing
    - 2 - Rating if new equipment is employed following successful prototype operation and adequate qualification testing

    These are the only acceptable ratings

For the commercial units, quantitative estimates were made of subsystem/component failure rates, maintenance requirements, mean time to repair, and system downtime (forced and planned outage) from which reliability/availability values were determined. For this system comparison, only the differences between system concepts were considered in these calculations. Availability predictions and maintenance hours for the commercial concepts are summarized on Table 3.4-7.

For the experimental units, the selection criteria are to be based on an assessment of successful plant startup probabilities and the operational reliability for at least two years after startup, with minimum forced outage attributable to design deficiencies and hardware failures. For this assessment, a qualitative evaluation was made in which the experimental concepts for each program duration were ranked (ranking of 0, 1, or 2) as indicated above.

Table 3.4-7

COMMERCIAL SYSTEM AVAILABILITY/MAINTENANCE COMPARISONS

	HTS STEAM	SYLTHERM		CALORIA		WATER STEAM	GAS/AIR
		STEAM	ORC	STEAM	ORC		
RELIABILITY/AVAILABILITY							
• FORCED OUTAGE (HR)	38.3	38.3	36.2	37.8	36.2	35.1	25.6
• PLANNED OUTAGE (HR)	104	104	104	104	104	104	140
• TOTAL OUTAGE (HR)	142.3	142.3	140.2	141.8	140.2	139.1	165.6
• TOTAL OUTAGE* (%)	4.04	4.04	3.98	4.03	3.98	3.96	4.73
• AVAILABILITY (%)	95.96	95.96	96.02	95.97	96.02	96.04	95.3
MAINTENANCE							
• CORRECTIVE (HR)	401	459	465	511	479	453	782
• PREVENTATIVE (HR)	2018	2127	1925	2090	1951	1797	1300
• TOTAL (HR)	2419	2586	2390	2601	2430	2250	2082

*BASED ON LOAD FACTOR, LF = 0.4 (3504 HR/YR)

The overall results of this assessment for the family of system candidates remaining after the program risk evaluation are shown on Table 3.4-8. The remaining 3.5 year and 4.5 year program concepts received a "0" rating because the selected subsystems/components are all based on existing equipment. All the 6.5-year program concepts received a "1" rating since substantial development is needed to achieve the near-commercial performance levels but time is available for extensive qualification testing. It should be noted that prior application of the program risk criterion excluded candidates that would be given a "2" rating or rejected by this criterion. Commercial unit availability estimates are comparatively close to each other, and thus, this criteria alone is not definitive in final concept selection.

3.4.3.3 Commercialization Potential

This criteria requires that selected concepts should use or contribute directly to the eventual systems that are likely to achieve commercial success in the late 1980's. In this context, commercial success requires that three important aspects be considered, namely, costs, flexibility, and institutional interfaces. Table 3.4-9 highlights these items together with objectives and key issues.

With respect to costs, the selected concepts should lead to systems with competitive energy costs. Energy costs, in turn, are very dependent upon capital costs, parts maintenance/replenishment costs, and operations/maintenance crew costs. System energy cost comparisons for the commercial versions of the seven candidate systems are shown on Table 3.4-10. The estimates for capital costs and annual costs of maintenance parts, make-up fluid and maintenance are approximate values only based on preliminary design definitions and exclude some common items, such as operations crew costs. The relative cost differences between concepts are more meaningful than the absolute values for system comparisons. These cost elements were then combined in accordance with JPL Report 5040-29, "The Cost of Energy from Utility-Owned Solar Electric Systems," and the corresponding constants for the cost model supplied by JPL for this application. As noted on the table, the levelized bus-bar energy costs are lowest for the HTS/steam system. The capital costs, maintenance and replacement parts costs, and bus-bar energy costs for the other systems increase from left to right across the family of system concepts shown on Table 3.4-10.

Table 3.4-8

RELIABILITY/AVAILABILITY SUMMARY

	HTS	SYLTHERM		CALORIA		WATER	GAS/AIR
	STEAM	STEAM	ORC	STEAM	ORC	STEAM	
EXPERIMENTAL UNIT RELIABILITY RATING*							
3.5 YEAR	0	X	X	X	0	X	X
4.5 YEAR	0	X	X	0	0	X	X
6.5 YEAR	1	1	1	1	1	1	1
COMMERCIAL UNIT AVAILABILITY ESTIMATE**	95.96	95.96	96.02	96.97	96.02	96.04	95.30

*RATINGS: 0 = STATE-OF-THE-ART
1 = EXTENSIVE QUALIFICATION TESTING
2 = ADEQUATE QUALIFICATION TESTING

** MODULARITY NOT COST EFFECTIVE IN IMPROVING AVAILABILITY

Table 3.4-9

COMMERCIALIZATION

ITEM	OBJECTIVES	KEY ISSUES
COSTS	SELECTED CONCEPTS SHOULD LEAD TO SYSTEMS WITH ENERGY COSTS THAT COMPETE WITH SMALL CONVENTIONAL POWER PLANTS IN THE LATE 1980'S	• CAPITAL COSTS • MAINTENANCE PARTS AND REPLENISHMENT • OPERATING AND MAINTENANCE CREW COSTS
FLEXIBILITY	SELECTED CONCEPTS SHOULD EXHIBIT THE FLEXIBILITY TO SUPPLY ENERGY OVER A WIDE RANGE OF APPLICATIONS WITHOUT MAJOR SYSTEM IMPACT	• POWER LEVEL VARIATIONS (0.5-10MW) • STORAGE VARIATION (0 STORAGE TO 0.7 CAPACITY FACTOR) • OTHER APPLICATIONS
INSTITUTIONAL INTERFACES	SELECTED CONCEPTS SHOULD SATISFY BASIC INTERFACES WITH COMMUNITIES, UTILITIES AND BUSINESS CONCERNS	• HAZARDS (FIRE, EXPLOSION, TOXICITY) • ENVIRONMENTAL POLLUTION • UTILITY KNOWLEDGE AND ACCEPTANCE OF TECHNOLOGY

With respect to flexibility, the selected system should exhibit the flexibility to supply energy over a wide range of applications without major system impact. This has been interpreted to imply rated power variations (from 0.5 to 10 MWe), capacity factor variations (from zero to 0.7) and other potential applications (such as stand-alone capabilities, shaft power, or heat source). For system comparisons, a rating was applied to each commercial system concept, as follows:

0 - Maximum flexibility
1 - Single limitation
2 - Two limitations

Table 3.4-10

SYSTEM ENERGY COST COMPARISON

	HTS	SYLTHERM		CALORIA		WATER	GAS/AIR
	STEAM	STEAM	ORC	STEAM	ORC	STEAM	
o CAPITAL COST	$2,289,000	$2,448,000	$2,570,000	$2,802,000	$2,826,000	$3,441,000	$3,474,000
o MAINTENANCE AND REPLACEMENT PARTS	$22,900	$24,500	$25,700	$28,000	$28,300	$34,400	$45,200*
o MAKE-UP FLUID	$0	$5,000	$5,000	$1,000	$900	$0	$0
o MAINTENANCE**	$36,300	$38,800	$35,900	$39,000	$36,500	$33,800	$31,000
o ENERGY OUTPUT (KWH/YR)	3,504,000	3,504,000	3,504,000	3,504,000	3,504,000	3,504,000	3,504,000
o LEVELIZED BUS BAR ENERGY COSTS (MILLS/KWH)***	144	157	161	173	172	201	209

* INCLUDES COST OF REPLACEMENT BATTERIES

** INCLUDES COST OF MAINTENANCE MAN HOURS ONLY

*** EXCLUDES OPERATIONS CREW COST

For this qualitative assessment, the steam systems for HTS, Syltherm and Caloria fluids were given a "0" rating due to their flexibility in meeting variations in power and capacity factor. The organic turbine systems were rated "1" because they were unsuitable at 10 MWe. The water/steam system with pressurized water storage and the air Brayton system with battery storage were rated "1" for limited ability to meet the 0.7 capacity factor. None of the candidates were given a "2" rating.

With respect to institutional interfaces, the selected system should satisfy basic interfaces with the communities, and the utilities. Key issues for these interfaces reduce to hazards (fire, explosion, toxicity), environmental pollution, and utility knowledge and acceptance of system technology. For system institutional interface comparisons, a qualitative rating, as described below, was applied to each commercial system concept:

0 - Comparable to existing practice
1 - Some departure from existing practice
2 - Maximum acceptable departure

The water/steam and gas/air concepts were considered to be reasonably safe, nonpolluting, and fully acceptable by utility companies, and therefore these concepts were given an "0" rating. The HTS/steam and Caloria/steam were rated as "1" because of some unfamiliarity of this technology by utility companies. The system concepts using oils (Syltherm and Caloria) with steam power conversion were given a "2" rating because of the increased hazard and pollution possibilities and unfamiliarity with the technology. The combination of oils for collector fluid with organic Rankine conversion were given 2+ because of potential hazards, pollution, and two unfamiliar technologies.

Overall system comparisons for commercialization potential considering costs, flexibility ratings and institutional ratings are summarized on Table 3.4-11. The overall system efficiency estimates are also included in this comparison. Considering all these variables, the seven basic commercial systems were ranked. The HTS/steam commercial system had the lowest cost, the highest efficiency, and good flexibility and institutional ratings. For these reasons,

Table 3.4-11

COMMERCIAL SYSTEM COMPARISONS

	HTS	SYLTHERM		CALORIA		WATER	GAS/AIR
	STEAM	STEAM	ORC	STEAM	ORC	STEAM	
• RELIABILITY/AVAILABILITY (%)	95.96	95.96	96.02	95.97	96.02	96.04	95.40
• UNIT COST ($M)	2.29	2.45	2.57	2.80	2.83	3.44	3.47
• REPLEN. & REPLACEMENT ($/YR)	22,900	29,500	30,700	29,000	29,200	34,400	45,300
• ENERGY COST (MILLS/KWH)	144	157	161	173	172	201	209
• PERFORMANCE/EFFICIENCY	0.228	0.178	0.170	0.154	0.162	0.184	0.136
• FLEXIBILITY RATING	0	0	1	0	1	1	1
• INSTITUTIONAL RATING	1	2	2+	2	2+	0	0
• COMMERCIALIZATION RANK	1	2	3	4	5	6	7

it was considered as the best of the commercial concepts and was therefore ranked as number "1". Corresponding rankings for the other commercial candidates are shown on the bottom line of Table 3.4-11.

3.4.3.4 Program Costs

To satisfy this requirement, concepts should be selected that minimize the estimated development and capital costs of Phase II and Phase III. In order to compare the candidate concepts, preliminary estimates of the Phase II and III costs were made for each of the concepts proposed for the 3.5, 4.5 and 6.5-year programs. Program cost differences are summarized on Table 3.4-12. The costs of system design and common elements have been excluded. The cost differences primarily represent the costs of the hardware required for development testing and for the installation of the first experimental plant. As can be noted on the table, the HTS/steam concept has the lowest total costs for all three program durations, however the cost differences between concepts are not substantial. The relative costs for all candidates considered are shown for completeness even though several were screened by the program risk criterion.

