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Characterization of Solar Thermal Concepts for Electricity Generation

Volume 1 - Analyses and Evaluation

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PREFACE

The research and development (R&D) described in this document was conducted within the U.S. Department of Energy's (DOE) Solar Thermal Technology Program. The goal of the Solar Thermal Technology Program is to advance the engineering and scientific understanding of solar thermal technology and to establish the technology base from which private industry can develop solar thermal power production options for introduction into the competitive energy market.

Solar thermal technology concentrates solar radiation by means of tracking mirrors or lenses onto a receiver where the solar energy is absorbed as heat and converted into electricity or incorporated into products as process heat. The two primary solar thermal technologies, central receivers and distributed receivers, employ various point- and line-focus optics to concentrate sunlight. Current central receiver systems use fields of heliostats (two-axis tracking mirrors) to focus the sun's radiant energy onto a single tower-mounted receiver. Parabolic dishes up to 17 meters in diameter track the sun in two axes and use mirrors or Fresnel lenses to focus radiant energy onto a receiver. Troughs and bowls are line-focus tracking reflectors that concentrate sunlight onto receiver tubes along their focal lines. Concentrating collector modules can be used alone or in a multi-module system. The concentrated radiant energy absorbed by the solar thermal receiver is transported to the conversion process by a circulating working fluid. Receiver temperatures range from 100°C in low-temperature troughs to over 1500°C in dish and central receiver systems.

The Solar Thermal Technology Program is directing efforts to advance and improve each system concept through the research and development of solar thermal materials, components, and subsystems, and the testing and performance evaluation of subsystems and systems. These efforts are carried out through the technical direction of DOE and its network of national laboratories who work with private industry. Together they have established a comprehensive, goal-directed program to improve performance and provide technically proven options for eventual incorporation into the nation's energy supply.

To be successful in contributing to an adequate national energy supply at reasonable cost, solar thermal energy must eventually be economically competitive with a variety of other energy sources. Components and system-level performance targets have been developed as quantitative program goals. The performance targets are used in planning research and development activities, measuring progress, assessing alternative technology options, and making optimal component developments. These targets will be pursued vigorously to ensure a successful program.

This study is aimed at providing a relative comparison of the thermodynamic and economic performance in electric applications of several concepts that have been studied and developed in the DOE solar thermal program. Since the completion of earlier systems comparison studies in the late 1970's, there have been a number of years of progress in solar thermal technology. This progress has included development of new solar components, improvements in component and system design detail, construction of working systems, and collection of operating data on the systems. This study provides an updating of the expected performance and cost of the major components and the overall system energy cost for the concepts evaluated. The projections in this study are for the late 1990's time frame, based on the capabilities of the technologies that could be expected to be achieved with further technology development.

This is the first volume of a two-volume report. This volume includes the analysis, characterization and evaluation of six concepts for solar thermal electricity generation. Volume II contains appendices.

SUMMARY

A large number of alternative solar thermal concepts for generating electricity have been proposed. Pacific Northwest Laboratory (PNL)^(a) performed this study to provide a relative comparison of the thermodynamic and economic performance of several of the concepts that have been studied and developed in the U.S. Department of Energy (DOE) Solar Thermal Technology Program for power generation applications. The projections in this study are for the late 1990's time frame, based on the capabilities of the technologies that could be expected to be achieved with further technology development. The levelized energy cost for the best solar thermal systems analyzed in this study ranged from \$0.05 to \$0.07 per kW/h (1984 dollars), which shows that the technology has considerable promise as an economical source of electricity.

A number of different central receiver, dish, and trough concepts are possible depending on the design for the concentrator, receiver, energy conversion, and other components. The concepts included in this study are:

- Molten Salt Cavity Central Receiver with Salt Storage (CR-Salt)
- Sodium External Central Receiver with Sodium Storage (CR-Na)
- Sodium External Central Receiver with Salt Storage (CR-Na/Salt)
- Water/Steam External Central Receiver with Oil/Rock Storage (CR-W/S)
- Parabolic Dish with Stirling Engine Conversion (Dish)
- Parabolic Trough with Oil/Rock Storage (Trough)

Each of the concepts were analyzed over a range of plant power ratings from 0.5 MWe to 100 MWe, and over a range of capacity factors from a no-storage case (capacity factor of roughly 0.25 to 0.3) up to intermediate-load capacity factors in the range of 0.45 to 0.6. The concepts were analyzed over a broad range of configurations to give an indication of any application niches (such

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as small no-storage plants or large, high-capacity-factor plants) where a particular concept might appear to have significant advantages relative to the other solar thermal concepts analyzed.

The annual plant energy output and levelized energy cost (LEC) were determined using the PNL code SOLSTEP. The energy analysis routines within SOLSTEP used hourly weather data to model the performance of the key plant subsystems over the course of a year (1976 climatic data for Barstow, California, was used in the analysis). The economic routines in SOLSTEP developed estimates of the capital and operating costs of the solar thermal plant, and calculated levelized busbar energy costs. For each of the plant sizes and capacity factors analyzed, SOLSTEP determined the optimum collector field and energy storage size based upon an economic optimization.

The dish system obtained the highest annual system operating efficiency of all the concepts. The annual efficiency for the dish no-storage cases was nearly 22% over a wide range of plant sizes. The central receiver concepts had annual average efficiencies for the no-storage cases ranging from 14 to 16% for the 100 MWe plant size, and from 4 to 6% for the 0.5 MWe size. The increase in the system efficiency for the central receiver concepts with increasing plant size is caused primarily by improved component efficiencies in the concentrator, receiver, and heat engine efficiency. The trough system consistently showed the lowest annual system efficiency for no-storage cases because of high receiver losses and low energy conversion efficiency. The annual efficiency of the trough no-storage designs was 9% for the 100 MWe plant and slightly below 4% for the 0.5 MWe plant.

Key annual efficiency and economic results of the study are highlighted in Table S.1 for the plant sizes and capacity factors that resulted in the lowest LEC over the analysis range. The lowest energy costs for the dish and central receiver systems ranged from \$0.05 to 0.07/kWh, and occurred for the 100 MWe plant size. The lowest LEC for the trough system was much higher than the other technologies and occurred at the 30 MWe plant size.

Within the ground rules of this study, all of the central receiver concepts and the dish system analyzed appear to show the potential to be economically competitive electricity producing technologies. In establishing energy

TABLE S.1. Summary of Key Results for Lowest Energy Cost Systems

PLANT SIZE

PLANT CAPACITY FACTOR	CR-Salt		CR-Na		CR-Na/Salt		CR-W/S		Dish		Trough	
	100 MWe 60%		100 MWe 60%		100 MWe 60%		100 MWe 24%		100 MWe 27%		30 MWe 52%	
Component	Effl- clency	Cost	Effl- clency	Cost	Effl- clency	Cost	Effl- clency	Cost	Effl- clency	Cost	Effl- clency	Cost
Concentrator	0.61	\$100/m ²	0.61	\$100/m ²	0.60	\$100/m ²	0.60	\$100/m ²	0.86	\$120/m ²	0.64	\$150/m ²
Receiver	0.87	\$30/m ²	0.83	\$19/m ²	0.83	\$19/m ²	0.79	\$35/m ²	0.82	\$21/m ²	0.60	\$14/m ²
Transport	1.0	\$14/m ²	1.0	\$14/m ²	1.0	\$14/m ²	1.0	\$9/m ²	0.96	\$17/m ²	0.93	\$48/m ²
Storage	0.99	\$12/ kWh _t	0.99	\$19/ kWh _t	0.99	\$12/ kWh _t	NA	NA	NA	NA	0.99	\$7/ kWh _t
Conversion	0.41	\$390/ kWe	0.41	\$360/ kWe	0.41	\$380/ kWe	0.38	\$300/ kWe	0.35	\$190/ kWe	0.26	\$630/ kWe
Balance of Plant	NA	\$32/m ²	NA	\$31/m ²	NA	\$31/m ²	NA	\$37/m ²	NA	\$31/m ²	NA	\$17/m ²
System	0.19	\$3500/ kWe	0.18	\$3700/ kWe	0.19	\$3400/ kWe	0.14	\$1800/ kWe	0.21	\$1300/ kWe	0.08	\$8200/ kWe
O&M	NA	\$5/m ² - Year	NA	\$5/m ² - Year	NA	\$4/m ² - Year	NA	\$5/m ² - Year	NA	\$20/m ² - Year	NA	\$6/m ² - Year
Levelized Energy Cost	NA	\$0.05/ kWe	NA	\$0.06/ kWe	NA	\$0.05 kWe	NA	\$0.07/ kWe	NA	\$0.07/ kWe	NA	\$0.15/ kWe

cost goals for solar thermal electric plants, DOE characterized the expected range of LEC from intermediate capacity factor coal plants to range from \$0.05 to 0.08/kWh in the late 1990's (DOE 1985). The results of this study show that all of the central receiver concepts can achieve energy costs within this range for the larger plant sizes. The dish system can achieve energy costs within this range over a significant range of plant sizes.

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ACRONYMS AND ABBREVIATIONS

APS - Arizona Public Service
CFR - capital recovery factor
CR - central receiver
CR-Na - sodium external central receiver with sodium storage concept
CR-Na/Salt - sodium external central receiver with salt storage concept
CR-Salt - molten salt cavity central receiver with salt storage concept
CR-W/S - water/steam external receiver with oil/rock storage concept
Dish - parabolic dish with Stirling engine conversion concept
Trough - parabolic trough with oil/rock storage
DOE - U.S. Department of Energy
DSH - parabolic dish with Stirling engine conversion concept
FCR - fixed charge rate
HVAC - heating, ventilation and air conditioning
LEC - levelized energy cost
MDAC - McDonnell Douglas Aerospace Company
NPV - net present value
O&M - operation and maintenance
PNL - Pacific Northwest Laboratory
SLT - molten salt cavity receiver with salt storage
TR - parabolic trough with oil/rock storage concept

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1.0 INTRODUCTION

Solar thermal energy systems are promising technologies for providing energy in a number of different forms. One of the alternatives being pursued by the U.S. Department of Energy (DOE) and private industry is the generation of electricity through a thermal conversion cycle. These systems are considered to have the potential to be economically viable in the long run with a variety of fossil energy resources.

A large number of alternative solar thermal concepts for generating electricity have been proposed. The fundamental design differences between the concepts appear in the concentrator subsystem (concentration ratio, type of tracking employed), the receiver subsystem (distributed vs. central receivers and receiver output temperature), the energy transport subsystem (thermal transport versus electrical transport, and the transport working fluid), and the energy conversion subsystem (thermodynamic cycle and working fluid). These concepts also differ in their expected performance and cost characteristics, and hence their feasibility for use in various electric power applications.

The goal of this study by Pacific Northwest Laboratory (PNL) is to provide a relative comparison of the thermodynamic and economic performance of several of the concepts that have been studied and developed in the DOE solar thermal program for power generation applications. The projections in this study are for the late 1990's time frame, based on the capabilities of the technologies that could be expected to be achieved with further technology development.

The DOE recently instituted major revisions to the performance and energy cost goals used to direct the development of solar thermal systems (DOE 1985, DOE 1986). Much of the information in developing the system-level goals came from comparative evaluation studies that had evaluated solar thermal technology that existed in the late 1970's time frame (Laity et al. 1980; Apley et al. 1980; Bird et al. 1981; Eicker et al. 1981; Thornton et al. 1980). Since the completion of these early studies there has been continuing research and substantial progress in solar thermal technology. This progress has included development of new solar components, improvements in existing components and system design, construction of working systems, and collection of operating

data on the test systems. This study provides an updating of the expected performance and cost of the major components and of the overall system energy cost for the concepts evaluated.

In summary, the major purposes of this study are:

- Provide a relative comparison of the solar thermal concepts that will indicate the most attractive electric applications for each concept.
- Provide an indication of the economic feasibility of the concepts in the late 1990's time frame assuming continued research and development.
- Update projections of the component and system performance for each concept, and provide the information in a format that can be used in planning research necessary to achieve the goals of the DOE Solar Thermal Technology Program.

1.1 STUDY SCOPE

Concepts analyzed in this study were selected from among many candidates. The intent of the study was to evaluate concepts that could see early commercial development (first plants constructed by the end of the 1980's) and that have the potential for significant market penetration by the turn of the century. This intent led to selecting concepts that had either been successfully demonstrated in system experiments or had been the subject of detailed design efforts.

A number of different central receiver, dish, and trough concepts are possible, depending on the design for the concentrator, receiver, energy conversion, and other components. The concepts included in this study represent four central receiver (CR) systems, one parabolic dish system, and a parabolic trough system. The basic characteristics of these technologies are summarized in Table 1.1; more detailed descriptions of the systems are found in Chapter 3.

Each of the concepts in this study were analyzed over a range of plant power ratings from 0.5 MWe to 100 MWe, and over a range of capacity factors from a no-storage case (capacity factor of roughly 0.25 to 0.3) up to intermediate load capacity factors in the range of 0.45 to 0.6. The concepts were

TABLE 1.1. Concept Design Features

	<u>CR-Salt</u>	<u>CR-Na</u>	<u>CR-Na/ Salt</u>	<u>CR-W/S</u>	<u>Dish</u>	<u>Trough</u>
Concentrator	150m ² glass- metal	150m ² glass- metal	150m ² glass- metal	150m ² glass- metal	86m ² glass- metal	78m ² glass- metal
Receiver	Cavity	External	External	External	Cavity	Glass Cover
Receiver Exit Temp °F	1050	1100	1100	950	1328	585
Transport Subsystem	Molten Salt	Sodium	Sodium	Water/ Steam	Electric	Oil
Storage Subsystem	Molten Salt	Sodium	Molten Salt	Oil/Rock	Lead Acid	Oil/Rock
Energy Conversion Subsystem	Steam Rankine	Steam Rankine	Steam Rankine	Steam Rankine	Stirling	Steam Rankine
Energy Conversion Cycle Max Temp °F	1000	1000	1000	950	1328	530

analyzed over a broad range of configurations to give an indication of any application niches (such as small no-storage plants or large, high-capacity-factor plants) where a particular concept might appear to have significant advantages relative to the other solar thermal concepts analyzed.

The concepts included in this study were judged to have reasonable potential for long-term cost-effective applications as electricity generating plants, but they do not necessarily represent the optimal configuration for each technology. Because of the wide diversity in the possible designs for each component, the results of this analysis are not necessarily indicative of the best possible cost/performance that could be achieved by solar technologies. Other design approaches, such as using a stressed membrane heliostat for the central receiver, could potentially have better cost/performance. For the dish system analyzed, other dish technologies (such as central engine dish systems) may have better economic performance at high capacity factors. Although battery energy storage systems are not being pursued by the solar

thermal program for dish systems, they were included in the scope of this study so that all concepts could be compared across capacity factors.

1.2 STUDY ANALYSIS METHODS

The analysis performed in this study has three essential features: to characterize the annual performance of the solar technologies when configured in working systems, to estimate the costs associated with constructing and operating the systems, and to determine the optimum combination of collector field size and energy storage size for power plants of different sizes and capacity factors. These activities were addressed in this analysis with the use of the computer code SOLSTEP.

SOLSTEP was developed at PNL and designed as a general solar thermal systems analysis tool (Bird 1980). It has been used in the past to analyze a wide range of solar thermal technologies for both electricity and industrial process heat applications. Key features of SOLSTEP are its ability to 1) quickly analyze plants over a wide range of size and capacity factors, 2) provide an annual simulation of the system performance using actual weather data, and 3) calculate capital, operation and maintenance (O&M), and levelized energy costs for the plant.

System performance is modeled in SOLSTEP by evaluating the performance of each component using a number of steady-state calculations over small time steps. One-hour time steps are typically used in the performance analysis for a one-year period. The plant energy output is calculated by SOLSTEP for each time step based on the insolation available, component performance characteristics during the time step, plant parasitics, and the plant operating mode. In this manner, SOLSTEP approximates the transient performance of the components, and hence of the entire plant.

Following the performance analysis of the plant, SOLSTEP performs an economic analysis. The economic portion of the code estimates the capital and operating costs of the solar thermal plant based upon user-supplied cost models. The code then uses the cost and performance assessment to calculate a levelized busbar energy cost.

In the performance and cost analysis, SOLSTEP investigates a number of combinations of collector field and energy storage combinations. These plant design points are then used by the code to determine the optimum plant configuration for any specific capacity factor.

1.3 STUDY APPROACH AND GROUND RULES

This study is aimed at evaluating specific solar thermal concepts; therefore the performance analysis concentrated on evaluating existing designs rather than developing new designs. Existing design reports were used to characterize system design and provide baseline performance estimates. In general, the design conditions for the concepts were taken from existing designs rather than being independently optimized by PNL.

Where design information was limited or did not span the range of system sizes required in the analysis, preconceptual designs and performance estimates were developed for each of the major plant components. In these cases, the design characteristics of the subsystem were developed using information from similar designs. Subsystem designs developed by PNL were felt to be reasonable approaches but were not extensively optimized.

Several ground rules were established to provide a framework for the cost and economic analysis. The analysis was based on a commercial solar thermal industry that could exist by the late 1990's. The industry is assumed to install several hundred megawatts of electric capacity per year and to have improved the solar thermal components so that technical and economic risks are similar to other energy systems. The implications of these assumptions are:

- Cost estimates are based on construction of a plant after several commercial plants of a similar design have been built, rather than on the first commercial venture.
- The annual installation rate of solar thermal capacity allows stable mass production of distributed solar thermal components.
- Plant financing is based on standard utility economics.
- Standard construction techniques will have been developed, precluding extraordinary indirect and contingency costs.

- Costs required to develop a "commercial" industry are excluded.

These general solar thermal industry characteristics provide the basis for establishing more specific cost estimating and economic analysis assumptions. Table 1.2 outlines the assumptions regarding the manufacture of distributed components.

A levelized energy cost (LEC) is calculated for each solar thermal system. The LEC serves as the summary figure of merit for making comparisons between alternative solar thermal systems. The LEC is calculated consistently with the method used in establishing technology goals for the Solar Thermal Technology Program (Williams, Dirks, and Brown 1985). Levelized energy costs were calculated in real (constant) dollar terms. Economic assumptions used in the analysis are based on recommendations for calculations to be compared to the long-term technology goals for solar thermal systems (Williams and Dirks 1985).

1.4 STUDY ORGANIZATION

Conclusions and recommendations are presented in Section 2.0. The six concepts are described in Section 3.0 followed by a discussion of the approach used to predict their performances in Section 4.0. The cost-estimating approach and economic assumptions are discussed in Section 5.0. Component performance and cost results are presented in Sections 6.0 and 7.0. Overall results and a concluding discussion are contained in Sections 8.0 and 9.0.

TABLE 1.2. Manufactured Component Ground Rules

<u>Parameter</u>	<u>Assigned Value</u>
Annual production rate	Equivalent to $2.5E6 \text{ m}^2$ of reflective aperture area per year
Real cost of capital	10%
Plant economic life	20 years
Investment tax credit	10%
Effective income tax rate	50%
Property and other taxes	1%

Supplementary discussions are contained in appendices in Volume II. The appendices include "Trough Orientation," "Levelized Energy Cost Calculations," "Plant Performance Results," "Central Receiver Optics Results," "Plant Cost and Sensitivity Analysis Data," and "Collector Manufacturing Cost Estimate Approach."

2.0 CONCLUSIONS AND RECOMMENDATIONS

This study provides a wide range of data and projections for solar thermal systems. The intent of this chapter is to provide a highlight of the most significant results of the study and provide recommendations for future research. A more detailed discussion of study results and conclusions is found in Chapter 9.

2.1 STUDY CONCLUSIONS

Within the ground rules of this study, all of the central receiver concepts and the dish system analyzed appear to show the potential to be economically competitive electricity-producing technologies. In establishing energy cost goals for solar thermal electric plants, DOE characterized the expected range of LEC from intermediate capacity factor coal plants to range from \$0.05 to 0.08/kWh in the late 1990's (DOE 1985). All of the central receiver concepts can achieve energy costs within this range for the larger plant sizes. The dish system can achieve energy costs within this range over a significant range of plant sizes.

The projections of system efficiency for the central receiver and dish systems in this study are significantly lower than the system efficiency goals established by DOE for these systems (DOE 1986). The major factors that cause the system annual performance to be lower than the goals are overall plant parasitics, turbine startup energy, turbine thermal mismatch (losses due to having an economically sized turbine, which on certain days cannot utilize the peak thermal output from the collector field), and plant availability. The first three of these factors are not accounted for explicitly in the DOE goals, and availability is not included in the DOE goals.

The trough technology analyzed in this study does not appear competitive with the central receiver concepts or the dish concept as a source of electric power. The lowest LEC achieved by a trough system in this study was \$0.15/kWh,

over twice as high as the best of the central receiver and dish results. Compared with the economic feasibility range cited above of \$0.05-0.08/kWh, the trough concept also does not appear to be a good candidate for an economic electric power producer.

Levelized energy costs decreased substantially for all concepts as plant size increased. This effect was most pronounced for central receivers and troughs; although design optimization could reduce the energy cost of small central receiver and (to a lesser extent) trough plants, it appears obvious that these concepts pay considerable economic penalties for plant sizes of 10 MWe and below. The highly modular dish concept was least affected by plant size and has the lowest LEC of all concepts at plant sizes of 10 MWe and below. However, even the dish system had a much higher LEC at the 0.5 and 2 MWe levels than at the larger plant sizes. For dish systems, the most important driver increasing LEC at the smaller sizes is balance-of-plant costs, which have relatively large fixed-cost components and therefore become a larger part of the total plant cost as plant size decreases. Reducing energy costs for very small dish systems would require innovative balance-of-system designs.

Although battery storage systems are not being pursued by the solar thermal program for dish systems, they were included in the scope of this study so that all concepts could be compared across capacity factors. Using battery storage to increase the capacity factor of the dish system proved to be very expensive because of the high storage cost for electric energy storage and low storage efficiency. Battery storage is not an economically attractive option for dish systems connected to a utility grid.

Highly automated operation appears to be an important aspect of producing low-cost solar electricity; O&M represented a significant portion of the LEC in this study, even when a significant amount of system automation was assumed. Because the number of operators required for solar thermal power plants is not expected to vary substantially with plant size, minimizing operating labor is particularly important for smaller plant sizes.

From a LEC standpoint, the differences in any of the central receiver concepts analyzed was trivial for plants without energy storage. For plants with moderate capacity factors (0.3 and higher) the CR-W/S concept is significantly

more expensive than the other three central receivers because of the impact of the storage system on the energy conversion efficiency. Because of this, the difference in LEC between the CR-W/S and the other central receivers becomes more pronounced at higher capacity factors.

2.2 RECOMMENDATIONS

Monitoring the progress of solar thermal technology and weighing the potential of new concepts should be done on an ongoing basis. Research in solar thermal technology results in innovative concepts and more refined data on the performance and cost of solar thermal components. A key to evaluating the portfolio of solar thermal research options is to have a consistent set of systems analysis projections, such as those presented in this study.

DOE performance and cost goals for solar thermal technology should be reevaluated in light of the results of this study. System performance goals should probably be decreased across the board to better consider system energy losses (such as parasitics and plant downtime) that are not adequately accounted for in the current DOE goals. For some plant components, the performance and cost goals may need to be adjusted to more ambitious levels, while other components may require more easily attainable goals.

The trough concept does not appear promising for electric power applications relative to central receivers and the dish technology. Central receivers and the dish technology analyzed all appear to be promising for electric applications and should have continued development.

The critical development item for the dish system analyzed here appears to be the receiver/power conversion unit. The combined capital cost and annualized engine maintenance cost for the dish system constitutes roughly 50 percent of the system LEC for that concept, thus the economic viability of the dish system would be substantially affected by increases in engine capital or maintenance cost. Achieving the Stirling engine cost and performance used in this study will require further engineering development coupled with significant production economies of scale. Other dish concepts may be attractive alternatives to the system analyzed in this study, particularly if Stirling engine

developments for other applications (such as automotive and space uses) do not progress toward providing an engine readily adaptable for solar applications.

System analysis studies that could be compared to the results of this study should be conducted on other promising dish and central receiver technologies. For central receivers, membrane heliostats and direct absorption receivers hold promise for energy costs lower than the costs of the concepts analyzed in this study, but the benefits are not well quantified. For dish systems, alternative concentrators, heat engines, and thermochemical transport systems should be investigated to determine if these are promising research areas to improve the economics of the dish technology.

3.0 CONCEPT DESCRIPTIONS

For the purposes of characterizing the key differences among the six concepts analyzed in this study each concept was organized into four subsystems.

- collector
- energy transport
- energy storage
- energy conversion.

All of the concepts included a collector subsystem that collects incident solar radiation and converts it to thermal energy. For all concepts except the dish system, the transport subsystem moves thermal energy from the collector subsystem to the thermal energy storage and conversion subsystems. The thermal energy storage subsystem stores excess energy collected during periods of high insolation for use when the field cannot supply adequate energy to the turbine. The energy conversion subsystem converts the collected thermal energy to electricity.

The dish concept uses a slightly different arrangement. The collector subsystem collects incident radiation and converts it to thermal energy, which is supplied directly to the Stirling engine. The transport subsystem transports electric energy from the energy conversion subsystem to a utility grid or to storage. Energy is added to the electric storage subsystem when the field output exceeds the maximum transformer input, and energy is taken out of storage when the field is supplying less than the maximum inverter input.

In this section, the general arrangement of each concept is outlined. The four subsystems are described in less detail.

3.1 MOLTEN SALT CAVITY CENTRAL RECEIVER WITH SALT STORAGE (CR-Salt)

The CR-Salt concept consists of a central receiver that uses molten salt for both the heat transfer fluid and the thermal energy storage medium.

3.1.1 General Arrangement

A large array of heliostats (mirrors) track the sun in two axes, redirecting the incident radiation onto a tower-mounted receiver. The absorbed

insolation is used to heat molten salt to 566°C (1050°F). The transport system moves the molten salt to the storage subsystem, where the hot salt is stored until needed. The hot salt is extracted from storage and used to generate steam for the energy conversion subsystem. The energy conversion subsystem consists of a three- or four-shell recirculating steam generator, a Rankine-cycle heat engine that uses water as a working fluid, and an evaporative cooling tower. A schematic of this system is shown in Figure 3.1.

3.1.2 Collector Subsystem

The same type of heliostat and support assembly was used for all central receiver concepts. The heliostat was based on the ARCO design (148.6 m²). Each heliostat has an individual tracking mechanism. The ARCO design consists of a separately adjustable mirror sections that can be used for canting.

The receiver design was based on the cavity receiver developed during the Saguaro repowering study (Weber 1983). This design uses a C-shaped cavity receiver with a vertical orientation for the absorber panels. The receiver was designed to limit peak incident solar flux to 0.60 MWt per square meter of absorber area.

FIGURE 3.1. Molten Salt Cavity Central Receiver with Salt Storage

3.1.3 Transport Subsystem

The field section of the transport subsystem takes low-temperature (288°C, 550°F) molten salt from the storage subsystem located at ground level to the tower-mounted receiver, then returns hot (566°C, 1050°F) molten salt to the storage subsystem. The storage discharge section of the transport system sends hot salt to the steam generator and returns the cooled salt to the storage subsystem.

The field transport system consists of the receiver supply pump, supply piping (riser), and return piping (downcomer). Appropriate valves and expansion joints have been included to provide for thermal expansion during startup and operation. The supply piping is carbon steel piping with fiberglass insulation. The return piping is stainless steel with calcium silicate insulation. The receiver supply pump is a carbon steel centrifugal pump with an electric motor for the driver. All valves are either carbon steel or stainless steel with an appropriate pressure rating. The transport system does not include any elevation head recovery. Elevation head is dissipated across a pressure-reducing valve. This process results in the conversion of work to heat, which slightly increases the temperature of the heat transfer fluid. Heat tracing is included for freeze protection.