3.4.4 System Comparisons and Selection

The seven candidate systems, described in Section 3.4.1, have been evaluated in Section 3.4.3 for each of the four primary selection criteria (reliability/availability, program risk, commercialization, and program cost). In this section, an overall system comparison and a final system selection will be given. Table 3.4-13 presents the final results of these comparisons.

The preferred system for each of the three startup programs is the HTS/steam system, as indicated by the check marks on Table 3.4-13. This system has: (1) excellent rankings for reliability/availability and program risk, (2) the best commercialization ranking which includes considerations of energy costs, flexibility and institutional interface aspects, and (3) the lowest program costs. The HTS/steam system is clearly the best choice of the candidate systems.

Table 3.4-12

PROGRAM COST DIFFERENCES

COSTS EXCLUDING DESIGN AND COMMON ELEMENTS ($1000 1978)

PROGRAM	HTS STEAM	SYLTHERM		CALORIA		WATER STEAM	GAS/AIR
		STEAM	ORC	STEAM	ORC		
3.5 YEAR							
- PHASE II	383	711			840		
- PHASE III	4,951	5,148			5,990		
- TOTAL	5,334	5,859			6,830		
4.5 YEAR							
- PHASE II	1,703	2,551	2,551	2,041	1,391	1,690	1,400
- PHASE III	4,195	4,426	4,609	5,373	5,829	5,388	6,440
- TOTAL	5,898	6,977	7,160	7,414	7,220	7,078	7,840
6.5 YEAR							
- PHASE II	3,035	2,907	2,807	2,330	2,230	2,210	2,000
- PHASE III	4,070	4,426	4,609	5,373	5,203	5,388	7,110
- TOTAL	7,105	7,333	7,416	7,703	7,433	7,598	9,110

Table 3.4-13

EXPERIMENT PROGRAM COMPARISONS

	HTS	SYLTHERM		CALORIA		WATER	GAS/AIR
	STEAM	STEAM	ORC	STEAM	ORC	STEAM	
<u>3.5 YEAR PROGRAM</u>	✓						
• RELIABILITY/AVAILABILITY*	0	–			0		
• PROGRAM RISK	ACCEPTABLE	REJECTED			ACCEPTABLE		
• PROGRAM COSTS** ($M)	5.3	5.9			6.8		
<u>4.5 YEAR PROGRAM</u>	✓						
• RELIABILITY/AVAILABILITY	2/0	–	–	2/0	0	–	–
• PROGRAM RISK	ACCEPTABLE	REJECTED	REJECTED	ACCEPTABLE	ACCEPTABLE	REJECTED	REJECTED
• PROGRAM COSTS** ($M)	5.9	7.0	7.2	7.4	7.2	7.1	7.8
<u>6.5 YEAR PROGRAM</u>	✓						
• RELIABILITY/AVAILABILITY	1	1	1	1	1	1	1
• PROGRAM RISK	ACCEPTABLE	MARGINAL	MARGINAL	ACCEPTABLE	ACCEPTABLE	ACCEPTABLE	ACCEPTABLE
• PROGRAM COSTS** ($M)	7.1	7.3	7.4	7.7	7.4	7.6	9.1
• COMMERCIALIZATION RANK	1	2	3	4	5	6	7

*RATINGS: 0 - STATE OF THE ART
1 - EXTENSIVE QUALIFICATION TESTING
2 - ADEQUATE QUALIFICATION TESTING

**EXCLUDES COMMON ELEMENTS

✓ - PREFERRED SYSTEMS

The major characteristics of the preferred experimental systems are summarized on Table 3.4-14. For the 3.5 and 4.5-year programs, Hitec will be used at the temperature limits indicated on the table. For the 6.5-year program, the binary nitrate (HTS) will be used which has higher operation temperature limits, as indicated. For the 3.5 and 4.5-year programs, a simple two-tank thermal storage system will be employed which minimizes development requirements and program risks. For the 6.5-year program, a dual media thermocline storage tank is employed. The primer mover for the 3.5-year and 4.5-year programs shall be an existing axial turbine. For the 6.5-year program, the radial outflow turbine currently under development by Energy Technology, Inc. (ETI) will be used. The major features of the commercial power plant that will evolve from the preferred experimental systems are reviewed in the following section.

3.4.5 Commercial System Definition

As a precursor activity to the selection of the 3.5, 4.5, and 6.5-year candidate systems, a commercial system definition was formulated. This system, which was postulated for the late 1980's time frame, served as a reference standard for the selection of the three candidate development systems since all of these systems should logically lead to the ultimate commercial system.

Table 3.4-14

SELECTIONS FOR THREE PREFERRED SYSTEMS

	3-1/2 Years	4-1/2 Years	6-1/2 Years
Receiver Fluid	Hitec	Hitec	HTS
Temperature Limit	450°C (842°F)	450-510°C (842-950°F)	510-580°C (950-1076°F)
Thermal Storage	2-Tank	2-Tank	Dual Media Thermocline
Prime Mover	Axial Turbine	Axial Turbine	Radial Turbine

The principal features of the commercial system are shown in Figure 3.4-2 and in Table 3.4-15. The state point conditions were defined on the basis of anticipated hardware and material development over the next 10-year period.

The heliostat design assumed for the commercial system is based on the 49 m^2 Second Generation Heliostat which is currently under design and development at MDAC as part of a continuing DOE heliostat development program.

It is assumed that the results of this parallel DOE heliostat development program can be utilized directly by the commercial system. Performance analyses have been carried out to verify the optical compatibility of the heliostat with the anticipated commercial system receiver.

The baseline commercial receiver will be of a spiral-tube, cavity-cone configuration. Towers required to support the receiver will be of the guyed design. The thermal storage subsystem will employ a single tank which uses a dual

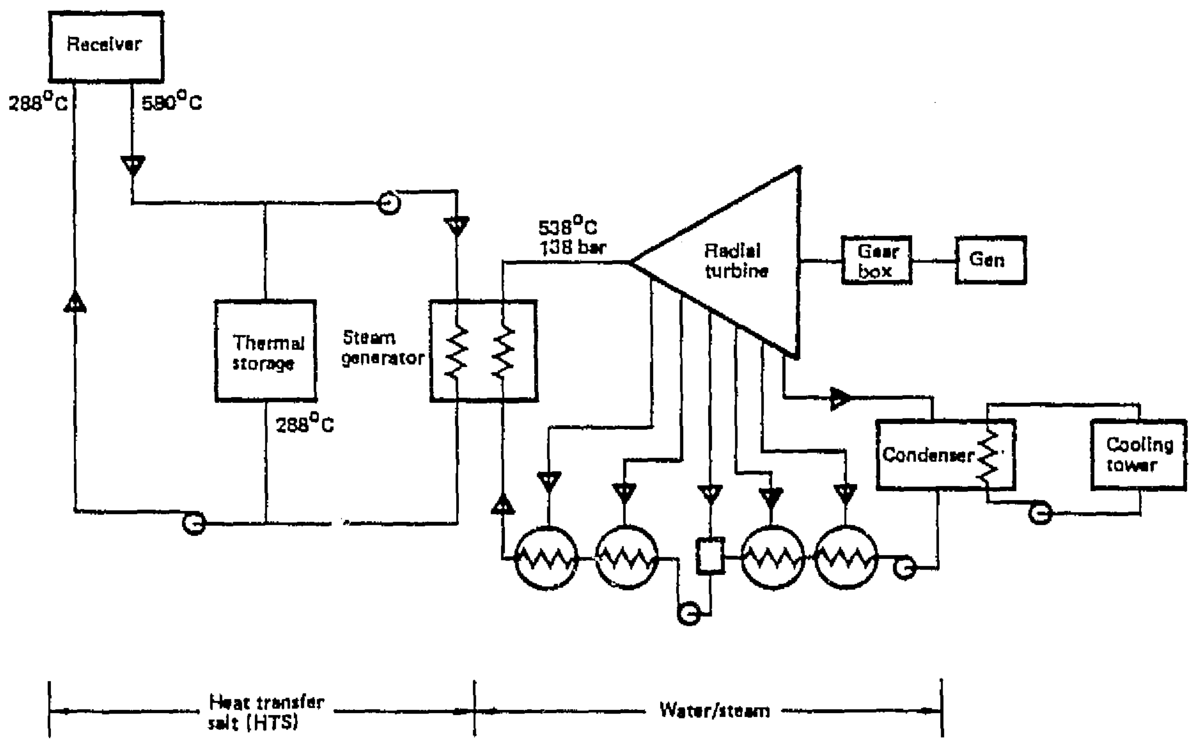


Figure 3.4-2. Commercial System Schematic

Table 3.4-15

PRELIMINARY COMMERCIAL SYSTEM CHARACTERISTICS

Receiver Fluid	Heat Transfer Salt (HTS)
Peak Receiver Power	4.40 MWt
Annual Collected Energy	10,140 MWHt
Plant Parasitic Load	0.1 MWe
Turbine Cycle Efficiency	0.38
Thermal Storage Rating	8.3 MWHt
Receiver Type	Spiral Cone
Number of Heliostats	122
Heliostat Size	49 m^2

The tank is constructed of 316 stainless steel, which is required to withstand the maximum HTS temperature. To prevent freeze-up, electrical immersion heaters are located throughout the tank.

The energy transport subsystem is made up of standard pipes, sensors, control valves, and pumps. All pipes and equipment exposed to the maximum HTS temperature will be 316 stainless steel while low temperature elements will be carbon steel. Horizontal centrifugal pumps will be used to provide HTS circulation.

The power conversion subsystem employs an advanced radial turbine which has an expansion efficiency of 84 percent and can result in a turbine cycle efficiency of 38 percent by employing the five turbine extractions shown schematically in Figure 3.4-2. The inlet conditions of 538°C, 138 bar represent practical values subject to material and exit moisture constraints. Conventional condenser, feedwater heater, and heat rejection equipment are used. Because of high cycle efficiency, these elements are in general smaller than the corresponding elements associated with the short term experimental systems.

Plant control for the commercial system employs the hardware and software developed as part of the experimental programs. It will be configured to minimize the involvement of operators and maximize the potential for unattended operation. All control commands will be initiated through a central processor or manually through a common keyboard.

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Appendix A
PRELIMINARY SYSTEMS COST ANALYSIS

Costing during preliminary system analysis was mainly concerned with determining the significant, relative cost differences between alternatives for the various subsystems and the impact an alternative may have on other subsystem costs. This analysis has been supported through the development of a cost data package on important material and equipment unit costs, installation factors, cost sensitivities, and subsystem cost baselines. The cost analysis proceeds with the extension of this data in accord with engineering variables developed relative to a constant net energy output (i.e., 1 MWe). Further detail on the costing groundrules and approach, unit costs and cost sensitivity, and baseline cost methodology is provided in the following subsections.