3.1.4 Storage Subsystem

The storage subsystem stores thermal energy as sensible heat, with molten salt acting as the storage medium. The subsystem consists of a "hot" tank for storing the hot molten salt at 566°C (1050°F), a "cold" tank for storing salt at 288°C (550°), and a salt treatment and makeup system. A charging loop is not required because the same fluid is used for transport and storage. During periods with available insolation, molten salt is pumped from the cold tank to the receiver, where it is heated. The hot molten salt is returned to the hot storage tank. Storage is discharged by pumping molten salt from the hot tank through the discharge piping to the steam generator where thermal energy is transferred from the molten salt to the water/steam. The low-temperature molten salt is then returned to the cold tank.

3.1.5 Energy Conversion Subsystem

The energy conversion subsystem takes thermal energy supplied from the storage subsystem and converts it to electrical energy. This subsystem consists of the steam generator, a Rankine-cycle heat engine, an electric generator, and an evaporative cooling tower.

The concept uses a three- or four-shell recirculating steam generator. The steam generator, which consists of preheating, evaporating, and superheating heat exchangers, is used to generate steam. The 100 MW design also includes a reheater. Although a detailed design optimization was beyond the scope of this study, a literature review suggested the design conditions in Table 3.1.

The Rankine-cycle heat engine is a conventional steam power cycle. Although a detailed design of the Rankine-cycle heat engine was beyond the scope of this study, a literature review suggested that the cycle conditions in Table 3.1 cover the range of power levels. Accessories include feedwater heaters, feedwater booster pump, feedwater pump, lubrication systems, steam seal system, and controls.

The generator is a three-phase, four-pole synchronous unit. The generator is assumed to be air cooled and equipped with a brushless exciter.

Waste heat from the Rankine cycle is rejected to the atmosphere in a conventional evaporative cooling tower. The heat rejection unit includes the circulating water piping, circulating water pumps, and a cooling tower.

TABLE 3.1. Design Conditions for Energy Conversion Subsystem of a CR-Salt System

<u>Power Level</u>	<u>Reheat</u>	<u>Main Steam Temperature, °C (°F)</u>	<u>Reheat Steam Temperature, °C (°F)</u>	<u>Turbine Inlet Pressure, MPa (Psia)</u>
100	yes	538 (1000°)	538 (1000)	12.4 (1800)
30	no	538 (1000)	-	10.0 (1450)
10	no	538 (1000)	-	10.0 (1450)
2	no	538 (1000)	-	10.0 (1450)
5	no	538 (1000)	-	6.89 (1000)

3.2 SODIUM EXTERNAL CENTRAL RECEIVER WITH SODIUM STORAGE (CR-Na)

The CR-Na concept consists of a central receiver that uses sodium for both the heat transfer fluid and the thermal energy storage medium.

3.2.1 General Arrangement

This design consists of a large array of heliostats (mirrors) that track the sun in two axes, redirecting the incident radiation onto a tower-mounted external receiver. The absorbed insolation is used to heat liquid sodium to 593°C (1100°F). The transport system moves the sodium to the storage subsystem, where the hot sodium is stored until needed. The hot sodium is extracted from storage and used to generate steam for the energy conversion subsystem. The energy conversion subsystem includes a three- or four-shell recirculating steam generator, a Rankine cycle heat engine, and an evaporative cooling tower. A schematic of this system is shown in Figure 3.2.

3.2.2 Collector Subsystem

The CR-Na concept uses the ARCO style heliostat (148.6 m²) design. The sodium design uses an external receiver with a surround field. Because of the

FIGURE 3.2. Sodium External Central Receiver with Sodium Storage

excellent heat transfer properties of sodium, the receivers were designed with an allowable peak incident flux of 1.2 MWt/m^2 .

3.2.3 Transport Subsystem

The field transport system transports low-temperature sodium (288°C (550°F)) from the storage subsystem located at ground level to the tower-mounted receiver. The transport subsystem returns hot 593°C (1100°F) sodium to the storage subsystem.

The field transport system consists of the receiver supply pump, supply piping (riser), and return piping (downcomer). Appropriate valves and expansion joints have been included to provide for thermal expansion during startup and operation. The supply piping is carbon steel piping with fiberglass insulation. The return piping is stainless steel with calcium silicate insulation. The receiver supply pump is a carbon steel centrifugal pump with an electric motor for the driver. All valves are either carbon or stainless steel with an appropriate pressure rating. The transport system does not include any elevation head recovery. Elevation head is dissipated across a pressure-reducing valve. This action results in the conversion of work to heat, which slightly increases the temperature of the heat transfer fluid. Heat tracing is included for freeze protection.

3.2.4 Storage Subsystem

The storage subsystem stores thermal energy as sensible heat with sodium as the storage medium. The subsystem consists of a hot tank for storing hot sodium at 593°C (1100°F), and a cold tank for storing sodium at 288°C (550°F). A charging loop is not required because the same fluid is used for both transport and storage. During periods of available insolation, sodium is pumped from the cold tank to the receiver, where it is heated. Hot sodium is then returned to the hot tank. Storage is discharged by pumping sodium from the hot tank through the discharge piping to the steam generator, where thermal energy is transferred from the sodium to the water/steam. The low-temperature sodium is then returned to the cold tank.

3.2.5 Energy Conversion Subsystem

The energy conversion subsystem takes thermal energy from the storage subsystem and converts it to electrical energy. This subsystem consists of the steam generator, a Rankine cycle heat engine, an electric generator, and an evaporative cooling tower. With the exception of the steam generator, which is designed to transfer energy from sodium to steam, the energy conversion subsystem is identical to the energy conversion subsystem of the CR-Salt concept described in Section 3.1.5.

3.3 SODIUM EXTERNAL CENTRAL RECEIVER WITH SALT STORAGE (CR-Na/Salt)

The CR-Na/Salt concept consists of a central receiver that uses sodium as the heat transfer fluid; but unlike the CR-Na system described in Section 3.2, this concept uses molten salt as the storage medium.

3.3.1 General Arrangement

The CR-Na/Salt design consists of a large array of heliostats (mirrors) that track the sun in two axes, redirecting the incident radiation onto a tower-mounted external receiver. The absorbed insolation is used to heat liquid sodium to 593°C (1100°F). The transport system moves the sodium to a sodium/molten salt heat exchanger, where the hot sodium is used to heat molten salt. The molten salt is taken to the storage subsystem, where it is stored until needed. The hot molten salt is extracted from storage and used to generate steam for the energy conversion subsystem. The energy conversion subsystem includes a three- or four-shell recirculating steam generator, a Rankine cycle heat engine, and an evaporative cooling tower. A schematic of this system is shown in Figure 3.3.

3.3.2 Collector Subsystem

The CR-Na/Salt concept uses the ARCO style heliostat (148.6 m²) design. The sodium design uses an external receiver with a surround field. Because of the excellent heat transfer properties of sodium, the receivers were designed with an allowable peak incident flux of 1.2 MWt/m².

electric motor for the driver. All valves are either carbon steel or stainless steel with an appropriate pressure rating. The sodium piping and heat exchanger are designed to withstand the pressure resulting from the elevation head. Therefore, a pressure-reducing device is not required. Heat tracing is included for freeze protection.

3.3.4 Storage Subsystem

The storage subsystem of the CR-Na/Salt concept stores thermal energy as sensible heat, using molten salt as the storage medium. The subsystem consists of a hot tank for storing hot molten salt and a cold tank for storing sodium at 288°C (550°F). During periods of available insolation, molten salt is pumped from the cold tank through the sodium/molten salt heat exchanger, where it is heated to 566°C (1050°F) and then returned to the hot tank. Storage is discharged by pumping molten salt from the hot tank through the discharge piping to the steam generator where thermal energy is transferred from the molten salt to the water/steam. The low-temperature molten salt is then returned to the cold tank.

3.3.5 Energy Conversion Subsystem

The energy conversion subsystem takes thermal energy from the storage subsystem and converts it to electrical energy. This subsystem consists of the steam generator, a Rankine cycle heat engine, an electric generator, and an evaporative cooling tower. The details of the energy conversion subsystem are identical to the energy conversion subsystem of the CR-Salt concept described in Section 3.1.5.

3.4 WATER/STEAM EXTERNAL CENTRAL RECEIVER WITH OIL/ROCK STORAGE (CR-W/S)

The CR-W/S concept consists of a receiver that converts high-pressure water to superheated steam. The steam can be used directly in a Rankine power generating cycle or used to charge an oil/rock storage subsystem.

3.4.1 General Arrangement

The CR-W/S design consists of a large array of heliostats (mirrors) that track the sun in two axes, redirecting the incident radiation onto a tower-mounted receiver. The absorbed insolation is used to generate steam at 510°C

(950°F). The transport subsystem transports the steam to the ground level, where the steam is either used directly in the energy conversion subsystem or used to charge the storage subsystem. The storage subsystem stores thermal energy as sensible heat in a storage medium consisting of oil and rock. During periods of little or no insolation, energy can be extracted from storage by pumping hot oil from storage through a steam generator. Because of chemical stability limitations, the oil temperature must never significantly exceed 304°C (580°F). The lower steam temperature (277°C, 530°F) and pressure (385 psig) reduces the efficiency of the energy conversion subsystem and causes a 30% reduction in the plants output when operating from storage. The energy conversion subsystem consists of a Rankine cycle heat engine, which includes a dual admission turbine and an evaporative cooling tower. A schematic of this system is shown in Figure 3.4.

FIGURE 3.4. Water/Steam External Central Receiver with Oil/Rock Storage

3.4.2 Collector Subsystem

The CR-W/S concept uses the ARCO style heliostat (148.6 m^2) design. The receiver consists of an external receiver with a surround field. The selected receiver design uses the Babcock & Wilcox screen tube approach with the evaporator tubes placed in front of the superheating tubes so that they act as a screen for the superheating tubes. This design reduces the flux on the superheating tubes and thermal losses are reduced because the higher-temperature superheating tubes have a reduced "view" of ambient, resulting in lower radiation losses. Although the screen tube approach has several clear advantages, some concern has been expressed about the possibility of large variations in the incident flux on the superheater tubes (El Paso Electric Company, 1982). As a result, the Newman Unit 1 Repowering Study recommended an interlaced panel configuration, even though the average flux is 20% lower in this configuration (El Paso Electric Company 1980).

For this study, we assume that the design average incident flux of 0.225 MW/m^2 , typical of the screen tube design, is reasonable for an advanced water/steam receiver (Bechtel Group, Inc. 1985). It is assumed that either the screen tube approach is acceptable or that an alternative design can be developed that will minimize the flux constraints caused by the low heat transfer in the superheater tubes.

3.4.3 Transport Subsystem

The field transport system uses water/steam as the heat transfer fluid to move thermal energy from the receiver to either the energy conversion or storage subsystems. Water at 475°F is supplied to the receiver, where it is converted to superheated steam at 310°C (950°F) and returned to ground level.

The field transport system consists of the receiver supply pump, supply piping (riser), and return piping (downcomer). Appropriate valves and expansion joints have been included to provide for thermal expansion during startup and operation. The supply piping is carbon steel piping with fiberglass

insulation. The return piping is stainless steel with calcium silicate insulation. The receiver supply pump is a carbon steel or centrifugal pump with an electric motor for the driver. All valves are either carbon steel or stainless steel with an appropriate pressure rating.

3.4.4 Storage Subsystem

The thermal energy storage subsystem consists of the storage medium, storage medium containment, and a charging heat exchanger. The thermal energy storage subsystem is similar to that proposed by MDAC for the Barstow Power Plant (Hallet and Gervais 1977).

Thermal energy is stored as sensible heat in an oil and rock medium. The medium is contained in a tank arranged to maintain a thermocline between its hot and cool regions. River sand and pebbles were chosen as the rock component of the heat transfer medium. Caloria HT45® cannot be used successfully above 304°C (580°F) because of excessive thermal decomposition. Therefore, the selection of this oil requires that the maximum storage temperature be below 304°C (580°F).

During periods of high insolation, the thermal storage subsystem is charged by extracting steam returning from the collector subsystem. The steam is used to heat oil in the thermal storage charging loop, but the steam is desuperheated before entering the charging loop to prevent excessive temperatures in the thermal storage unit. The charging loop consists of charging oil pumps, charging piping, charging heat exchangers, and appropriate controls.

During discharge the thermal storage unit provides heat to generate steam in the energy conversion subsystem. Steam is generated in the thermal storage discharge loop, which contains a discharge oil pump, discharge piping, discharge heat exchangers, and appropriate controls.

The storage subsystem has two additional components that remove the residue of any thermal degradation of the heat transfer oil. The fluid maintenance system filters the oil to remove suspended solids, distills a side stream to remove high boiling polymeric compounds, and adds fresh makeup fluid to replace

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decomposed fluid. In addition to maintaining an oxygen-free gas above the heat transfer fluid, the ullage maintenance unit must also remove the volatile fractions of the degradation products that evaporate into the ullage space above the heat transfer fluid.

The oil and rock medium is contained in one or more carbon steel tanks, depending on the size of the thermal storage subsystem. The tanks are insulated to reduce heat loss from storage. Appropriate foundations and miscellaneous equipment are included. All piping is Schedule 40 carbon steel with calcium silicate insulation. All pumps are carbon steel centrifugal pumps with electric motor drives.

Two alternative storage media were investigated, molten salt storage and combined oil/rock and molten salt. The single-medium molten salt system was unattractive because of pinch point problems that limited the maximum storage temperature to 345°C (653°F). The economic benefit of a 45°C rise in storage temperature did not compensate for the cost of the increased volume of salt. The combined oil/rock and molten salt system was studied by Southern California Edison, and the result indicated that the combined storage could result in increased cost (Southern California Edison 1982).

3.4.5 Energy Conversion Subsystem

The energy conversion subsystem of the CR-W/S concept takes steam from either the receiver or the storage system steam generator and uses it to produce electric power. The subsystem consists of the Rankine-cycle heat engine, an electric generator, and an evaporating cooling tower.

The Rankine-cycle heat engine is a conventional steam power cycle. Although a detailed design of the Rankine-cycle heat engine was beyond the scope of this study, a literature review suggested the cycle conditions in Table 3.2 for the CR-W/S concept.