A.1 COSTING ASSUMPTIONS AND APPROACH

Table A-1 indicates assumptions and the general costing approach used for comparative cost analysis. As requested, cost of energy calculations employ the constants provided by the 12 July 1978 JPL letter on Phase I Study Data. Other major assumptions concerning programmatics and the approach employed are shown in the second half of the table. Perhaps the most significant projection is that the required heliostats will be drawn from a going high-rate production line. Also, no cost reduction from current prices for site construction oriented cost elements has been assumed. This is somewhat controversial but is based on the view that each customer, general contractor, and set of suppliers may differ between installations.

For convenience during preliminary analysis, the cost breakdown was oriented to JPL "Table E-2" (Ref. 1) only down to the subsystem level. Lower level breakdown simply follows material and equipment lists with costs accounting for certain major assemblies such as heliostats or the receiver. As the study progressed

Table A-1

GROUNDRULES AND ASSUMPTIONS

JPL RECOMMENDED ASSUMPTIONS		Factors
A)	RAW LAND PER ACRE	\$5,000
B)	COST OF CAPITAL TO A "TYPICAL" UTILITY, k	0.086
C)	RATE OF GENERAL INFLATION, g	0.060
D)	ESCALATION RATE FOR CAPITAL COSTS, g_c	0.060
E)	ESCALATION RATE FOR OPERATING COSTS, g_o	0.070
F)	ESCALATION RATE FOR MAINTENANCE COSTS, g_m	0.070
G)	CAPITAL RECOVERY FACTOR (8.6%, 30 YRS), CRF_k, N	0.0939
H)	FIXED CHARGE RATE, ANNUALIZED, FCR	0.1565
I)	ACCOUNTING LIFETIME, n	30 YRS
J)	SYSTEM LIFETIME, N	30 YRS
K)	INSURANCE + "OTHER TAX" FRACTION, $\beta_1 + \beta_2$	0.020
L)	INVESTMENT TAX CREDIT FRACTION, α	0.100

MDAC ASSUMPTIONS - PRELIMINARY

- BASE PRICES IN 1978 DOLLARS
- 25,000 HELIOSTATS PER YEAR PRODUCTION LINE
- CATALOG PRICES FOR COMMON MATERIALS
- VENDOR QUOTES ON SPECIALIZED ITEMS
- 100 PLANTS INSTALLED PER YEAR -- 1990's TIMEFRAME
- NO COST REDUCTION FOR POWER CONVERSION, TOWER, AND TANK COSTS
- MAINTENANCE & SPARES BASED ON PRELIMINARY FAILURE RATES
- LABOR BASED ON RESOURCE LOADS, STANDARDS AND HISTORIC FACTORS

towards the selected systems, costs were accumulated in accordance with the JPL cost breakdown structure (Table E-2).

A.2 SUBSYSTEM COST ELEMENTS

In support of the selection process during the system analysis, the unit costs shown in Table A-2 and the associated Table A-3 along with Figures A-1 and A-2 were generated. These are equipment and material costs only, and do not include installation unless indicated individually. These costs were obtained through vendor quote or construction estimating catalogs described later.

Additional cost sensitivity relationships were developed in Section 3 for subsystems and components. Tower cost sensitivity was developed by Stearns-Roger through direct estimates at heights of 42 and 48 meters (138 and 158 feet) and receiver weights of 7,250 and 34,000 kilograms as follows:

	Cost ($000)			
	Free Standing		Guyed	
Receiver Weight (Kg [lb])	42 m	48 m	42 m	48 m
7,250 (16,000)	$172	$193	$ 87	$ 97
34,000 (75,000)		$284		$152

Receiver cost sensitivity has been expressed as follows:

$$C = A \times ESA^{.64} \left(\frac{\left(\frac{ESA}{EOD}\right) * IDP^{-1}}{12.92} \right)^{.284}$$

where:

A = $2,213 for carbon steel absorbers
A = $2,539 for 4130 steel absorbers
A = $3,037 for stainless steel absorbers
ESA = The product of the plain surface area of the absorber and $\pi/2$ in square feet

Table A-2

UNIT MATERIAL COSTS - SMALL POWER SYSTEM EXPERIMENT

(Sheet 1 of 4)

Element	Size	Unit	Unit Cost ($)
COLLECTOR SUBSYSTEM			
Receiver			
Plate, low-carbon (LC) steel		lb	0.28
Corrugation, LC steel		lb	0.82
Insulation, fiber glass batting		ft^2	0.36
Pipe, stainless steel (SS) SCH 40	1.0"	ft	4.83
Pipe, SS SCH 40	3.0"	ft	15.55
Pipe, 4130 13 ga	2.0"	ft	2.50
Insulation, 3.0 in-thick sheet SCH 40		ft^2	1.93
Insulation, pipe	3.0"	3-ft l	9.04
Insulation, pipe	1.0"	3-ft l	7.39
Insulation, pipe	8.0"	ft	5.77
Insulation, pipe	3.0"	ft	3.21
Insulation, pipe	14.0"	ft	15.66
I-Beams, 3.0-inch		lb	0.20
Heliostats			
Sheet, LC steel, galvanized	0.020	lb	0.24
Sheet, LC steel, galvanized	0.063	lb	0.257
Tube	10.0	lb	0.21
Channel, LC steel		lb	0.24
Tube		lb	0.266
Flange, steel		unit	180.00
Casting, Azimuth Drive		lb	0.90
Drive, Azimuth		unit	550.00
Bearing		unit	170.00

Table A-2

UNIT MATERIAL COSTS - SMALL POWER SYSTEM EXPERIMENT

(Sheet 2 of 4)

Element	Size	Unit	Unit Cost ($)
Drive, Elevation		unit	300.00
Casting, Elevation		lb	0.88
Motor		unit	110.00
Concrete Foundation w/Reinforcements		yd^3	55.00
Tower			
Excavation		yd^3	2.00
Consolidated Backfill		yd^3	2.00
Concrete foundation, installed		yd^3	270.00
Structure Steel Tower, installed		Ton	1,475.00
Guy Wires		ft	1.34
Paint, applied		Ton	75.00
Electrical		unit	16,000.00
ENERGY STORAGE SUBSYSTEM			
Hitec		lb	.36
Syltherm		gal	19.00
Caloria		lb	.13
Medium Transport		lb	.10
Rock & Sand		Ton	19.00
Rock Transport		Ton	5.00
Tank Insulation		ft^3	51.84
Tanks - LC		(Figure A-1)	
Tanks - SS		(Figure A-2)	
Pressurized Tank (A-285) 600 psi	4,400 ft^3	unit	100,000

Table A-2

UNIT MATERIAL COSTS - SMALL POWER SYSTEM EXPERIMENT

(Sheet 3 of 4)

Element	Size	Unit	Unit Cost ($)
ENERGY TRANSPORT SYSTEM/GENERAL PIPING			
Pipe, Low Carbon Steel SCH 40	1.5"	ft	1.39
Pipe, Low Carbon Steel SCH 40	2.0"	ft	2.20
Pipe, Low Carbon Steel SCH 40	2.5"	ft	3.01
Pipe, Low Carbon Steel SCH 40	3.0"	ft	3.83
Pipe, Low Carbon Steel SCH 40	4.0"	ft	5.45
Pipe, Low Carbon Steel SCH 40	5.0"	ft	7.07
Pipe, Stainless Steel	2.0"	ft	11.96
Trace Heating. Elec Resist Elements			
250°F Δ, 20 PTV 1	1.5-3.0"	ft/in diam	13.94-17.35
30 PTV 1	1.5-3.0"	ft/in diam	17.70-22.03
300°F Δ, 20 PTV 1	1.5-3.0"	ft/in diam	18.33-22.37
30 PTV 1	1.5-3.0"	ft/in diam	23.27-28.41
Piping Insulation, 4.0-in thick			
	1.5"	ft	20.70
	2.0"	ft	21.00
	2.5"	ft	21.90
	3.0"	ft	23.65
	4.0"	ft	25.30
	5.0"	ft	27.05
Valves and Pumps		(Table A-3)	

Table A-2

UNIT MATERIAL COSTS - SMALL POWER SYSTEM EXPERIMENT

(Sheet 4 of 4)

Element	Size	Unit	Unit Cost ($)
POWER CONVERSION SUBSYSTEM	1 MWe		
Radial Outflow Turbine/Gearbox		unit	68,000
Axial Steam (CS casing)		unit	199,000
Axial Steam (CM casing)		unit	330,000
Generator		unit	31,000
Organic Subcritical Turbine Sys		unit	600,000
Organic Supercritical Turbine Sys		unit	650,000
Deaerator		unit	4,000
LP Heater #1		unit	1,363
HP Heater #3		unit	1,827
HP Heater #4		unit	1,567
HP Heater #5		unit	1,285
Steam Generator Feed Pump		unit	17,500
Condensate Pump		unit	11,500
Circ Water Pump		unit	4,500
Condenser Exhaust Pump		unit	11,000
Steam Generator		unit	55,000
Cooling Tower (5" Hga)	3 MWth	unit	35,000
Condenser (5" Hga)	3 MWth	unit	3,000
Condenser (5" Hga)	2 MWth	unit	2,000
Cooling Tower (2.5" & 5" Hga)	2 MWth	unit	25,000
Condenser (2.5" & 5" Hga)	2 MWth	unit	4,000
Demineralizer		unit	30,000
Condensate Polisher		unit	32,000
Boiler Chemical Feed System		unit	25,000
Cooling Tower Chem Feed System		unit	18,000
Water Treatment Panel		unit	75,000
Cooling Tower Control Panel		unit	4,000

Table A-3

PUMP & VALVE COSTS

VALVES

Size, Mat'l	1.5"CS	2"CS	2"SS	2"SS	2.5"CS	3"CS	4"CS	5"CS	5"CS
Psi	300	300	150	300	300	300	150	150	300
Flow Ctl, Posit & Airset	$915	$1065	$1060	$1501	$1245	$1365	$1795	$2330	$2330
Remote (On, Off), Act									
2-Way	667	797	1218	1218	959	1081	1502	2009	2009
3-Way	-	-	-	-	-	-	1867	-	-
Check	211	245	250	281	341	341	358	436	670
Manual (Gate)	205	297	232	296	389	389	375	532	746

PUMPS

Size, Mat'l	2.5"CS	2"SS	2"CS	1.5"SS	4"CS	4"CS	5"CS	4"CS
Flow Rate	158 GPM	108 GPM	82 GPM	57 GPM	410 GPM	320 GPM	525 GPM	370 GPM
Head Rise	220 PSI (270 FT)	13 PSI (16 FT)	185 PSI (227 FT)	14 PSI (17 FT)	107 PSI (328 FT)	5 PSI (15 FT)	190 PSI (580 FT)	5 PSI (15 FT)
Oper Temp	550°F	950°F	550°F	950°F	450°F	750°F	425°F	600°F
In Line	$4250	$ -	$2850	$ -	$3600	$4800	$4550	$4400
Submerged	$7200	$12,900	$7200	$12,900	$ -	$ -	$ -	$ -

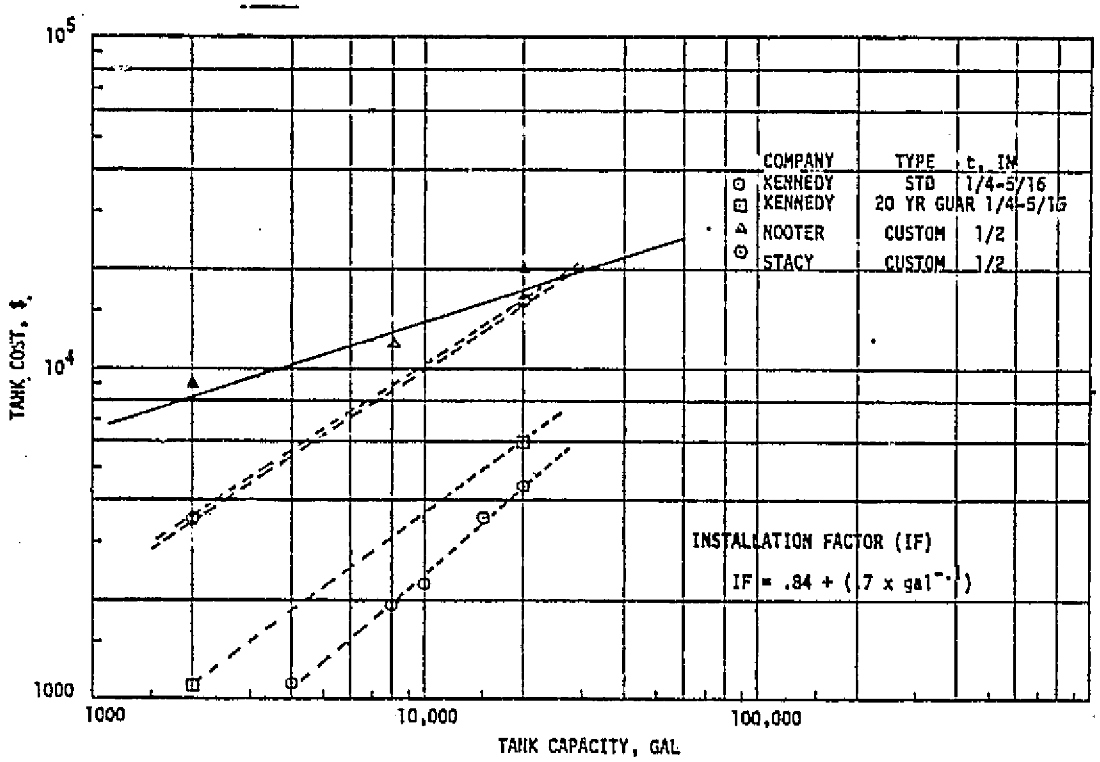

Figure A-1. Carbon Steel Tank Costs

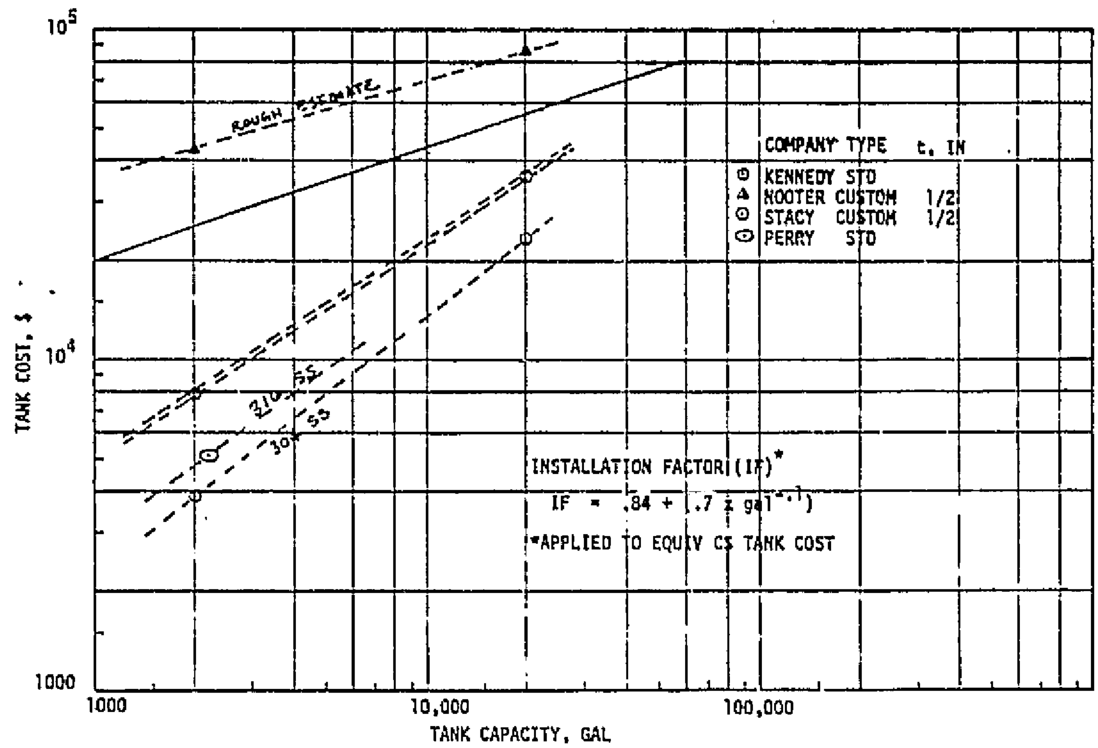

Figure A-2. Alloy Steel Tank Costs

EOD = The circular equivalent profile diameter of absorber panels in feet

IDP = Internal diameter of absorber tubing in inches

The ratio developed in the second part of the equation considers packaging complexity relative to a baseline complexity (12.92).

Heliostat costs are most usefully related to square meters or dollars per unit as follows:

$\$/m^2$ = 81

$/unit = 3,973

These relationships are based on the results of the DOE/Sandia funded Prototype Heliostat Study adjusted to account for added costs expected with small field installations.

An explanation of the methods used in developing heliostat and other costs is described in the subsection that follows.

A.3 COST METHODOLOGY

The life cycle cost methodology has been to incorporate study results using extensions of the above costs in the JPL supplied Cost of Energy program. The program is applied where significant differences in funds flow are apparent. An indication of the source of the cost inputs is provided by the following descriptions.

A.3.1 Sources of Unit Costs

Unit costs were obtained from the following vendors and supply catalogs:

Equipment	Source
Tanks	Kennedy Tank & Mfg Co., Inc.
	Nooter Corporation
	Perry Equipment Co., Inc.
	Stacey Mfg. Company

Equipment	Source
Steam Turbine	
Radial Steam	Energy Technology Inc.
Axial Steam	Thermo-Electron Corporation
	Terry Turbine
Organic Turbine	Thermo-Electron Corporation
	Sunstrand Energy Systems
Gas Turbine	Solar Turbines International (ETI)
Generator	Electric Machinery Company
Gear Box	Western Gear
Pipe	Stearns-Roger
	"The Richardson Rapid System" 1978-79 edition
Pipe Insulation	"The Richardson Rapid System"
Trace Heating	"The Richardson Rapid System"
Valves	"The Richardson Rapid System"
	Honeywell: "Measurement and Control Instrumentation"
Pumps	"The Richardson Rapid System"
	Ingersoll-Rand
	Lawrence Pumps, Inc.
	Dean Brothers Pumps, Inc.

A.3.2 Heliostat Methodology

Costs developed for the Second Generation Heliostat study were used as a basis for this methodology. The approach employed in developing costs for the 25,000 heliostats per year scenario is based on annual resource loading for labor and, in the main, on vendor information quoted at the level of parts and materials required to support annual factory output. For certain electronic components that currently do not exist, the costs of like components were used based on the projection that demand will cause the required components to be produced in the near future. The balance of material costs (e.g., fasteners) were based on catalog prices, while transportation costs were based on the experience at MDC in Long Beach who operate their own transportation fleet.

Although manhours have been primarily developed through manning of the required factory equipment, direct support hours for planning, sustaining tooling, and product support were based on standard factors. Quality control hours were derived by a specially studied factor for the Second Generation heliostat. Other areas such as material handling and supervision were covered within the applied burden rates.

Various factors have been applied to the costs derived in the above manner. Material has been factored by visibility, scrap and rework, and fee. Labor hours have been adjusted to reflect scrap and rework, and efficiency. Fee was covered in the labor rate. Applied efficiency factors mainly cover impacts on lapsed time while other efficiencies are implicit in the crew loads. This is most apparent in the field where a crew of 7 may be accomplishing a task, but at any one time only 2 or 3 members may be actually involved at any one time.

For the rate of 25,000 units per year, cost reduction curves have been applied only to factory labor. In the 25,000 unit scenario, production was assumed to commence after 100,000 heliostats have been produced for pilot plants, demonstration plants, and first commercial plants, and to continue out to unit 600,000 for a total of 500,000 heliostats over 20 years. The manloads have been projected as those required at the start of the second year of rate production in the factory, or at unit 125,000. In order to arrive at unit hours in the tenth year of operations, labor has been extended down on 89 percent cost reduction curve from unit number 125,000 to the average hours for units 335,000 to 360,000. This is intended to reflect tooling improvements, more efficient alignment of material flows, and better utilization of manpower as the plant matures.

Applied labor and burden rates vary between factory, field, and operations. Factory rates are based on low side National average labor costs and MDAC burden and GA&A experience at volume production facilities. Installation rates were based on Riverside, California trade labor and fringe rates adjusted to allocate distributable cost. Both the factory and field rates include an 8 percent fee.

A.3.3 Receiver Methodology

Costs have been developed for the receiver using a detailed "bottoms-up" procedure similar to that used by MDAC in committing to contractual effort. Parts and materials mainly have been costed based on vendor quotes obtained for a level of 2,000 end item sets procured over a 20 year period. Costs for some common equipment were obtained from catalogs. First unit labor has been estimated for each part by fabrication and assembly detail using experienced manufacturing estimators and manufacturing engineers.

Factory first unit hours have been extended down an 86 percent cost reduction curve to arrive at an average unit cost for the first 100 units. These results have been taken as a conservative average estimate for the 100 per year production rate. A like procedure for field effort has been applied using a 94 percent cost reduction curve. Raw materials were quoted directly for 2,000 units, so that material costs were adjusted back up a 95 percent curve to the average cost expected for 120 units in order to represent the lower production rate. Purchased equipment quotes were assumed supplied at a 30 percent discount for volume procurement and were rediscounted to 20 percent for the lower rate of production.

Basic labor and material estimates were extended by various factors in order to bring estimates up to experience levels. Labor includes factors for visibility, setup, efficiency, rework/scrap, shop liaison, and processes such as passivation. Also, allocations have been made for quality control, production control and planning, sustaining tooling, product support and other miscellaneous expense. Material dollars have been factored to include visibility, scrap, transportation, and material burden.