The Rankine cycle includes a dual admission turbine with admission points for main steam both from the field (1000 to 1800 psig) and from storage (385 psig). Accessories include feed water heating, feedwater booster pumps, feedwater pumps, lube systems, steam seal system, and controls. The generator

TABLE 3.2. Design Conditions for Energy Conversion
Subsystem of a CR-W/S System

<u>Power Level</u>	<u>Reheat</u>	<u>Main Steam Temperature, °C (°F)</u>	<u>Reheat Steam Temperature, °C (°F)</u>	<u>Turbine Inlet Pressure mPa (Psig)</u>
100	Yes-indirect	510 (950)	362 (683)	12.4 (1800)
30	No	510 (950)	-	10.0 (1450)
10	No	510 (950)	-	10.0 (1450)
2	No	510 (950)	-	10.0 (1450)
.5	No	510 (950)	-	6.89 (1000)
storage (all power levels)	No	277 (530)	-	2.72 (385)

is a three-phase, four-pole synchronous unit. The heat rejection unit includes the circulating water piping, circulating water pumps and cooling tower.

3.5 PARABOLIC DISH WITH STIRLING ENGINE CONVERSION (dish)

The dish concept consists of a number of separate dish type collector/receivers that provide thermal energy to a Stirling engine mounted at the focal point of each dish.

3.5.1 General Arrangement

The collector subsystem consists of an array of parabolic dishes that track the sun in two axes, redirecting the incident radiation onto individual receivers located at the focal point of each separate parabolic concentrator. The cavity type receiver absorbs the insolation on the heater tubes of a 25 kW kinematic Stirling engine. The Stirling engine will normally operate with a heater tube temperature of 720°C and drive a generator to produce electric power. The electric power is transported to storage or the utility power grid by the transport subsystem. The storage subsystem stores electric energy in a battery installation. The dish concept is illustrated schematically in Figure 3.5.

3.5.2 Concentrator Subsystem

The concentrator used in the dish concept is based on existing designs such as those proposed by Advanco and MDAC. The McDonnell Douglas Company has designed and tested an original reflector concept that is derived from its

FIGURE 3.5. Parabolic Dish with a Stirling Engine Conversion

heliostat work (Rogan 1985; Jaffe 1982). The Advanco Corporation has built and tested an entire dish system. The concentrator used in this study is based on the MDAC design, with receiver performance being based on the Advanco design.

The receiver is a cavity type receiver with an aperture diameter of 20.3 cm. The internal absorbing surfaces are formed by the heater tubes of the Stirling engine. This design uses a Stirling engine working fluid temperature of 720°C, which results in an average cavity temperature of around 755°C.

3.5.3 Transport Subsystem

The transport subsystem of the dish concept is an AC power transmission network and consists of the cables, transformers, circuit breakers, converters, and miscellaneous equipment necessary for transmitting power from numerous

small Stirling engines to the central storage facility. All transport subsystem components are commercially available and no problems would be anticipated with either the subsystem design or operation.

3.5.4 Storage Subsystem

Electric energy generated in the energy conversion subsystem can be stored for use during the night or periods of low insolation. The electric energy is stored as electrochemical energy in batteries. Several electric energy storage subsystem types were considered for this concept, each using a different type of battery. The lead acid battery was chosen for the base case because it was considered to have the best chance to be commercially available by the end of the century.

The electric energy storage subsystem consists of the batteries, AC/DC converters, and miscellaneous equipment such as the monitoring, cooling water, makeup water, and wiring systems. The AC/DC converters convert the AC power from the energy conversion subsystem to DC power, which can be stored in the batteries.

3.5.5 Energy Conversion Subsystem

The energy conversion subsystem takes thermal energy supplied from the collector subsystem and converts it to electrical energy. The energy conversion subsystem consists of the receiver, the Stirling engine, the generator, and the heat rejection unit.

The heat engine is assumed to be a 25 kW kinematic Stirling engine. The engine is water cooled with the cooling water rejecting waste heat to the atmosphere through a fan-equipped air-cooled heat exchanger. The Stirling engine was assumed to be similar to the United Stirling 25 kW engine (Nelving 1983).

Free piston Stirling engines were also considered and appeared to offer significant advantages compared to kinematic Stirling engines (Slaby 1985). However, to date these engines can not be considered to be successfully demonstrated and consequently were not included in this study.

3.6 PARABOLIC TROUGH WITH OIL/ROCK STORAGE SYSTEM (trough)

The parabolic trough system heats oil, which is subsequently used as a heat source for a Rankine cycle heat engine.

3.6.1 General Arrangement

The trough concept uses a collector segment consisting of a single-curvature parabolic concentrator that focuses solar insolation on a fixed linear receiver. The collector is rotated around one axis to track the sun's diurnal motion. The concentrator is a parabolic trough; the receiver consists of a glass-enclosed, selectively coated receiver tube. The collector field is used to heat oil (Caloria HT43®), and the hot oil can either be stored in the thermal storage subsystem or immediately used in the energy conversion subsystem. The energy conversion subsystem is a steam Rankine cycle; steam is generated in a heat exchanger/steam generator assembly that uses the hot oil as a heat source. Waste heat from the Rankine power cycle is rejected to the atmosphere in an evaporative cooling tower. A schematic of this concept is illustrated in Figure 3.6.

Two alternatives to oil were considered: molten salt and Siltherm. Molten salt was rejected because of freeze control problems in the extensive field piping system. Siltherm was rejected because of high cost. In both

FIGURE 3.6. Parabolic Trough with Oil/Rock Storage

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cases, the benefits of higher energy conversion efficiency would to some extent be offset by higher receiver thermal losses and transport system thermal losses. Current trough collectors such as the Acurex design have good efficiency at 700 to 800°F, and higher-temperature working fluids may deserve a more detailed evaluation at a later date. However, a more conventional design using a heat transfer oil such as Caloria HT-45 was selected for the base case trough system.

3.6.2 Collector Subsystem

The concentrator optical performance for the trough concept was based on test data for the Acurex trough (Cameron and Dudley 1986). Based on the results of sensitivity studies, a north-south orientation was selected.

3.6.3 Transport Subsystem

The transport subsystem includes the collector field supply and return piping and the field supply pump. The field transport system moves the 218°C (425°F) oil from the storage subsystem to the collector field, where the oil is heated to 310°C (590°F) and returned to the storage subsystem. Two types of transport system layouts were considered: a "ladder" and a "fold-over" network (Figure 3.7). A preliminary comparison of the two layouts indicated that the overall costs were approximately equal, so the "foldover" layout was selected to maintain consistency with previous studies. In all cases, carbon steel piping and fiberglass insulation were used for both supply and return piping. Expansion loops were used to compensate for thermal expansion. Pumps were assumed to be carbon steel centrifugal pumps with an electric motor for the drives. All valves are carbon steel with an appropriate pressure rating.

3.6.4 Storage Subsystem

The thermal storage subsystem consists of the storage medium, storage medium containment, a discharge pump, and piping. A charging loop is not required because the Caloria HT45® is used for both the heat transfer fluid and

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the storage fluid. With the exception of the charging loop, the thermal storage subsystem for this concept is identical to the thermal storage subsystem used in the CR-W/S concept.

3.6.5 Energy Conversion Subsystem

The energy conversion subsystem of the trough concept takes thermal energy from the storage subsystem and converts it to electrical energy. This subsystem consists of the steam generator, a Rankine cycle heat engine, an electric generator, and an evaporative cooling tower.

The concept uses a three-shell recirculating steam generator consisting of a preheater, an evaporator, and a superheater that uses hot oil to generate steam. The Rankine cycle heat engine is a conventional steam power cycle. Although a detailed design of the Rankine-cycle heat engine was beyond the scope of this study, a literature review suggested the cycle conditions in Table 3.3.

Accessories include feedwater heating, a feedwater booster pump, a feedwater pump, lubrication systems, a steam seal system and controls.

The selection of the working fluid involved consideration of working fluids other than water/steam. At a maximum temperature of 530°F, several organic working fluids are viable alternatives to water/steam. These alternatives include toluene and F-85. Organic Rankine cycles were rejected for the following reasons:

- Prototypical low-temperature (300°F) organic Rankine cycle heat engine have been tested in sizes up to 5 MW, but prototypes do not exist in the 30 MW to 100 MW range.

TABLE 3.3. Design Conditions for Energy Conversion Subsystem of a Trough System

<u>Power Level</u>	<u>Reheat</u>	<u>Main Steam Temperature, °C (°F)</u>	<u>Turbine Inlet Pressure, MPa (Psia)</u>
100 MW	No	277 (530)	2.72 (385)
30 MW	No	277 (530)	2.72 (385)
10 MW	No	277 (530)	2.72 (385)
2 MW	No	277 (530)	2.72 (385)
0.5 MW	No	277 (530)	2.72 (385)

- No large (10 to 100 MW) prototype high-temperature (500°F) organic Rankine cycle heat engines exist.
- Based on unpublished data provided by Bechtel to PNL in 1979, the efficiency of a steam Rankine cycle exceeds a toluene Rankine cycle for plant sizes greater than 5 MW and turbine inlet temperatures of 530°F.

While a toluene heat engine might be slightly more efficient than the steam cycle at 0.5 MW and 2.0 MW plant sizes, one working fluid was desired for all plants and therefore water/steam was selected. In addition, complications associated with using an unfamiliar working fluid worked against the selection of toluene.

The generator is a three-phase, four-pole synchronous unit with air cooling and is equipped with brushless exciters. Waste heat from the Rankine cycle is rejected to the atmosphere in a conventional evaporative cooling tower. The heat rejection unit includes the circulating water piping, circulating water pumps, and cooling towers.

4.0 PERFORMANCE APPROACH

The calculation of LEC for a solar thermal concept requires information on the predicted performance of the concept and the capital and operating costs of the various components. The approach used to predict the performance of the various solar thermal concepts is discussed in this section. Section 4.1 presents a definition of the five major subsystems and describes the figures of merit used to report their performance. Section 4.2 discusses the analytical approach used to model the subsystems, and Section 4.3 describes the system simulation using the SOLSTEP computer program.

4.1 DEFINITION OF SUBSYSTEMS

For the purposes of performance modeling, the solar thermal concept was divided into five major subsystems: concentrator, receiver, transport, storage, and energy conversion. The subsystems for a central receiver concept are shown in Figure 4.1. The dish and trough concepts have similar subsystems.

The concentrator subsystem includes the reflecting surfaces, which focus insolation on the receiver and the reflecting surface support components. For central receivers, the concentrator subsystem consists of the heliostats. The figure of merit used to describe the performance of the concentration subsystem is concentrator efficiency. The definition of concentrator efficiency used in this study is given by Equation 4.1.

$$\eta_{\text{concentrator}} = (\rho)(\eta_{\text{SD}})(\eta_{\text{geo}})(\eta_{\text{a}})(\eta_{\text{i}}) \quad (4.1)$$

where ρ = "out of box" reflectivity of the reflecting surfaces
 η_{SD} = efficiency multiplier to account for surface degradation
 η_{geo} = geometric efficiency, which includes cosine losses, blocking and shadowing, and changes in the optical efficiency due to variation in the incident angle of the solar beam.
 η_{a} = efficiency multiplier to account for atmospheric attenuation
 η_{i} = efficiency multiplier to account for receiver intercept factor

The receiver subsystem includes the receiver and supporting components. For central receiver concepts, the receiver subsystem includes the tower-mounted receiver and auxiliary equipment. The dish receiver subsystem includes the small receiver located at the focal point of the dish concentrator. The trough concept receiver subsystem includes the glass annulus and receiver tube located at the focal line of the trough concentrator. The figure of merit used to describe the performance of the receiver subsystem is receiver efficiency, which is defined by Equation 4.2

$$\eta_{\text{receiver}} = ((\eta_{\text{NC}} \alpha Q_I - Q_{\text{Th}}) / Q_I) \quad (4.2)$$

where α = receiver effective absorptivity (surface absorptivity effects geometric effects caused by cavity receivers, and transmittance of receiver annulus or window)

Q_I = solar energy incident on receiver

Q_{th} = thermal losses from receiver (during hours of operation)

η_{NC} = efficiency multiplier to account for night-time cooldown

Two distinct types of transport subsystems are included in this study. The central receiver and trough systems use a thermal transport subsystem, and the dish system uses an electric transport system. The thermal transport subsystem includes the pumps and piping used for moving low-temperature heat-transfer fluid from the storage subsystem to the receiver and returning high-temperature heat-transfer fluid. The subsystem also includes pumps and piping used for moving heat-transfer fluid to and from the steam generators. The electric transport subsystem includes the electric cable and components necessary for transporting electric energy from the dish heat engine generators to either battery storage or the grid. The transport figure of merit is given by Equation 4.3.

$$\text{Transport figure of merit} = \frac{Q_{\text{IN}} (\eta_{\text{TS}}) - W_{\text{PT}} / \eta_{\text{EPGS}}}{Q_{\text{IN}}} \quad (4.3)$$

where Q_{IN} = thermal energy entering transport subsystem
 η_{TS} = transport system thermal efficiency
 W_{PT} = transport system pump work
 η_{EPGS} = net thermal to electric conversion efficiency

Two distinct types of storage subsystems are included in this study. The dish system stores electric energy in a battery storage subsystem, and the central receiver and trough systems use thermal storage. The electric storage subsystem consists of the batteries, rectifier, and auxiliary equipment necessary to operate the battery storage subsystem. The thermal storage subsystem includes the storage charging heaters, thermal storage medium, storage tanks, and other auxiliary equipment. The storage figure of merit is given by Equation 4.4

$$\text{Storage figure of merit} = \frac{Q_{INS} \times \eta_{in} \times \eta_{out} - Q_{SL} - W_{PS}/\eta_{EPGS}}{Q_{INS}} \quad (4.4)$$

where Q_{INS} = energy entering the storage subsystem
 η_{in} = storage input efficiency
 η_{out} = storage output efficiency
 Q_{SL} = energy lost from storage
 W_{PS} = storage charge and discharge pump work
 η_{EPGS} = net thermal to electric conversion efficiency

The energy conversion system consists primarily of the heat engine and generator for converting thermal energy to electric energy. The Rankine energy conversion subsystems used by the central receiver and trough concepts include steam generators, piping, pumps, turbine, electric generator, condensers, and cooling towers. The Stirling cycle energy conversion subsystem used by the dish concept includes the heat engine, generator, and cooling fan. The energy conversion figure of merit is the gross efficiency given by Equation 4.5

$$\eta_{\text{gross design point}} = \eta_{\text{gross}} \times \eta_{PL} \times \eta_{AT} \quad (4.5)$$

where η_{gross} = gross cycle efficiency including generator

η_{gross} = design point gross efficiency
design point

η_{PL} = part load correction

η_{AT} = ambient temperature correction

In addition to individual subsystem performance, the overall plant parasitic power requirements are calculated and subtracted from the gross plant electric output.