Labor estimates were then extended by appropriate composite industry factory labor, fringe and burden rates for a plant doing the projected volumes of production. Current composite trade labor rates, fringes, and general contractor field support and equipment rental rates were applied to field labor hours. The applied burdened factory rate is 23 dollars per hour while the applied field rate including general contractor distributables ranges between 26 to 29 dollars per hour.

A.3.4 Operations and Maintenance Methodology

Operations and maintenance costs were based on both resource loading and direct estimates of hours, unit investment cost for replaced or spared parts, and on quotes or prior study information on operations materials such as washing solution. Spares and repair parts were the product of annual failures (based on failure rates tables), hardware unit costs estimated for investment, and repair or replacement factors. Corrective maintenance was the product of crew size and lapsed time or a direct hour estimate for bench labor, annual failures, repair factors for bench labor, and burdened labor rates. Scheduled maintenance was based on direct estimates or crew size and burdened labor rates, material quotes, and estimated frequencies. Results were factored to consider efficiency, added first year failures or problems, and refix where the first attempt at repair is not successful and must be redone. The O&M labor rate was estimated at 15 dollars per hour.

A.4 PROGRAM COST METHODOLOGY

In addition to the impact of designs on commercial costs and the resulting cost of energy, the Phase II and III program development costs also were considered. However, this consideration was applied only after serving other selection criteria, including cost of energy. Where analyzed, the various Phase II and III development costs associated with alternative systems were compared both absolutely and on the basis of discounted savings (or dissaving) versus increased (or decreased) development program investment. The resulting net present values may serve as a basis for supporting or rejecting development/benefit variations.

Appendix B
PRELIMINARY AVAILABILITY ESTIMATES

Preliminary availability estimates were generated to aid in selection between the alternative system candidates. In addition, the increase in availability with system modularity and the associated costs were also evaluated.

B.1 SUBSYSTEM REQUIREMENTS AND ASSUMPTIONS

The availability of a solar power plant should be equal to or better than a fossil (or nuclear) plant of similar size. This should be true if solar plants are to be competitive with fossil plants and from an engineering standpoint can be true due to the unique nature of solar plants. A large percentage of the failures of a solar power plant will occur in the collector (heliostat) field, but the failure of one heliostat results in a loss of about 0.67 percent of sun power to the receiver. The electric power industry does not consider a loss of less than two percent as a system outage (partial or total); therefore, we must experience a loss of at least three heliostats simultaneously (or within the repair time) before a system outage is charged. Therefore, a failure in the heliostat field that affects only one heliostat (all but the power junction box) are noncritical in a single failure analysis, and due to the low failure rates for an individual heliostat (0.3 failures/year), the contribution to system downtime would be small for a multiple failure analysis. In addition, a small number of heliostats (1 to 3) could be added to assure that 100 percent of available sun energy is focused onto the receiver at all times.

The remaining components (except the energy storage) are similar to conventional power plants. However, the solar plant does not operate 24 hours per day. The solar dominated portions (collector, energy transport, energy storage) operate only when sun power is available (average of 11.7 hrs/day) while the power generation system is scheduled for a 40 percent load factor (average of

9.6 hours/day). Therefore, there is a large block of time each day (night) to perform preventative and corrective maintenance. Therefore, the downtime during expected operating periods should be less for a solar power plant.

Analysis of the baseline version of the 1 MW solar power plant produced results as shown in Table B-1. The operating availability is calculated to be about 96 percent with a forced outage rate of one percent and a planned outage rate of three percent. The total maintenance manhours was about 2,500 hours/year. These results compare favorably with an extrapolation of data from larger power plants (References 49 and 50).

B.2 AVAILABILITY ANALYSIS

A preliminary availability and maintenance analysis was conducted on alternative subsystem candidates. Three variations of the energy transport and energy storage subsystems were analyzed -- dual media, two-tank and trickle charge. Four variations in the power conversion subsystem were also analyzed -- one using a near-term axial turbine and three using a radial turbine at progressively more advanced steam conditions. Various combinations of these seven cases represent many of the alternate configurations discussed in Section 3.2.

A bottoms-up approach was used in this preliminary analysis. Each component type (control valve, check valve, pump, etc.) was assigned a value for failures per year, mean time to repair (MTTR), number of men to perform the repair, and the expected downtime for planned (preventative) maintenance, from previous studies of similar systems. This basic data allowed a calculation of the forced outage time (hours/year) and rate (percentage of total time), the planned outage time and rate, and the required maintenance manhours required to maintain the system.

Results of this analysis are summarized on Table B-2. Basic data for each type of component in each subsystem is shown in Table B-3 for each of the seven configurations. These preliminary results were later upgraded by a more detailed analysis of the selected configurations.

Table B-1

AVAILABILITY ANALYSIS RESULTS

	Collector	Power Conversion	Energy Transport	Energy Storage	Master Control	System
Total Failures/yr	47.88	1.56	0.60	0.17	1.86	52.07
Critical Failures/yr	0.38	1.20	0.35	0.13	0	2.06
Forced Outage/hrs/yr	2.95	29.40	2.76	0.61	0	35.72
Planned Outage, hrs/yr	15.60	104.00	15.60	15.60	0	104.00**
Total Outage, hrs/yr	18.55	133.40	18.36	16.21	0	139.72
Forced Outage Rate, %	0.0764	0.8390	0.0715	0.0158	0	1.0027
Planned Outage Rate, %	0.4040	2.9680	0.4040	0.4040	0	2.9680*
Total Outage Rate, %	0.4804	3.8070	0.4755	0.4198	0	3.9707
Operating Availability, %	99.52	96.20	99.52	99.58	100.00	96.03
CMTBF** (hrs)	10,160	2,920	11,031	29,700	-	1,769
CMTTR*** (hr)	7.76	24.50	7.89	4.69	-	17.34
Corrective MMH/yr****	289	145	14	1	4	453
Preventive MMH/yr	548	1,388	50	68	10	2,064
Total MMH/yr	837	1,533	64	69	14	2,517

*If all planned outages performed simultaneously
**Cumulative mean time to failure
***Cumulative mean time to recover
****Maintenance man-hours

Table B-2

RESULTS OF AVAILABILITY/MAINTENANCE STUDY

SUBSYSTEM / PARAMETER	ENERGY TRANSPORT ENERGY STORAGE			POWER CONVERSION			
	DUAL MEDIA STORAGE	DUAL TANK	TRICKLE CHARGE	AXIAL TURBINE	RADIAL 288°C TURBINE	RADIAL 371°C TURBINE	RADIAL 482°C TURBINE
FORCED OUTAGE HRS (%)	4.73 (0.1225)	4.36 (0.1388)	7.35 (0.1904)	29.83 (0.8513)	30.07 (0.8582)	30.38 (0.8670)	30.54 (0.8716)
PLANNED OUTAGE HRS (%)	15.60 (0.4040)	15.60 (0.4040)	15.60 (0.4040)	104 (2.9680)	104 (2.9680)	104 (2.9680)	104 (2.9680)
TOTAL OUTAGE HRS (%)	20.33 (0.5265)	20.96 (0.5428)	22.95 (0.5944)	133.83 (3.8193)	134.07 (3.8262)	134.38 (3.8350)	134.54 (3.8396)
AVAILABILITY, %	99.47	99.46	99.41	96.28	96.27	96.26	96.26
CORRECTIVE MAINTENANCE MMH	15.82	17.11	23.46	142.62	143.21	143.81	144.39
PREVENTATIVE MAINTENANCE MMH	204.00	204.00	204.00	1152.00	1220.00	1288.00	1356.00
TOTAL MAINTENANCE MMH	219.82	221.11	227.46	1294.62	1363.21	1431.81	1500.39

Table B-3

DUAL MEDIA - ENERGY TRANSPORT

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	3	3861	24.9	4.7	117.2	2	234.5	0	0.235	0.352	0.705	0
REMOTE VALVES	7		2.0	4.2	8.2	2	16.5	0	0.017	0.015	0.119	0
CHECK VALVES	1		15.4	3.5	54.1	2	108.1	0	0.108	0.054	0.108	0
HAND VALVES FTRO	25		1.16	3.5	4.05	2	8.1	0	0.008	0.101	0.200	0
HAND VALVES FTRC	0		0.39	3.5	1.4	2	2.7	0	0.003	0	0	0
PUMPS	2		116.8	9.7	1123.6	4	4494.2	0	4.494	2.247	8.988	0
SENSORS	5		3.9	2.0	7.7	2	15.44	0	0.015	0.039	0.075	0
H.E.	3		46.3	3.5	162.2	2	324.3	68	68.324	0.487	204.97	15.6
MIXER TANK	1		3.9	10	39.0	4	156.0	0	0.156	0.039	0.156	0
HEATERS	1		39	20	780	4	3120	0	3.12	0.780	3.12	0

TABLE B-3

DUAL MEDIA - ENERGY STORAGE

(SHEET 2 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVE	0	3861	24.9	4.7	117.2	2	234.5	0	0.235	0	0	0
REMOTE VALVE	0		2.0	4.2	8.2	2	16.5	0	0.017	0	0	0
HAND VALVE FTRC	4		1.16	3.5	4.05	2	8.11	0	0.008	0.016	0.032	0
CHECK VALVE	1		15.4	3.5	54.1	2	108.1	0	0.108	0.054	0.108	0
REGULATOR	1		69.5	4.7	326.6	2	653.3	0	0.653	0.327	0.653	0
SENSORS	10		3.9	2.0	7.7	2	15.4	0	0.015	0	0.150	0
RELIEF VALVES	1		38.6	3.5	135.1	2	270.3	0	0.270	0.135	0.270	0
HEATERS	10		0.04	10	0.4	2	0.8	0	0.001	0.004	0.010	0
TANK	1	8760	8.8	10	88	4	156	0	0.156	0.088	0.156	0

Table B-3

DUAL TANK - ENERGY TRANSPORT

(SHEET 3 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	3	3861	24.9	4.7	117.2	2	234.5	0	0.235	0.352	0.705	0
REMOTE VALVES	6	↓	2.0	4.2	8.2	2	16.5	0	0.017	0.049	0.102	0
CHECK VALVES	1	↓	15.4	3.5	54.1	2	108.1	0	0.108	0.054	0.108	0
HAND VALVES FTRO	19	↓	1.16	3.5	4.05	2	8.1	0	0.008	0.077	0.152	0
HAND VALVES FTRC	0	↓	0.39	3.5	1.4	2	2.7	0	0.003	0	0	0
PUMPS	2	↓	116.8	9.7	1123.6	4	4494.2	0	4.494	2.247	8.988	0
SENSORS	5	↓	3.9	2.0	7.7	2	15.44	0	0.015	0.039	0.075	0
H.E.	3	↓	46.3	3.5	162.2	2	324.3	68	68.324	0.487	204.97	15.6
MIXER TANK	1	↓	3.9	10	39.0	4	156.0	0	0.156	0.039	0.156	0
HEATERS	1	↓	39	20	780	4	3120	0	3.12	0.780	3.12	0

Table B-3

DUAL TANK - ENERGY STORAGE

(SHEET 4 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVE	0	3861	24.9	4.7	117.2	2	234.5	0	0.235	0	0	0
REMOTE VALVE	0	↓	2.0	4.2	8.2	2	16.5	0	0.017	0	0	0
HAND VALVE FTRC	5	↓	1.16	3.5	4.05	2	8.11	0	0.008	0.020	0.040	0
CHECK VALVE	2	↓	15.4	3.5	54.1	2	108.1	0	0.108	0.108	0.216	0
REGULATOR	2	↓	69.5	4.7	326.6	2	653.3	0	0.653	0.653	1.306	0
SENSORS	20	↓	3.9	2.0	7.7	2	15.4	0	0.015	0	0.300	0
RELIEF VALVE	2	↓	38.6	3.5	135.1	2	270.3	0	0.270	0.270	0.540	0
HEATERS I_M	20	↓	0.04	10	0.4	2	0.8	0	0.001	0.008	0.020	0
TANK	2	8760	8.8	10	88	4	156	0	0.156	0.176	0.312	0

Table B-3

TRICKLE CHARGE – ENERGY STORAGE

(SHEET 5 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVE	0	3861	24.9	4.7	117.2	2	234.5	0	0.235	0	0	0
REMOTE VALVE	0		2.0	4.2	8.2	2	16.5	0	0.017	0	0	0
HAND VALVE FTRC	6		1.16	3.5	4.05	2	8.11	0	0.008	0.024	0.04	0
CHECK VALVE	3		15.4	3.5	54.1	2	108.1	0	0.108	0.162	0.324	0
REGULATOR	3		69.5	4.7	326.6	2	653.3	0	0.653	0.980	1.96	0
SENSORS	30		3.9	2.0	7.7	2	15.4	0	0.015	0	0.45	0
RELIEF VALVE	3		38.6	3.5	135.1	2	270.3	0	0.270	0.405	0.810	0
HEATERS	30		0.04	10	0.4	2	0.8	0	0.001	0.012	0.030	0
TANK	3	8760	8.8	10	88	4	156	0	0.156	0.264	0.468	0

Table B-3

TRICKLE CHARGE - ENERGY TRANSPORT

(SHEET 6 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	3	3861	24.9	4.7	117.2	2	234.5	0	0.235	0.352	0.705	0
REMOTE VALVES	22.		2.0	4.2	8.2	2	16.5	0	0.017	0.180	0.374	0
CHECK VALVES	1		15.4	3.5	54.1	2	108.1	0	0.108	0.054	0.108	0
HAND VALVES FTRO	47		1.16	3.5	4.05	2	8.1	0	0.008	0.190	0.376	0
HAND VALVES FTRC	0		0.39	3.5	1.4	2	2.7	0	0.003	0	0	0
PUMPS	3		116.8	9.7	1123.6	4	4494.2	0	4.494	3.371	13.48	0
SENSORS	6		3.9	2.0	7.7	2	15.44	0	0.015	0.046	0.090	0
H.E.	3		46.3	3.5	162.2	2	324.3	68	68.324	0.487	204.97	15.6
MIXER TANK	1		3.9	10	39.0	4	156.0	0	0.156	0.039	0.156	0
HEATERS	1		39	20	780	4	3120	0	3.12	0.780	3.12	0

Table B-3

AXIAL TURBINE POWER CONVERSION

COMPONENT	NO.	OP TIME	FAILURES/YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	5	3504	22.6	4.7	106.4	2	212.8	0	0.21	0.532	1.05	0
REMOTE VALVES	1		1.9	4.2	7.8	2	15.6	0	0.016	0.008	0.016	0
CHECK VALVES	4		14.0	3.5	49.1	2	98.1	0	0.098	0.196	0.39	.0
HAND VALVES FTRO	20		0.4	3.5	1.2	2	2.4	0	0.002	0.024	0.040	0
HAND VALVES FTRC	3		1.1	3.5	3.7	2	7.4	0	0.007	0.011	0.021	0
FILTER	2		38.5	3.5	134.9	2	269.8	20	20.270	0.270	40.54	0
SENSORS	8		3.5	2	7.0	2	14.0	0	0.014	0.056	0.112	0
PUMPS	3		106.1	9.7	1019.7	4	4078.7	0	4.078	3.059	12.23	0
HE	1		6.3	10	63.1	4	252.4	68	68.252	0.063	68.252	15.6
GENERATOR	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
CONDENSOR	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
COOLING TOWER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
TURBINE	1		357.4	40	14296	5	71482	520	591.482	14.296	591.482	104
GENERATOR	1		280.3	40	11212	5	56064	320	376.064	11.212	376.064	76

Table B-3

RADIAL TURBINE (288°C) POWER CONVERSION

(SHEET 8 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	6	3504	22.6	4.7	106.4	2	212.8	0	0.21	0.638	1.26	0
REMOTE VALVES	1	↓	1.9	4.2	7.8	2	15.6	0	0.016	0.008	0.016	0
CHECK VALVES	5		14.0	3.5	49.1	2	98.1	0	0.098	0.246	0.49	0
HAND VALVES FTRO	25		0.4	3.5	1.2	2	2.4	0	0.002	0.030	0.050	0
HAND VALVES FTRC	4		1.1	3.5	3.7	2	7.4	0	0.007	0.015	0.028	0
FILTER	2		38.5	3.5	134.9	2	269.8	20	20.270	0.270	40.54	0
SENSORS	9		3.5	2	7.0	2	14.0	0	0.014	0.063	0.126	0
PUMPS	3		106.1	9.7	1019.7	4	4078.7	0	4.078	3.059	12.23	0
HE	2		6.3	10	63.1	4	252.4	68	68.252	0.126	136.50	15.6
GENERATOR	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
CONDENSER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
COOLING TOWER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
TURBINE	1		357.4	40	14296	5	71482	520	591.482	14.296	591.482	104
GENERATOR	1	↓	280.3	40	11212	5	56064	320	376.064	11.212	376.064	76

Table B-3

RADIAL TURBINE (371°C) POWER CONVERSION

(SHEET 9 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/ YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	7	3504	22.6	4.7	106.4	2	212.8	0	0.21	0.745	1.47	0
REMOTE VALVES	1	↓	1.9	4.2	7.8	2	15.6	0	0.016	0.008	0.016	0
CHECK VALVES	6		14.0	3.5	49.1	2	98.1	0	0.098	0.295	0.59	0
HAND VALVES FTRO	31		0.4	3.5	1.2	2	2.4	0	0.002	0.109	0.062	0
HAND VALVES FTRC	5		1.1	3.5	3.7	2	7.4	0	0.007	0.019	0.035	0
FILTER	2		38.5	3.5	134.9	2	269.8	20	20.270	0.269	40.54	0
SENSORS	10		3.5	2	7.0	2	14.0	0	0.014	0.070	0.140	0
PUMPS	3		106.1	9.7	1019.7	4	4078.7	0	4.078	3.059	12.23	0
HE	3		6.3	10	63.1	4	252.4	68	68.252	0.189	204.76	15.6
GENERATOR	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
CONDENSER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
COOLING TOWER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
TURBINE	1		357.4	40	14296	5	71482	520	591.482	14.296	591.482	104
GENERATOR	1	↓	280.3	40	11212	5	56064	320	376.064	11.212	376.064	76

Table B-3

RADIAL TURBINE (482°C) POWER CONVERSION

(SHEET 10 OF 10)

COMPONENT	NO.	OP TIME	FAILURES/YR 10^{-3}	MTTR HRS	DOWNTIME YEAR (HRS) 10^{-3}	MEN	MMH/YR (10^{-3})	SCH. MMH YR (HRS)	COMPONENT MMH/HR (HRS)	TOTAL DOWNTIME/YR HRS	TOTAL MMH/YR HRS	PLANNED OUTAGE HRS
CONTROL VALVES	8	3504	22.6	4.7	106.4	2	212.8	0	0.21	0.851	1.68	0
REMOTE VALVES	1	↓	1.9	4.2	7.8	2	15.6	0	0.016	0.008	0.016	0
CHECK VALVES	7		14.0	3.5	49.1	2	98.1	0	0.098	0.344	0.69	0
HAND VALVES FTRO	37		0.4	3.5	1.2	2	2.4	0	0.002	0.044	0.074	0
HAND VALVES FTRC	6		1.1	3.5	3.7	2	7.4	0	0.007	0.022	0.042	0
FILTER	2		38.5	3.5	134.9	2	269.8	20	20.270	0.270	40.54	0
SENSORS	11		3.5	2	7.0	2	14.0	0	0.014	0.077	0.154	0
PUMPS	3		106.1	9.7	1019.7	4	4078.7	0	4.078	3.059	12.23	0
HE	4		6.3	10	63.1	4	252.4	68	68.252	0.252	273.0	15.6
GENERATOR	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
CONDENSER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
COOLING TOWER	1		3.5	10	35.0	4	140.2	68	68.140	0.035	68.14	15.6
TURBINE	1		357.4	40	14296	5	71482	520	591.482	14.296	591.482	104
GENERATOR	1	↓	280.3	40	11212	5	56064	320	376.064	11.212	376.064	76

B.3 TOTAL SYSTEM AVAILABILITY/MAINTENANCE ESTIMATES

The estimates of the total system availability and maintenance requirements will depend on the specific combinations of subsystems that are selected. As an example, the combination of a dual tank energy storage system with a radial turbine/900°F power conversion subsystem would give the results of Table B-4.

It should be pointed out that results of this availability and maintenance analysis are applicable to a mature system only; a system that is mature in the design sense (beyond the prototype phase) and a system that has operated for some time to get beyond the infant mortality phase. In order for a new system (the first of a kind) to achieve this level of availability at the beginning of its operational life, the components and subsystems must be subjected to rigorous testing to assure that they will operate as designed when they are placed in the total system configuration, high quality components must be used and sufficient burn-in testing should be accomplished on components to assure that no infant mortality failures will occur.

Table B-4

AVAILABILITY ANALYSIS RESULTS

	Collector	Power Conversion	Energy Transport	Energy Storage	Master Control	System
Failures/yr	0.38	1.41	0.56	0.35	0	2.70
Forced Outage/hrs/yr	2.95	30.54	4.12	1.24	0	38.85
Planned Outage, hrs/yr	15.60	104.00	15.60	0	0	104.00*
Total Outage, hrs/yr	18.55	134.54	19.72	1.24	0	142.8
Forced Outage Rate, %	0.0764	0.8716	0.1067	0.0321	0	1.086
Planned Outage Rate, %	0.4040	2.9680	0.4040	0	0	2.9680*
Total Outage Rate, %	0.4804	3.8396	0.5107	0.0321	0	4.0540
Operating Availability, %	99.52	96.26	99.49	99.97	100.00	95.95
CMTBF** (hrs)	10,160	2,485	6,894	11,031	-	1,358
CMTTR*** (hr)	7.76	21.66	7.36	3.54	-	14.39
Corrective MMH/yr****	289	144	14	3	4	451
Preventive MMH/yr	548	1,356	204	0	10	2,118
Total MMH/yr	837	1,500	218	3	14	2,569

*If all planned outages performed simultaneously
**Cumulative mean time to failure
***Cumulative mean time to recover
****Maintenance man-hours

Appendix C
MODULARITY IMPACT ON RELIABILITY AND COST

A brief study was conducted to determine the possible benefits of using a modular approach to the design of a solar power plant. In this study, the total power output of the system was held at 1 MWe, but the number of modules was varied.