4.2 SUBSYSTEM ANALYTICAL APPROACHES

The analytical approaches used in modeling the various subsystems depended on the concept and subsystem being considered. This section will provide an overview of the analytical approaches used in the study. In all cases, the analytical approaches reflected three key ground rules:

- Where possible, designs and performance results will be based on documented research and design studies.
- Original analysis will be undertaken only when sufficient information is not available in the literature or when concept-specific applications require original designs.
- In general, we trusted previous investigators to have identified optimum design condition. Extensive trade-off studies were not conducted as part of this investigation.

4.2.1 Concentrator Subsystem

The concentrator subsystem is the key design feature characterizing the various concepts. Therefore, this subsystem was investigated in some detail.

4.2.1.1 Central Receiver

The optical performance, field layout, tower height, and receiver design for all four central receiver systems were determined by DELSOL2, a computer code developed at Sandia National Laboratories. For each power level, six different system designs were developed at solar multiples ranging from 1.0 to 2.8. System designs in DELSOL2 are optimized by minimizing the system LEC.

For all of the central receiver designs in this study, the design insolation was 950 W/m^2 , and the design point was noon on the spring equinox. All designs use an ARCO type heliostat with a reflective area of 1600 ft^2 (148.64 m^2). The heliostats were assumed to be canted at the slant range, and the focal length of all the mirror panels was assumed to be equal to the slant range of the most distant heliostat. As for all of the other concepts, the reflectivity for new mirror surfaces was assumed to be 0.94. Because of the flat surface of the heliostats and the relative ease in which they can be cleaned, an average degradation factor of 0.9787 was assumed.

Receiver design flux limits varied with the receiver working fluid. For the molten salt cavity receiver, the receivers were designed with an allowable peak incident flux of 0.6 MW/m^2 . The sodium receivers, with a much larger heat transfer rate, were designed with a peak allowable incident flux of 1.2 MW/m^2 . The water/steam receiver had two flux design constraints. The peak incident flux limit was 0.6 MW/m^2 , and the average incident flux limit was 0.225 MW/m^2 .

An azimuth and elevation matrix bounding all possible sun positions for Barstow was generated by DELSOL2 for each design. Included in this matrix are all losses that vary as a function of sun position: cosine loss, shadowing, blocking, atmospheric attenuation, and spillage.

4.2.1.2 Dish Concept

The performance analysis of the dish concept was based on experimental data for the MDAC 86 m^2 glass metal dish concentrator and the Advanco Vanguard dish/Stirling system. A consistent reflectivity of 0.94 has been assumed for new mirror surfaces. This value has been used for all concepts. Reflectivity degradation can vary between concepts. We have assumed a degradation factor of 0.95 for the dish concept. Reflectivity degradation is more severe for the dish concept than for the central receiver concept because of the difficulty in cleaning the curved surface.

The cosine loss for a two-axis tracking solar collector that maintains a fixed angle with respect to the sun is a constant. This fact is accounted for through the use of the dish aperture area rather than surface area in calculating the incident flux.

A dish receiver and its support structure could shadow part of the dish's reflecting surface. However, the MDAC dish design, with its cutout for the dish/receiver support structure, has no mirror facets in the shaded area, and the MDAC design totally avoids blocking losses.

Shadowing of dishes by adjacent collectors at certain times is a function of ground cover ratio and field layout. The solar power plant is assumed to be sited in an area of low land cost so that a ground cover ratio of 0.25 can be achieved economically. With no expensive interconnects (such as piping) among dishes, such a low density appears quite feasible.

Dish spacing will be optimized by using a triangular grid, with the east-west spacing between dishes approximately 1.4 times the north-south spacing (Washom 1984b). These field design features taken together will limit shadowing losses to about 2% on an annual basis.

An examination of the shadow pattern of a dish collector indicates that adjacent dish shadowing occurs only briefly, in the early morning and late evening. Insolation is low at this time, and the flux is used to increase, or maintain, the receiver temperature. Direct insolation $>250 \text{ W/m}^2$ is required to overcome losses and for the generator to produce power (Washom 1984a). All shadowing by adjacent collectors is assumed to take place in this time frame, and no additional penalty beyond the threshold insolation need be included.

The intercept factor (fraction of reflected energy entering the receiver) is taken from vendor calculations. Data was available for the Advanco dish, which has slope errors of about 1 mrad and a pointing/tracking error of approximately 0.1° for an accumulated 2.03 mrad error (Washom 1984). The MDAC dish has a similar design and error budget, so it was assumed that the intercept factor for the Advanco receiver could be used for a system using a MDAC dish. The selected intercept factor for the Advanco receiver was 0.966.

Few data are available regarding the effects of wind on the intercept factor. The existing data indicate that the consequences of high wind speeds are negligible at least up to a windspeed of 13 m/s (29 mph), when tracking ceases and the device is stowed (Washom 1984). Therefore, wind effects on spillage were not included in the performance model.

4.2.1.3 Trough Concept

The performance analysis of the trough concept was based on experimental data for the Acurex Corporation 78 m² glass-metal trough collector. As with the two previously described concepts, a consistent reflectivity of 0.94 has been assumed for new mirror surfaces for all concepts. We have assumed a degradation factor of 0.95 for the trough concept. Reflectivity degradation is more severe for the trough concept than for the central receiver concepts because of the difficulty in cleaning the curved surfaces.

A north-south axis (east-west tracking) trough system will yield a higher annual energy output than an east-west axis (north-south tracking) trough system. Because the capital and operating costs are identical, the north-south system will have a lower LEC. Thus, the north-south system was selected as the baseline trough case and the east-west orientation was done as a sensitivity analysis (see Appendix A).

The cosine loss for a north-south axis trough can be readily calculated for any sun position. Whenever the sun's rays are not parallel to the aperture normal there will be a cosine loss; this effect reduces the size of the aperture. The effective aperture area of a north-south axis trough is equal to the product of the aperture area and the cosine of the incident angle (the angle between the aperture normal and the sun's rays). The cosine of the incident angle is calculated according to Equation 4.6 (Boes 1981).

$$\cos(i) = \sqrt{1 - \cos^2(\text{alt})\cos^2(\text{az})} \quad (4.6)$$

where i = incident angle
alt = solar altitude angle
az = solar azimuth angle

The total optical loss is increased by several effects, such as the variation of optical properties with angle of incidence, end loss, and receiver support structure shadowing and blocking. This overall loss is called the

incident angle modifier, K, and must be measured. For the Acurex collector, a regression analysis (Cameron and Dudley 1986) provides an equation relating K to the incident angle of the solar beam, A,

$$K = \cos A - 0.001553A + 3.965E-6(A)^2 \quad (4.7)$$

where A is in degrees.

The Acurex receiver consists of a steel absorber tube concentric with a glass tube enveloping it. The respective diameters are 4.18 and 5.4 cm; the reflector aperture width is 213 cm (Cameron and Dudley 1986). The solar beam that passes through the glass but misses the absorber tube is deflected so much that it misses the receiver altogether upon reflection from the concentrator (Bendt et al. 1979). The attendant receiver shadowing loss can be calculated:

$$1 - \frac{(5.4 - 3.18)}{213} = 0.9896 \quad (4.8)$$

Due to the short distance between the concentrator and receiver, the atmospheric attenuation should be nil.

A field of parabolic troughs, with the north-south rows spaced one collector width apart, can have significant row-to-row shadowing losses at low sun angles. To calculate the row-to-row shadowing, the effective angle for shadowing (as if the sun were due east or west) must be calculated. This calculation is done using Equation 4.9 (first row and row-end effect are ignored).

$$\cos(\text{eff}) = \frac{\sin(\text{az})\cos(\text{alt})}{\sqrt{\sin^2(\text{alt}) + \sin^2(\text{az})\cos^2(\text{alt})}} \quad (4.9)$$

where eff = effective angle for shadowing

alt = solar altitude angle

az = solar azimuth angle

With the north-south rows spaced one collector width apart (a 50% ground cover ratio), the percentage of the mirror area not shadowed (effective aperture area) is equal to the lesser of 1 or the value given by Equation 4.10.

$$EAAF = \min[1, 2\sin(\text{eff})] \quad (4.10)$$

where EAAF = effective aperture area factor
 eff = effective angle for shadowing

Few data are available regarding the quality of the Acurex reflector or on the intercept factor. However, a typical trough, with a concentration ratio of about 25 (defined as aperture width/ π x absorber dia.), attains an intercept factor of 0.98 at noon (Bendt et al. 1979). Given the Acurex concentration ratio of 21.4 (Cameron and Dudley 1986) and assumed good optics, this value should apply to it also. As for perturbations caused by the wind, no data are available, but any effects should be minimal.

4.2.2 Receiver Subsystem

The receiver subsystem is the second component unique to solar concepts, and the receiver designs vary widely between concepts. Therefore, the receivers were investigated in substantial detail.

4.2.2.1 Central Receivers

Three generic types of central receivers were considered in this study: molten salt, sodium, and water/steam. The analysis of each concept followed the same set of procedures. First, the receiver base case design was selected. Then designs were determined using the DELSOL2 field/receiver optimization code to meet the requirements of a specific design point (combination of field size and power level). The thermal performance of each design-point receiver was then determined by calculating thermal losses and cooldown.

The molten salt cavity receiver was based on the design of the Saguaro receiver (Weber 1983). The receiver dimensions were calculated using the DELSOL2 computer code, applied for each design point. A peak incident flux

limit of 0.60 MW per m^2 was used, and all receivers had a square aperture. Separate receiver designs were developed for each combination of field and storage size.

The thermal losses from the molten salt cavity receiver were calculated for each receiver design. Degradation losses were taken from the Saguaro design and were scaled based on aperture area. Convective losses were estimated using correlations developed by Siebers and Kraabel (1984). Nighttime cooldown was estimated by calculating the receiver mass and determining the transient temperature. Effective receiver absorptivity (including both material properties and cavity geometric effects) was estimated to be 0.98.

The sodium receiver was assumed to be an external receiver with a surround field. The receiver dimensions were calculated using the DELSOL2 computer code applied for each design point. A peak incident flux limit of 1.2 MW per m^2 was assumed. A separate design was developed for each combination of field and power level.

The thermal losses from the sodium receiver were calculated for each receiver design. Effective receiver absorptivity was taken from Solar One operations experience and is assumed to be 0.96. Convective and radiative losses were calculated by considering Solar One receiver performance and making adjustments based on absorber average flux and absorber wall temperature. Nighttime cooldown was estimated by calculating the receiver mass and determining the transient temperature.

The water/steam receiver was assumed to be an external receiver with a surround field. The receiver dimensions were calculated using the DELSOL2 computer code applied for each design point. A peak incident flux limit of 0.6 MW/ m^2 was assumed and the design was constrained to have a design average incident flux of 0.225 MW/ m^2 . A separate design was developed for each combination of field size and power level. The performance analysis of the water/steam receiver used the same method used for the sodium receiver.

4.2.2.2 Dish Receiver

The receiver used by the dish concept consists of a cavity receiver located at the focal point of the parabolic dish. The receiver was assumed to

be similar to the Advanco design. The performance of the receiver can be characterized by three measures of performance: the effective absorptance of the receiver, the effective emittance, and receiver thermal losses.

The effective absorptance of the receiver includes both the intrinsic absorptance of the receiver material and the geometric effects of the cavity. Because a cavity receiver acts similarly to a black body, the effective cavity absorptance is relatively insensitive to the receiver material absorptance. Advanco has calculated an effective receiver absorptance of 0.96 (Washom 1984b). Therefore, the vendor-calculated value of 0.96 was used. Receiver emittance was also taken from vendor calculations (Washom 1984a), where a value of 0.945 was given.

Thermal losses for the cavity receiver were estimated based on experimental data from Advanco (Washom 1984a). For a nominal 720°C engine hot side temperature, Advanco estimated total thermal losses as being 7.4 kWt. Because of the lack of reliable information, wind effects on thermal losses were not included.

4.2.2.3 Trough Receiver

The receiver used by the trough concept consists of a steel absorber tube with a glass tube surrounding it. The receiver was assumed to be similar to an Acurex design. As with the dish receiver, the trough receiver will be characterized by three measures of performance; effective absorptance, effective emittance, and receiver thermal losses.

The receiver tube is coated with black chrome, which has an absorptance of 0.94 at solar wavelengths. The emittance at longer wavelengths is 0.20 (Dudley and Workhoven 1981). The surrounding cover glass is made from borosilicate glass ("Pyrex"). With a proprietary anti-reflective surface treatment, its transmittance is 0.97 to 0.98 (Alvis 1983).

In normal operation, the inlet temperature of the fluid entering the receiver is 216°C (420°F) and the outlet temperature is 310°C (590°F), for a average receiver temperature of 263°C (505°F). Assuming an ambient temperature of 23°C, the resulting average receiver-ambient temperature difference is 240°C.

Field testing has resulted in two empirically derived equations describing total receiver thermal losses as a function of average receiver-ambient temperature difference for two insolation levels^(a)

$$\text{Loss (@950 W/m}^2\text{)} = 0.2280(dT) + 0.001781(dT)^2 \quad (4.11)$$

$$\text{Loss (@0 W/m}^2\text{)} = 0.1054(dT) + 0.001424(dT)^2 \quad (4.12)$$

Using the receiver-ambient temperature difference of 240°C, these equations result in predicted thermal losses of 157 W/m² at an insolation level of 950 W/m² and 107 W/m² at an insolation level of 0 W/m² where the thermal losses represent losses per m² of aperture area.