The system represented by Table C-1 was used as a basis for this study. The variation in initial cost of the power conversion subsystem was determined by using the published variations in component costs as a function of component size (Reference 51). The cost of the turbine-generator was obtained by reference to manufacturers and data from Reference 52. Maintenance costs were estimated using data from Reference 53.

The results of this study showing the change in initial cost, maintenance cost, and availability as a function of the number of modules are presented in Figures C-1 and C-2. As shown in Figure C-1, the initial capital cost increases by a factor of 2.13 and the maintenance costs increase by a factor of 1.74 when the number of modules is increased from 1 to 10 (i.e., 10-100 KWe modules). Also, as shown in Table C-2 and Figure C-2, the increased number of components actually decreases the availability at the rated output of 1,000 KWe, and increased availability is not obtained until the rating is dropped to about 800 KWe.

The available component cost versus size data was used to determine the cost factor for each component to be used in the equation for modular system cost:

$$\mathrm{COST}(N) = \mathrm{COST}(1)\, N \left(\frac{1}{N}\right)^{CF}$$

Table C-1

AVAILABILITY ANALYSIS RESULTS

	Collector	Power Conversion	Energy Transport	Energy Storage	Master Control	System
Failures/yr	0.38	1.41	0.56	0.35	0	2.70
Forced Outage/hrs/yr	2.95	30.54	4.12	1.24	0	38.85
Planned Outage, hrs/yr	15.60	104.00	15.60	0	0	104.00*
Total Outage, hrs/yr	18.55	134.54	19.72	1.24	0	142.8
Forced Outage Rate, %	0.0764	0.8716	0.1067	0.0321	0	1.086
Planned Outage Rate, %	0.4040	2.9680	0.4040	0	0	2.9680*
Total Outage Rate, %	0.4804	3.8396	0.5107	0.0321	0	4.0540
Operating Availability, %	99.52	96.26	99.49	99.97	100.00	95.95
CMTBF** (hrs)	10,160	2,485	6,894	11,031	-	1,358
CMTTR*** (hr)	7.76	21.66	7.36	3.54	-	14.39
Corrective MMH/yr****	289	144	14	3	4	451
Preventive MMH/yr	548	1,356	204	0	10	2,118
Total MMH/yr	837	1,500	218	3	14	2,569

*If all planned outages performed simultaneously
**Cumulative mean time to failure
***Cumulative mean time to recover
****Maintenance man-hours

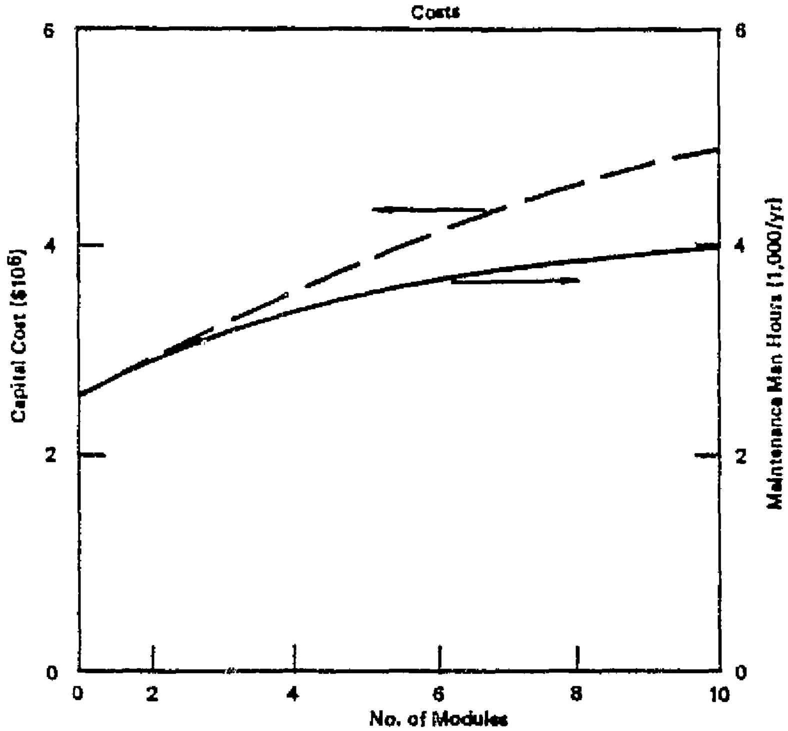

Figure C-1. Cost Impact of Modularity

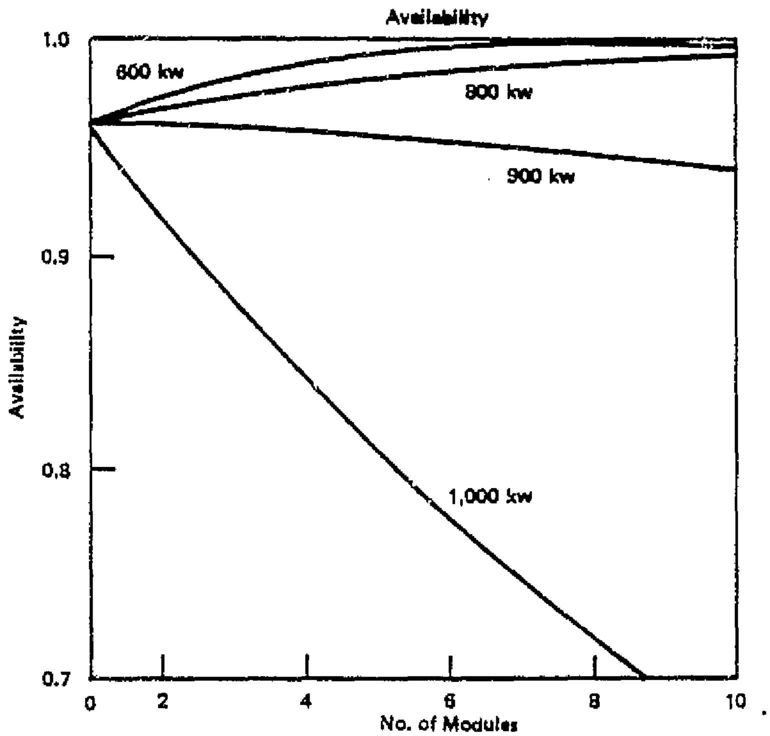

Figure C-2. Modularity Impact on Power Availability

Table C-2

SYSTEM AVAILABILITY (%)

Number of Modules	1	2	5	10
Power Level (KWe)				
At Least				
1000	95.95	91.85	81.60	66.98
900				93.97
800			98.01	98.82
700				99.34
600			99.32	99.38
500		99.34		
400			99.37	
300				
200			99.37	

where:

COST(N) = cost of N modular units

COST(1) = cost of component when only one is used (N=1)

N = number of modular units

CF = cost factors

Typical plots used to determine component cost factors are presented on Figures C-3 and C-4. Table C-3 presents the resulting cost factors used in evaluating the impact of modularity on the system cost. Figure C-5 shows the effect of modularity upon the various subsystem maintenance costs and Figure C-6 shows the modularity impact upon the capital costs of the various subsystems.