It was assumed that the Acurex collector would be employed at a geographical site having high direct insolation. Therefore, the higher thermal loss value will be used. Loss increase resulting from wind is not well characterized. Given the Pyrex envelope to reduce convective losses, any additional loss from wind ought to be relatively small.

4.2.3 Transport Subsystem

The transport subsystems required applying existing technology for fluid and electric power transport to a solar application. The characterization of the transport subsystems involved the development of preconceptual designs for each of the solar thermal concept. This section will describe the methods used in developing the preconceptual designs and in characterizing their performance.

4.2.3.1 Central Receiver Transport Subsystems

The requirements for the central receiver transport systems were determined by the receiver thermal output (which determines the mass flow rate) and the tower height. With the mass flow rate determined, the pipe diameter was selected by using an appropriate fluid velocity taken from previous designs.

(a) C. P. Cameron, Sandia National Laboratory, Personal Communication to M. K. Drost, Pacific Northwest Laboratory, September 16, 1985.

The insulation was selected so that thermal losses would not exceed thermal losses calculated in previous design studies. The design velocities and thermal losses are presented on Table 4.1.

The field pumps were sized to provide the required mass flow rate to the receiver, overcoming the transport piping pressure drop, receiver pressure drop, and the elevation change associated with pumping the fluid from ground level to the receiver elevation. Head recovery devices were not included, and it was assumed that the elevation head would be dissipated in a pressure reducing nozzle. In the case of the CR-Na/Salt system, the sodium is not transferred to a low-pressure thermal storage tank. Therefore, the sodium elevation head is not dissipated.

There are a number of design issues that were not resolved in the literature. Some of the issues were resolved and incorporated in the designs. Others were beyond the scope of the study, and assumptions were made in order to incorporate these issues in the designs.

TABLE 4.1. Design Velocity and Design Thermal Losses for Central Receivers

	Design Velocity, m/s		Maximum Allowable Thermal Losses, KW/linear m		Material	
	Hot Piping	Cold Piping	Hot Piping	Cold Piping	Hot Piping	Cold Piping
Molten salt ^(a)	3	3	305	174	stainless	carbon steel
Sodium ^(b)	6	4	305	174	stainless	carbon steel
Oil (storage discharge piping)	3	3	174	120	carbon steel	carbon steel
Water/steam ^(c)	3-4	18-20	305	374	stainless	carbon steel

(a) Weber 1983.

(b) Rockwell 1983.

(c) Hallet and Gervais 1977.

The choice of expansion loop or expansion joint for thermal expansion compensation affects capital cost, pump size and pumping power, and thermal losses. An analysis was done to compare the levelized capital cost of an expansion joint to the levelized capital cost of an expansion loop plus the additional cost of pumping fluid through the loop. Using the U-bend expansion loop specifications from Barnhart (1980) and the bellows expansion joint and costing information from Broyles (1985), joints were determined to be more economical than loops for all pipe sizes. This analysis was based only on cost factors, and there may be operational or safety factors involved in the choice of expansion compensation.

Thermal siphoning and uninsulated valves and anchors can significantly degrade thermal performance (Meyers 1983). However, the reference gave no indication of the cost of remedies. Thus, our piping network performance assumes that anti-siphon loops and insulated anchors are included and that the cost of these is negligible.

4.2.3.2. Dish Electric Transport Subsystem

The dish electric transport subsystem used in this study was based on an electric transport system developed for earlier assessment studies (Laity et al. 1979; Apley et al. 1980). The field is divided into modules, which are in turn combined to form the field. Each module includes power cable and one transformer. The modules are connected by power cable, which transports the electric power to a central export location. The field transport efficiency is calculated by multiplying the various component efficiencies.

4.2.3.3 Trough Transport Subsystem

The design and performance of the trough transport subsystems were determined using the ETRANS computer code. Performance of the energy transport system is characterized by the pump power required and the heat losses incurred. The pump power and daytime heat losses are calculated on a section-by-section basis via a steady-state analysis using average values for the fluid flow, ambient temperature, and collector performance. The nighttime heat loss modes also use an average ambient temperature and assume that the working fluid lies stagnant in the pipe network. Local fluid properties (section inlet values)

are used in all cases except for collector risers, downcomers, and connecting pieces, for which nominal fluid properties are used.

Pump Work

The energy required to pump the working fluid through a section for a period of t hours per day is calculated as

$$P_w = \frac{f}{2\rho_s} \left(\frac{L}{D}\right)_{eq} \int_0^t \frac{m \rho V^2}{\eta_p \eta_m} dt \quad (4.13)$$

where

P_w = pump work

f = Fanning friction factor

ρ = density of fluid

$\left(\frac{L}{D}\right)_{eq}$ = equivalent length over pipe diameter

m = mass flow rate

V = velocity of fluid

η_p = pump efficiency

η_m = motor efficiency

Important assumptions utilized in Equation (4.6) are:

- fully turbulent flow through smooth pipes
- resistance to flow of valves and fittings expressed in terms of $(L/D)_{eq}$ with values taken from Crane (1974).
- steady-state flow at average conditions.

This equation is used with each candidate pipe size for a given section to determine the corresponding pumping costs.

Implicit in this section-by-section optimization routine is the assumption that the optimal diameter for a section may be specified without regard to its location in the field. However, because each field consists of several parallel flow paths, all of which are constrained to undergo the same net pressure drop, this assumption is incorrect. Those sections located in shorter loops

can often be made smaller than the apparent optimum with no resulting increase in pump work because of the need to throttle most of the flows. During the first pass through the field, however, the amount of "extra" pressure drop available in each loop is unknown, and each section is sized according to the method described above. To take advantage of the need to equalize loop pressure drops, ETRANS makes a second pass through the field, downsizing sections where possible to reduce capital costs. A further advantage of downsizing is the simultaneous reduction in heat loss due to the smaller pipe sizes.

Heat Loss

For both daytime and nighttime heat loss calculations, the pipe is assumed to be at fluid temperature and the outside of the insulation at ambient temperature. This implies that the insulation is the only significant barrier to heat flow. The daytime heat loss calculation thus reduces to the straightforward analysis of a fluid flowing at steady state through a single-walled cylinder.

For nighttime heat loss calculations, the fluid is assumed to be stagnant. The analysis is complicated by two factors: 1) the thermal capacitance of the pipe (assumed to be at fluid temperature) must be accounted for; and 2) the thermal conductivity of the insulation may change greatly with temperature. ETRANS covers these contingencies by way of a rather complicated analysis.

4.2.4 Storage Subsystems

Two generic types of storage systems are included in this study: thermal storage and electric storage. Although a variety of thermal storage mediums were considered, the analysis method was the same for all concepts. Only one electric energy storage subsystem using lead acid batteries was considered.

4.2.4.1 Battery Storage

A review of advanced batteries identified lead acid batteries as being the most reasonable battery technology available in the time frame of this study. The design of the battery storage was based on using batteries with 3-h storage for all systems. A system with more than 3-h storage would still use batteries with 3-h storage but would use parallel sets of batteries. The 3-h batteries were selected because they can be fully charged in 6 h. A lead calcium grid

was used because this type of grid can accommodate the high charging rate. In all cases, performance estimates were taken from published sources.

Round-trip battery efficiencies vary with discharge time; the lowest was associated with batteries with 3-h storage. The selected efficiencies are:

<u>Storage Period</u>	<u>Round-Trip Efficiency</u>
3 h	50%
6 h	60%
9 h	65%
9 h	70%

The efficiencies were taken from Sandia National Laboratories (1980) and EPRI (1982) and are based on optimistic discharge rates.

All battery systems were oversized by 25% so that the last 20% of capacity will never be drained.

4.2.4.2 Thermal Storage

The analysis of all thermal storage systems (oil/rock, sodium, and molten salt) followed a similar procedure. The storage tank design was based on a cost optimization, and storage system piping was included in the transport subsystem.

The volume of the storage media was determined by the storage requirements and power rating of the design. The tank dimensions and insulation thickness were cost optimized to identify the least cost design. As part of the optimization procedure, the thermal losses from the tanks were determined. For molten salt and sodium it was assumed that each tank was isothermal. The oil/rock storage subsystem was designed to form a thermocline in the storage tank with hot oil filling the upper region of the tank and cold oil filling the lower region.

4.2.5 Energy Conversion Subsystem

Most concepts included in this study use a steam Rankine cycle for converting thermal energy to electrical energy. The analysis method used in the design of the steam generators for the steam Rankine cycle are described in Section 4.2.5.1. Section 4.2.5.2 presents the method of analysis used in

evaluating the steam cycle. Unlike the other concepts, the dish concept uses a small distributed engine located at the focus of the parabolic concentrator. Section 4.2.5.3 discussed the analysis of the dish energy conversion subsystem. For our analysis, a Stirling cycle heat engine was selected for dish applications.

4.2.5.1 Analysis of Steam Generators for Steam/Rankine Cycles

The central receiver concepts and the trough concept include a steam generator for the production of steam for use in a Rankine power cycle. All concepts use a three- or four-shell recirculating steam generator. The steam generator includes a separate preheating, evaporating, and superheating heat exchanger. A reheater is also included in the 100 MW central receiver designs.

Heat exchangers were sized by examining previous designs in order to determine a reasonable heat transfer coefficient. With the heat transfer coefficient, inlet and outlet temperatures, and required energy transfer known, the log-mean method was used to select the heat exchanger area. The log-mean method was initially used to develop a design comparable to an existing design. Normally, the log-mean method would underestimate the area of the heat exchanger. Therefore, a design margin was included to increase the calculated area. Table 4.2 summarizes the information used in the steam generator design. Heat exchanger inlet and outlet temperatures are given on the system configurations reported in Chapter 3.

TABLE 4.2. Steam Generator Design Information

Concept	Heat Transfer Coefficient (BTU/ft ² -°F-hr)				Design Margin
	Preheater	Evaporator	Superheater	Reheated	
Molten salt ^(a) Sodium/molten salt	273	292	385	200	25%
Sodium ^(b)	275	275	490.8	490.8	30%
Water steam ^(c) Trough	86	102	44.6	--	0%

(a) Based on Saguaro Design.

(b) Based on Carrisa Plains Design.

(c) Based on Barstow design.

4.2.5.2 Analysis of Steam Cycle

The design of a steam power cycle is the result of a complicated performance and cost optimization procedure that was beyond the scope of this study. Our approach was to review appropriate solar design studies and select the most reasonable steam cycle for each combination of power level and steam conditions. The performance of the steam cycle was then taken from the design study. Occasionally the performance would be adjusted for small variations in steam conditions.

The SOLSTEP computer code models the steam cycle performance as a design-point gross efficiency that is adjusted by a part-load correction factor and an ambient temperature correction factor. The design-point gross efficiency was normally taken from previous design studies. Table 4.3 reports the heat engine gross efficiencies for each concept. Steam conditions are reported in Chapter 3.

The gross efficiencies were calculated for design point conditions consisting of a turbine backpressure of 2.5" Hg and full load. As ambient temperature changes, the turbine backpressure will change, which in turn will cause the steam cycle gross efficiency to vary. The SOLMET meteorological data tapes do not report wet bulb temperature. Therefore, the ambient temperature correction is based on dry bulb temperature. Most performance information is based on wet bulb temperature, so a wet bulb temperature was calculated from a dry bulb temperature assuming a 35% relative humidity. The efficiency corrections were taken from published information (Bechtel Group 1985).

Part-load efficiency corrections were also taken from published information. The Bechtel Topical Report (1979) provided part-load corrections for small turbines (0.5 MW, 2.0 MW, 10.0 MW). The part-load correction for the 100 MW turbine was taken from part-load heat balances for a water/steam system presented in Darnell et al. 1985. The part-load correction for the 30 MW case was calculated by taking a weighted average between the 10 MW and 100 MW corrections.

TABLE 4.3. Heat Engine Design-Point Gross Efficiencies

	Size	Gross Efficiency	Design Source of Information
Molten salt, molten salt/sodium, sodium	100 MW	0.423	Solar 100 (Southern Cal Edison 1982)
	30 MW	0.385	Carrisa Plains (Rockwell 1983)
	10 MW	0.357	Bechtel Topical Report
	2 MW	0.328	Bechtel Topical Report (Laity et al. 1980)
	0.5 MW	0.316	Bechtel Topical Report
Water/Steam	100 MW	0.407	(Bechtel Group 1985)
	30 MW	0.385	Carrisa Plains (a)
	10 MW	0.354	Bechtel Topical Report
	2 MW	0.327	Bechtel Topical Report (Laity et al. 1980)
	0.5 MW	0.314	Bechtel Topical Report
4.20 Water/steam (oil/rock storage)	100 MW	0.275	Solar One conceptual designs
	30 MW	0.272	Estimated graphically (a)
	10 MW	0.264	Bechtel Topical Report
	2 MW	0.243	Bechtel Topical Report (Laity et al. 1980)
	0.5 MW	0.233	Bechtel Topical Report
Trough	100 MW	0.272	Vendor Information (b)
	30 MW	0.266	Estimated graphically (c)
	10 MW	0.264	Bechtel Topical Report
	2 MW	0.239	Bechtel Topical Report (Laity et al. 1980)
	0.5 MW	0.21	Bechtel Topical Report

(a) Estimated from a plot of gross efficiency versus power level.

(b) 6/30/80 letter from General Electric to D. R. Brown (PNL).

(c) Estimated from plot of gross efficiency versus power level.

4.2.5.3 Analysis of Stirling Engine Performance

The dish concepts include a 25 kW Stirling engine located at the focal point of each dish. The analysis procedure used for analyzing the Stirling engine was similar to that used for analyzing the steam Rankine cycle heat engine. First, a design-point gross efficiency (including generator losses but not including parasitic loads) was identified, then ambient temperature and part-load corrections were determined.

The design-point gross efficiency was based on experimental performance results for a 25 kW Stirling engine developed by United Stirling for Advanco (Washom 1984a). The design operating conditions associated with the design point efficiency were a full load and an ambient dry bulb temperature of 20°C. The Stirling engine does not use evaporative cooling. Therefore, it was not necessary to calculate a wet bulb temperature.

Part-load corrections were taken from previous systems evaluations (Laity et al. 1979), and ambient temperature corrections were taken from experimental data on the Advanco/Stirling system.

4.2.6 Parasitic Losses

Parasitic losses are explicitly considered in this analysis. Each parasitic load was identified and appropriate scaling relationships were developed. The constants in the various scaling relationships were either calculated or taken from previous studies. Parasitic losses were assumed to have five components.

Energy Conversion Subsystem Parasitic Losses - These losses consist of parasitics that scale with turbine load and include feed pumps, condensate pumps, circulating water pumps, and cooling tower fans. The constants in the scaling relationships were calculated from data for the Solar 100 design. Where appropriate, parasitics were adjusted for greater flow rates and heat rejection associated with the low-temperature trough system. Parasitic losses associated with the Stirling engine include losses through the water pump and the cooling fan. The cooling fan is assumed to operate when ambient temperature exceeds 16°C.

Field and Storage Pumps - This parasitic load includes the power required to drive the field pumps, storage charging pumps, and discharge pumps. These pumps were sized, and the part load adjusted energy consumption was included as a parasitic load.

Concentrator Tracking Power - Concentrator tracking power was scaled with field size, and the constants in the scaling relationships were calculated from tracking power estimates included in previous studies (Southern California Edison 1982; Washom 1984).

Fixed Operating Power Requirements - These parasitics account for loads that occur during plant operation but do not vary with plant load. They include the condenser vacuum pump, turbine controls, bearing cooling water, gland seal condenser, water treatment system, service water system, HVAC, and the master control system. These loads are estimated from the Solar 100 design and scaled to a plant size.

Standby Power Requirements - Standby power requirements are incurred while the plant is not operating. Standby parasitics dependent on field size include transport oil tracing, concentrator standby power, and receiver pumping power. Parasitics that are modeled as being independent of field size include receiver (for central receivers) and storage heat tracing, circulation pumps for standby steam, turbine turning gears, turbine bearing cooling water, gland seal steam, service water, cooling tower fans (for seal steam), HVAC, lighting, plant control system and any miscellaneous uses. These loads are estimated from data for the Solar 100 design.

4.3 SYSTEM SIMULATION METHODOLOGY

The systems analysis, integrated performance, and cost estimates were obtained from the engineering and economics evaluation tasks. The aim of the systems analysis was to estimate the system energy production cost that could be expected from each of the concepts.

The primary tool used in the systems analysis was a plant simulation computer model, SOLSTEP (Simulation Of Localized Solar Thermal Electric Production) developed at PNL as part of the small power systems study (Bird 1980). It is based on the framework of computer programs used in earlier solar thermal studies. Specifically, these were a set of programs developed by the Aerospace Corporation in 1975 that were subsequently consolidated and modified at the California Institute of Technology's Jet Propulsion Laboratory (Caputo 1977; Fujita et al. 1977; El Gabalawi et al. 1978).

The code simulates the operation of a solar thermal plant design on a macroscopic, subsystem basis. Concentrator, receiver, energy transport, electric power generation, and energy storage subsystems are considered in simple, yet complete models. The code accounts for collector optical performance variations with respect to the time of day and time of year. Other aspects of concentrator and receiver performance can be treated as functions of component power level, ambient air temperature, and wind speed. The power generation subsystem model accounts for the effects of ambient air temperature, partial load operation, auxiliary power demands, and engine standby and startup energy requirements. Both central and distributed generation systems may be modeled using either electric or thermal energy storage systems.

The plant designs were analyzed using the 1976 Barstow, California, solar insolation and meteorological data compiled by the Aerospace Corporation. However, the code does have the capability to use a large number of insolation data formats for numerous sites throughout the United States. The simulations are conducted on a time-step basis, performing steady-state computations at a number of discrete time steps to attain an approximate estimate of the plant's transient performance over the total analysis period.

Following the thermodynamic simulation, an economic evaluation of each plant design is performed. The costing methodology calculates capital costs for the component subsystems and equipment replacements and determines annual operation and maintenance charges. An economic analysis using a discounted cash flow methodology (Doane et al. 1976) then combines the lifetime plant costs and yearly plant electrical production to determine a levelized busbar energy cost.

An important requirement of the study was to determine plant performance over a range of capacity factors. To achieve this, the code actually analyzes a matrix of collector field sizes and storage capacities for each nominal plant design. The various combinations of field and storage yield an array of plant capacity factors and LECs. Typical results for an analysis of this type are presented in Figure 4.1. The series of adjacent curves represents the results for each specified field size. Points on each curve correspond to the array of storage capacities (in terms of hours of storage) considered. Each point on the graph depicts an energy production figure, specified in terms of a plant capacity factor, and an LEC for a particular plant configuration. At smaller field sizes, only the smaller storage capacities are represented because the analysis considers one field size at a time and calculates the performance at increasing storage capacities. When it is clear that the addition of further

FIGURE 4.1. Sample Plot of Energy Cost Versus Capacity Factor

storage capacity yields no increase in plant output, and therefore is not cost-effective, the calculation for that field size is completed and the next field size is considered, again starting at the lowest storage capacity.

The significant portion of the plot in Figure 4.1 is the broad curve across the bottom of the matrix. This envelope represents the locus of field and storage combinations that yields the minimum LEC over the range of capacity factors. These are the plant designs in which field size and storage capacity are matched to achieve least-cost performance. This curve has a number of uses: it shows how LEC varies with capacity factor for a concept; it may be used to select the least-cost field and storage combination at a specific capacity factor; or it may be used to select the overall least-cost plant design and the corresponding capacity factor and LEC.

The SOLSTEP performance simulation modeled four major component subsystems: the collector (concentrator and receiver), energy transport, energy storage, and energy conversion subsystems. Each of these subsystems was characterized by several performance parameters, which are briefly described in the rest of this subsection. Values for each parameter were developed for each concept considered.

Every concept considered used only the normal direct insolation striking the collector field. A geometric optics efficiency for the concentrator field was used to quantify the shading, blocking, and cosine losses experienced by the field. The geometric optics were considered as a variable function of the solar altitude and azimuth angles. Other parameters used to characterize the concentrator included the surface optical efficiency (reflectivity for mirrors) and an average surface degradation factor. The energy transfer efficiency from the concentrator to the receiver was quantified by two parameters. The concentrator-receiver interception factor accounted for tracking and spillage errors and can be treated as a variable function of wind speed. The second parameter was an average value for the atmospheric attenuation in the collector field. Receiver thermal performance was modeled by three parameters: absorptivity, nighttime cooldown efficiency, and a receiver thermal loss term. The thermal loss parameter can be treated as a variable function of receiver power, ambient air temperature, and wind speed. The final parameter describing the collector

field was a field efficiency correction ratio that can account for variations in the collector performance with changing field sizes.

The energy transport subsystem was characterized by an efficiency factor that varied with field size. The factor was considered as thermal or electrical efficiency, depending on the type of transport used in a particular concept.

The energy storage subsystem was modeled by several parameters. The first two were a storage input efficiency and a storage output efficiency. A third term specified a storage energy loss with time. Also, the maximum storage drawdown was limited by setting a minimum allowable storage charge. Like the parameters used to describe the transport subsystem, the storage parameters were expressed in thermal or electrical terms, depending on the type of storage used by a concept. For electric storage systems, a maximum storage charge rate was specified. In concepts using thermal storage, the maximum charge rate was set by the storage input pump size.

Several parameters were used to model the performance of the heat engines used in each of the generic concept designs. A nominal heat engine efficiency specified the thermal to electric conversion efficiency of the heat engine/generator unit. This parameter was a gross value, including generator efficiency. The nominal figure represented conversion efficiency at full load and standard ambient air temperature. It was corrected for partial load conditions and variations in ambient air temperature. For concepts using thermal storage, thermal transport, and large central heat engines, the standby and startup energy requirements and plant auxiliary loads were also characterized. Standby loads included warming requirements for storage heat exchangers and heat engine standby requirements for seal steam and maintenance of condenser vacuum. Two parameters were used to characterize engine startup: a bulk amount of energy required to warm up the heat engine and a minimum time allowed to ramp the turbine to full power. Four pumping loads were considered: field supply pump, field recirculating pump, storage input pump, and storage output pump. In each case the installed capacity for the auxiliary load was varied with plant size, field size, and storage size, as appropriate.

Other plant parasitics were explicitly considered and were divided into operating and standby parasitics. Operating parasitics included three components, parasitics that are a function of heat engine load, concentrator tracking parasitics, and fixed operating parasitic power requirements. Parasitics that depend on heat engine load include the feed pump, condensate pump, circulatory water pump, cooling tower fans, cooling water pump (Stirling engine) and cooling fan (Stirling engine). Concentrator tracking parasitics include power required to track the sun with the concentrator. This parasitic is a function of field size. Fixed operating power requirements account for parasitics that occur during plant operation but do not vary with plant load. They include the condenser vacuum pump, turbine cooling water, gland seal condenser, water treatment system, service water system, HVAC, and central control system.

Standby power requirements are incurred while the plant is not operating. Parasitics that are dependent on field size include transport heat tracing, any concentrator standby power, and receiver pumping power. Parasitics that are modeled as being independent of field size include receiver (for central receivers) and storage heat tracing, circulation pumps for standby steam, turbine turning gears, turbine bearing cooling water, gland seal steam, service water, cooling tower fan (for seal steam), HVAC, lighting, plant control system, and any miscellaneous uses.

5.0 ECONOMIC APPROACH

This section of the report discusses the approach taken for estimating component costs and calculating system LECs. Economic assumptions employed in the cost and financial analyses are also presented. The results of the component cost analysis are discussed in Chapter 7.0, with additional details tabulated in Appendix E. Levelized energy cost results are presented in Section 8.4.

5.1 COST ESTIMATING

Initial capital, annual operating and maintenance (O&M), and interim capital replacement costs were estimated for concentrator, receiver, transport, storage, energy conversion, and balance-of-plant components for each of the six systems evaluated. Emphasis was placed on developing independent and consistent estimates that would allow equitable system and component comparisons. Cost estimating models developed from design-point estimates are combined with system performance and financial characteristics to yield an estimate of LEC from SOLSTEP, the annual simulation model (see Chapter 4.0).

A similar general approach to cost estimating was employed for each of the components. The first step was to characterize the component design. This involved either specifying a design basis based on component descriptions in the literature (this approach was used for concentrators and receivers) or developing specifications of material type and quantity (this approach was generally used for the remaining components). The next steps were to acquire cost estimating data and to review and normalize them to the study ground rules. Costs were then estimated at the subcomponent level and aggregated to component totals. For example, the transport component cost represents an aggregation of individual estimates for pipe, insulation, valves, fittings, pumps, and other piping components. Several design point estimates were usually estimated to develop cost estimating models for the system simulation.

The sources of the cost data were as varied as the individual components are unique. Previous solar thermal component and system studies served as the

cost basis for those components unique to solar thermal technology (e.g. concentrators and receivers). Cost data for other components (e.g., transport and steam generators) were developed from estimating manuals and other published data sources in addition to our own cost data bases. Vendor data were acquired where gaps were left from the above sources.

Additional details concerning the cost estimating approach and the sources of data are discussed individually below for each component.

5.1.1 Concentrator Cost Approach

The concentrator is one of the key distinguishing characteristics of a solar thermal system and is usually the largest contributor to system capital cost. The importance of the concentrator was recognized by giving it the most detailed analysis of all the components.

Independent manufactured cost estimates were prepared for each concentrator analyzed. The manufacturing cost analysis began by identifying the material and manufacturing requirements for the chosen design. Direct material costs, direct labor hours, capital equipment costs, factory floor space, and factory land area were estimated for each design. Common assumptions were employed for unit costs and productivity of capital and labor. Each of the above factors was input to a manufacturing cost algorithm to estimate the factory cost of the concentrator. The collector manufacturing cost estimate approach is presented in Volume II, Appendix F.

Transportation costs were estimated by determining the number of loads required (either weight limited or volume limited) to deliver concentrator sub-assemblies from the factory to the site and multiplying by a standard assumption for delivery distance and cost per mile. Field assembly, construction, and installation charges were added to complete the estimate.

5.1.2 Receiver Cost Approach

Central receivers, trough receivers, and dish receivers each required a unique approach to cost estimating due to differences in the level of design and cost detail available, and the distributed versus centralized nature of the components. The approach taken for each receiver type is discussed below.

Several studies have prepared preliminary design and cost estimates during the 1980's for molten salt, sodium, and water/steam central receivers. Raw cost data were first adjusted to include only direct costs in 1984 dollars. Unit costs were then compared for absorber, structure, circulation, and instrumentation and control subcomponents. Cost comparisons were made between different sources using the same working fluid, and across working fluids.

Cost data were found to vary widely, even after normalizing to include only direct costs and adjusting to 1984 dollars. For example, direct unit costs for molten salt absorbers were found to vary by a factor of five for absorbers that were similar in size. Absolute costs were established for a single design point by averaging the cost estimates from several sources when there was no clear reason to choose one over the other. Relative costs (e.g., sodium versus molten salt) were established by considering the fundamental design differences between the receivers. For example, nickel alloy absorbers employed in the molten salt receiver are more expensive than the stainless steel absorbers used in the sodium receivers.

Central receiver tower cost data were extracted from many of the same reports that had data on the receiver itself. Site-specific variables such as wind speed, seismic loading, and soil bearing capacity have a dramatic effect on the design and cost of a tower as do the external dimensions of the receiver and, of course, the tower height. Thus, it was not surprising to find up to a factor of two variance in the estimated cost of a tower for a given height. Tower cost data were normalized to include only direct construction costs and adjusted to a common price year. A regression analysis of tower cost versus tower height yielded estimating equations for towers less than 300 ft and for towers 300 ft and taller.

Both the trough and the dish receivers are part of distributed systems and were assumed to be manufactured at high volume in a central facility. The trough receiver is an integral part of the trough collector, and the same estimating approach used for the concentrator was used for the receiver. In fact, both trough concentrator and receiver were assumed to be manufactured in the same facility.