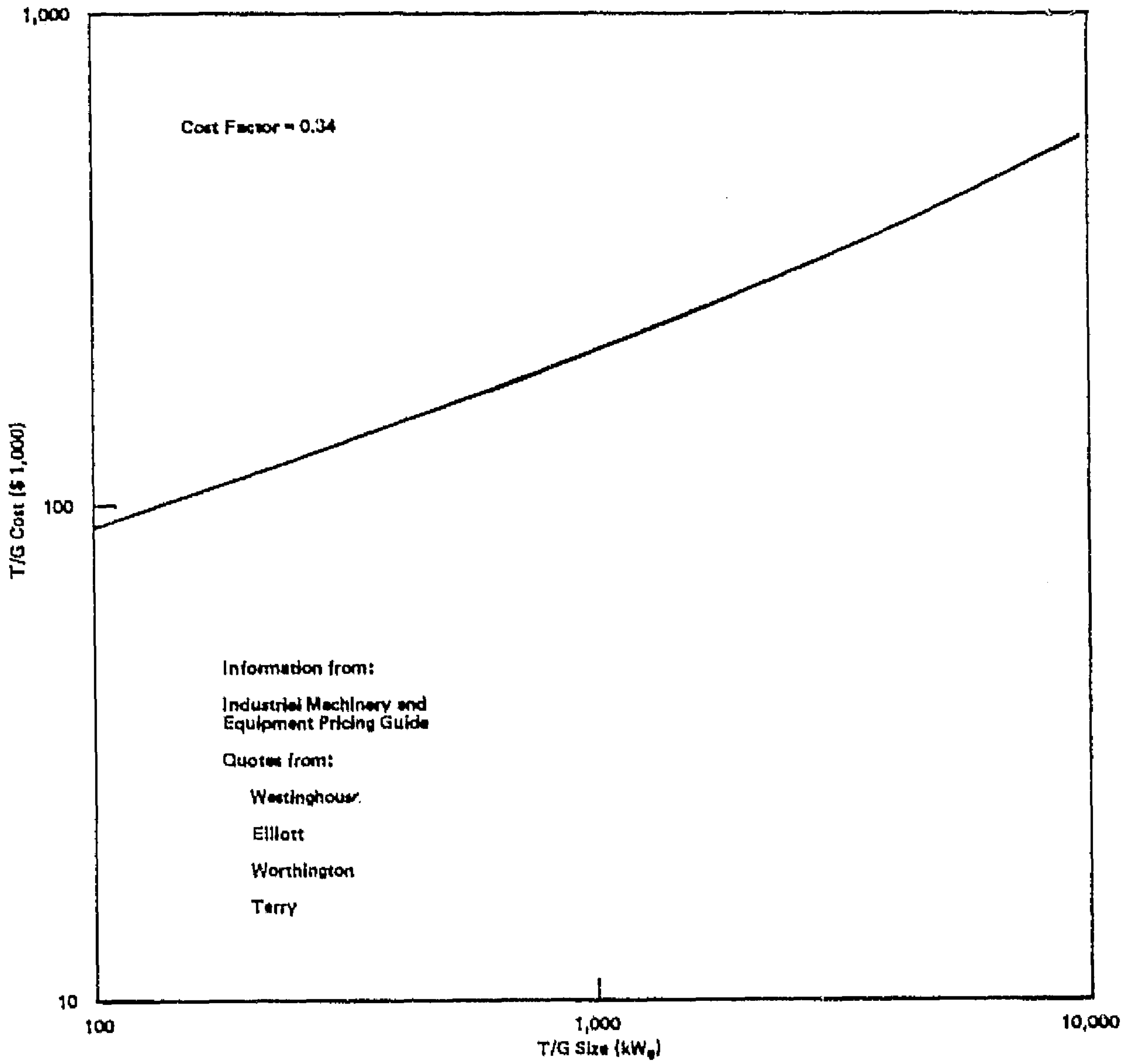

Figure C-3. Turbine-Generator Cost Data Base

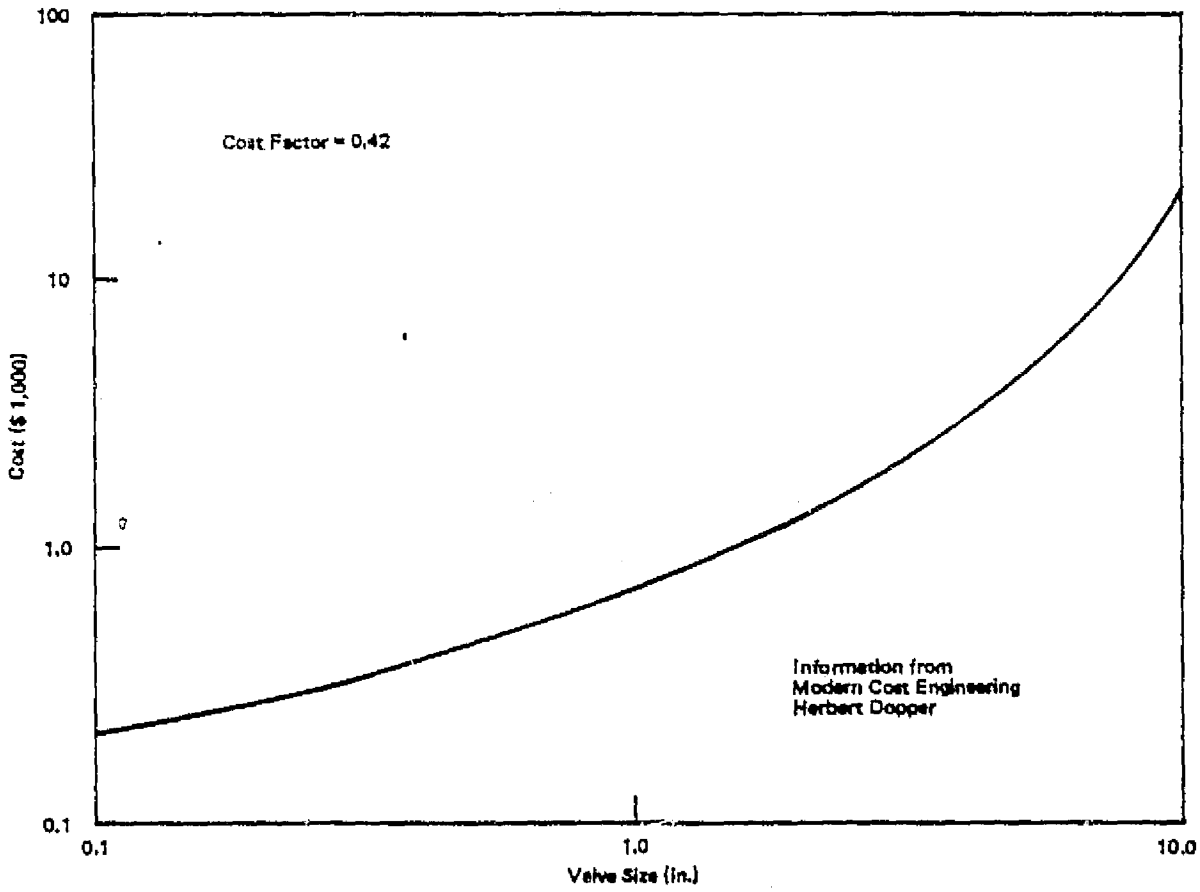

Figure C-4. Control Valve (Air Diaphragm Motor) Cost Data Base

Table C-3

COST FACTORS USED IN MODULARITY STUDY

Component	Cost Factor
Turbine/Generator	0.34
Valves	0.42
Pumps	0.30
Heat Exchangers	0.65
Storage Tanks	0.63
Condensers	0.63
Filters	0.58
Water Treatment	0.58
Cooling Tower	0.55

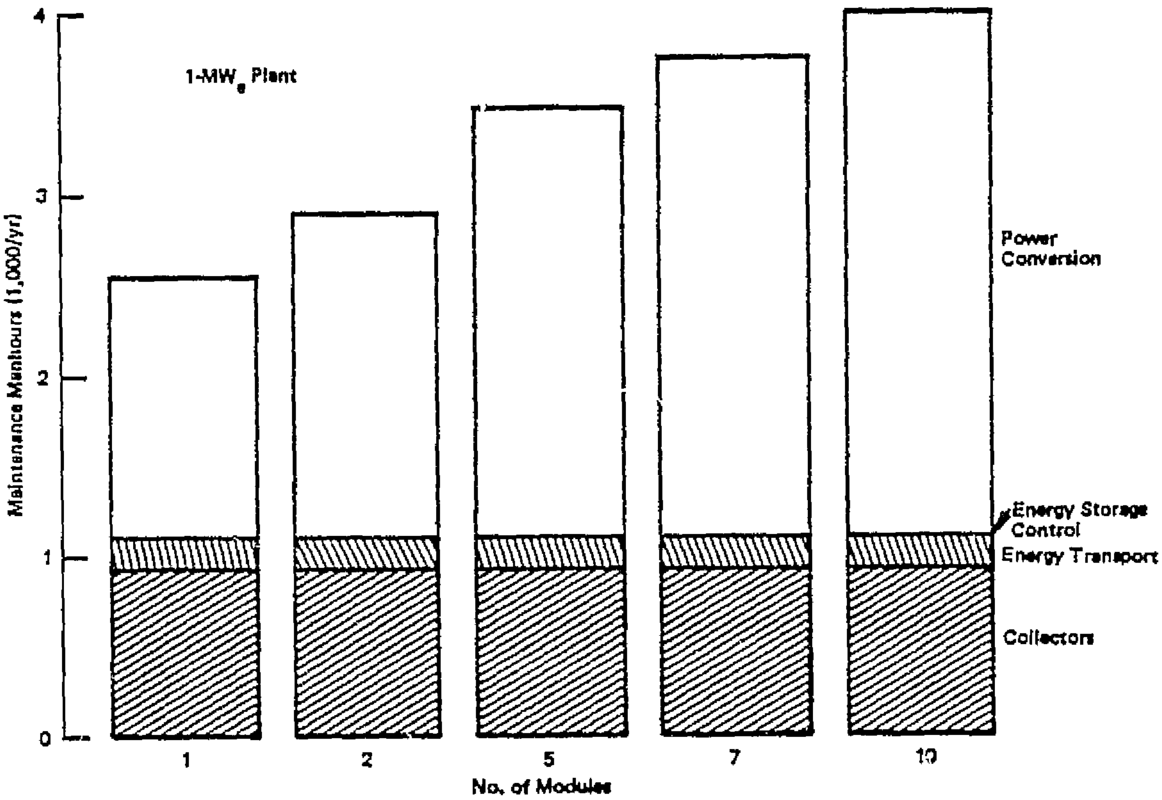

Figure C-5. System Maintenance Requirements

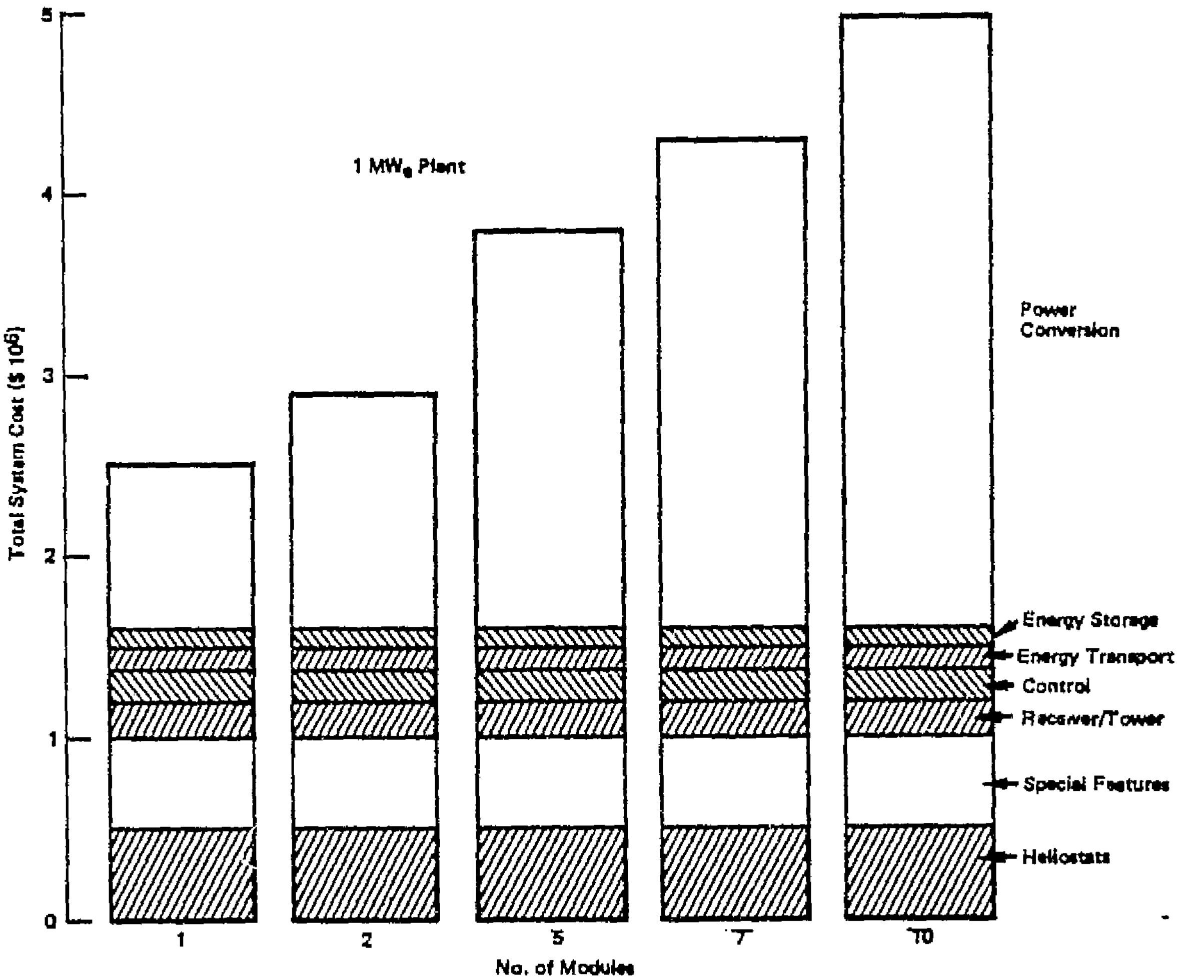

Figure C-5. Capital Costs for Modularity