The dish receiver is actually part of the power conversion unit that includes the Stirling engine and heat rejection equipment. Limited availability of design and cost data precluded the detailed manufacturing cost approach taken for all concentrators and the trough receiver. Manufacturing cost estimates that were available were first normalized to a standard price year and production level, and then the normalized estimates were averaged.

5.1.3 Transport Cost Approach

Transport costs were estimated in a similar manner for each system. In each case a preliminary design was developed for the transport component. This design identified the specific material requirements. For example, thermal designs (trough and central receiver systems) specified the material, length, diameter, and wall thickness of each piece of pipe and provided similar detail for other piping components. Electric transport designs (dish systems) specified the quantity and type of cable, switches, breakers, transformers, and other electrical equipment. Unit cost data were taken from several cost-estimating manuals and from PNL data bases developed on previous projects.

The wide range of transport cost estimates found in other reports provided an incentive for developing our own designs to the level of detail described above. Independent and consistently developed designs and cost estimates were required to differentiate the six systems, especially the four central receiver systems.

Costs were estimated for five design points, uniformly covering the range of possible field sizes for each system. SOLSTEP cost estimating models were developed from the design point estimates by regressing cost against the number of collectors (field size).

5.1.4 Storage Cost Approach

Storage costs were estimated for molten (nitrate) salt, liquid sodium, and oil-rock thermal systems and a lead-acid battery electric system. Subcomponent cost models were developed for each of the thermal storage systems. Example thermal storage subcomponents include the tank, foundation, storage media, insulation, and ancillary equipment. Cost data for equipment unique to storage systems were obtained from the reports of other solar thermal contractors.

Cost data for the more common components (e.g., tanks, insulation, and miscellaneous piping) were acquired from PNL's internal data bases and estimating manuals.

The subcomponent cost models were part of a larger model that simultaneously developed storage system designs and evaluated their performance and cost. Performance and cost estimates were developed for a range of thermal capacities at each of five design point power levels. Storage costs were regressed against storage capacity (kWht rating) at each power level to provide a series of cost models for SOLSTEP.

Battery storage system costs were aggregated from estimates of the following subcomponents: batteries, energy-related battery ancillaries, and power-related battery ancillaries. Cost data were acquired from several reports that reviewed a large number of individual battery system development efforts. Cost data were normalized to our study ground rules and assumptions, and then estimates were made for several design points. Battery storage system costs were regressed against storage capacity (kWhe) at each power level to provide SOLSTEP cost modeling equations.

5.1.5 Energy Conversion Cost Approach

Two approaches to estimating energy conversion costs were used: one for central Rankine systems (central receivers and trough) and one for the distributed Stirling system (dish). Energy conversion for the dish is an integral part of the power conversion unit, which also includes the receiver, the heat engine, and the generator. As discussed in Section 5.1.2, limited availability of design and cost information precluded a detailed manufacturing cost study of the power conversion unit. Available manufacturing cost estimates were normalized to standard assumptions for price year and annual production volume, and then the normalized estimates were averaged.

Rankine system energy conversion subcomponents include the steam generator, turbine generator, condenser, cooling tower, feedwater heating, and miscellaneous piping and ancillaries. Turbine generator costs were acquired from

vendors and published reports. The cost of the condenser, cooling tower, feed-water heaters and other miscellaneous equipment was set equal to the turbine generator cost based on trends seen in the same reports as above.

Preliminary design information (heat exchanger type, heat transfer area, operating temperatures and pressures, and construction materials) was specified for each unit of the steam generator. A unit includes a preheater, boiler, and superheater for all systems and a reheater for 100 MWe systems. In this way we were able to differentiate energy conversion costs of the Rankine systems.

5.1.6 Balance-of-Plant Cost Approach

The balance-of-plant category includes land and support equipment and facilities that are not included in the five components discussed above but are equally important to the successful operation of a solar thermal facility. Balance-of-plant was broken into the following six subcomponents in this analysis: land and site preparation, service facilities, power conditioning (switchyard), structures, master control, and spare parts. Three of these subcomponents were further broken down as shown in Table 5.1. Balance-of-plant is a direct capital cost account; indirects, contingency, and operation and maintenance costs are accounted for elsewhere.

Balance-of-plant requirements and costs were developed from data in past solar thermal system studies, other solar thermal studies, and cost estimating manuals. In general, requirements and costs were identified at each power level. Distinctions between central receiver, trough and dish systems fell out of the subcomponent by subcomponent review. Cost models for four of the balance-of-plant subcomponents scale with the plant power rating. The two

TABLE 5.1. Balance-of-Plant Sub-Subcomponents

<u>Land and Site Preparation</u>	<u>Structures</u>	<u>Service Facilities</u>
Land purchase	Control room	Fire protection
Fees and permits	Administration	Security
Surveying	Warehouse	Maintenance vehicles
Clearing and grubbing	Maintenance	Communication
Grading		Sanitary water
Ditches		Feedwater treatment
Roads		

exceptions are land and site preparation, which scales with field size, and spare parts, which is estimated as a fraction of initial capital cost.

5.1.7 Indirect and Contingency Cost Approach

Indirect and contingency costs were estimated as a fraction of the direct costs (concentrator, receiver, transport, storage, energy conversion, and balance-of-plant). The rates (25% for indirects and 10% for contingency) were established by reviewing estimated charges from several solar thermal design reports and comparing them with expected charges from construction cost estimating manuals.

5.1.8 Operation and Maintenance Cost Approach

Operation and maintenance costs were aggregated from individual estimates of operating labor, maintenance labor and materials, and overhead labor. Operating labor costs were established by specifying the number of operators and shifts required for each technology at each power level and multiplying the resultant man-hour figure by a burdened labor rate. Labor requirements and hourly charges from currently operating solar facilities as well projected requirements from past design studies were considered in developing the labor estimate. Maintenance labor and materials cost estimates were also taken from various design reports and expressed as fractions of initial capital cost for each of the direct cost components. Overhead labor (expressed as a fraction of direct labor) was based on rules-of-thumb presented in several cost estimating manuals.

The LEC approach to economic evaluation of solar thermal power plants is discussed in Volume II, Appendix B.

5.2 ECONOMIC ASSUMPTIONS

Several ground rules were established to provide a framework for the cost and economic analysis. The analysis was based on a commercial solar thermal industry that could exist by the mid to late 1990's. This future industry is described as follows:

- solar thermal systems and components are relatively mature
- a few hundred megawatts of capacity are installed each year

- technical and economic risks are similar to other energy options

A commercial solar thermal industry was assumed to have the following implications 1) cost estimates are based on construction of the nth plant of each type rather than the first commercial venture, 2) the annual installation rate of solar thermal capacity allows stable mass production of distributed solar thermal components, 3) plant financing is based on standard utility economics, 4) standard construction techniques will have been developed that preclude extraordinary indirects and contingency, and 5) developmental costs required to achieve a commercial industry are excluded.

The general solar thermal industry characteristics provide the basis for establishing more specific cost estimating and economic analysis assumptions. Tables 5.2 and 5.3 outline assumptions regarding the manufacture of distributed components and power plant economics, respectively. The economic assumptions outlined in Tables 5.2 and 5.3 correspond to assumptions in the Solar Thermal Five Year Plan (DOE 1984) for industrial (Table 5.2) and utility (Table 5.3) ownership.

TABLE 5.2. Manufactured Component Ground Rules

Parameter	Assigned Value
Annual production rate	Equivalent to $2.5 \times 10^6 \text{ m}^2$ of reflective aperture area per year
Real after tax cost of capital	10%
General inflation rate	4%
Plant economic life	20 years
Investment tax credit	10%
Effective income tax rate	50%
Property and other taxes	1%
Depreciation life	5 years (ACRS)

TABLE 5.3. Power Plant Economic Ground Rules

<u>Parameter</u>	<u>Assigned Value</u>
Price year	1984
Real after tax cost of capital	3.15%
General inflation rate	4%
Investment tax credit	10%
Effective income tax rate	50%
Plant economic life	30 years
Property and other taxes	1%
Depreciation life	10 years (ACRS)
Construction period	0.5 and 2 MWe = 1 year 10 MWe = 2 years 30 and 100 MWe = 3 years

6.0 COMPONENT PERFORMANCE RESULTS

This section discusses the performance of the various components of the six solar thermal concepts included in the study. The performance of the components was determined using the SOLSTEP computer code. The analysis procedure consisted of:

- Characterization of Design-Point Performance--The design-point performance of each component was investigated and characterized in a form that could be used as an input to SOLSTEP. The design-point performance is the performance of a component when operating at design point conditions. Design-point conditions are defined as a solar insolation of 950 W/m^2 with typical ambient conditions and steady-state operation.
- Characterization of Off-Design Operation--After design-point performance was characterized, the influence of off-design operation was evaluated and characterized in a form that could be used as input to SOLSTEP. Typically, off-design operation, such as part load operation or operation with reduced insolation, will result in a degradation in performance when compared to the design point.
- Annual Average Performance--As each of the six concepts were simulated, the annual performance of the various components was calculated in addition to overall plant performance. During a complete year, the plant is rarely operating at the design point. Consequently, annual component performance will always be less than design-point performance. The annual average performance is a better estimate of component performance than design-point performance, but its calculation requires a meteorological data base and a simulation code like SOLSTEP.

This section will report both design-point performance (SOLSTEP inputs) and annual average performance (SOLSTEP output) for each component. The impact of off-design operation is included in the calculation of annual average performance. The results are presented by concept and component. Each concept is discussed separately, and the component performance results are presented for

five components: the concentrator subsystem, the receiver subsystem, the transport subsystem, the storage subsystem, and the energy conversion subsystem. The performance of each component is described in two ways; the design-point performance is presented followed by the annual average performance of the component. Lastly, for each of the six concepts, parasitic losses are discussed.

6.1 MOLTEN SALT CAVITY CENTRAL RECEIVER WITH SALT STORAGE

The CR-Salt concept consists of a heliostat field for the concentrator subsystem, molten salt cavity receiver subsystem, molten salt transport subsystem, molten salt storage subsystem, and a steam Rankine heat engine for the energy conversion subsystem. The performance of each subsystem is discussed below.

6.1.1 Concentrator Subsystem

The concentrator subsystem includes a north field of heliostats with performance characteristics representative of the ARCO 148.6 m² heliostat.

6.1.1.1 Design-Point Performance

The primary figure of merit for characterizing concentrator performance is concentrator efficiency. Concentrator efficiency includes the impact of cosine losses, heliostat reflectivity losses, shadowing, blocking, atmospheric attenuation, and receiver intercept losses. The design-point concentrator performance was determined using DELSOL2 and was then converted to a form suitable for input into SOLSTEP. In general, the design-point performance is taken from DELSOL2 and reported as a function of solar multiple. In most cases, average annual results are taken from SOLSTEP and reported as a function of capacity factor.

The concentrator efficiency is shown in two ways. First, in Figure 6.1 concentrator efficiency is shown as a function of altitude and azimuth for a 30 MWe plant with a solar multiple of 1.0. The maximum concentrator efficiency occurs with an azimuth of 0° and an altitude of approximately 40°. Second, the concentrator efficiency is presented in Table 6.1 as a function of plant size for a solar multiple of 1.0. A review of Table 6.1 shows that concentrator

FIGURE 6.1. Concentrator Efficiency for CR-Salt System

TABLE 6.1. Concentrator Efficiency for a CR-Salt System

System Size, MWe	Solar Multiple = 1.0 Azimuth = 0° and Altitude = 40°	Average Annual	
		No Storage	0.4 Capacity Factor
0.5	0.60	0.53	0.50
2.0	0.72	0.61	0.61
10.0	0.74	0.64	0.63
30.0	0.76	0.64	0.63
100.0	0.74	0.63	0.62

efficiency is severely degraded at small plant sizes. The main cause for this is the large intercept losses associated with trying to concentrate insolation from large heliostats on a small receiver. The concentrator efficiency actually shows a peak at plant sizes of 30 MWe and declines slightly as plant size

increases to 100 MWe. This decline is caused by increased atmospheric attenuation associated with larger field sizes.

6.1.1.2 Average Annual Performance

The true measure of concentrator performance is average annual concentration efficiency. This section contains a discussion of the components of average annual concentrator performance followed by a discussion of the overall concentrator performance.

The components of average annual concentrator efficiency are shown on Table 6.2. These results are from DELSOL2 simulations and are reported for a large and small field where the large field is sized for a 100 MWe solar power plant with a solar multiple of 2.8 and the small field is sized for a 0.5 MWe solar power plant with a solar multiple of 1.0.

Heliostat reflectivity loss is a special case because it is assumed to be constant regardless of field size or the sun's position. For this study, heliostat "out of box" reflectivity is assumed to be 0.94, and the degradation factor is assumed to be 0.9787. The degradation factor is based on bi-weekly heliostat cleaning. These two factors result in an annual average reflectivity of 0.92.

A review of Table 6.2 shows that the major loss mechanisms for the larger field size are cosine losses, shadowing, and atmosphere attenuation. The major loss mechanisms for the small field size are cosine losses, blocking, and intercept losses. A comparison of the large and small field design-point

TABLE 6.2. Components of Average Annual Concentrator Efficiency for a CR-Salt System

<u>Loss Mechanism</u>	<u>Field Size</u>	
	<u>Small</u>	<u>Large</u>
Cosine Loss	0.83	0.83
Reflection Loss	0.92	0.92
Shadowing	0.97	0.95
Blocking	0.91	0.99
Atmospheric Attenuation	0.98	0.88
Intercept Losses	0.69	0.96

performance clearly shows that performance for the small field is substantially below that for the large field. The major cause of the performance difference is the intercept losses.

The annual average concentrator efficiencies are listed in Table 6.1 for the no-storage and 0.4-capacity-factor designs. The efficiency peaks between 10 and 30 MWe; at larger field sizes the atmospheric attenuation of the large north-only field replaces the receiver spillage loss as the dominant loss.

6.1.2 Receiver Subsystem

The receiver subsystem includes a molten salt receiver that is assumed to be similar to the design developed during the Saguaro Study (Weber 1983).

6.1.2.1 Design-Point Performance

The molten salt receiver is based on the Saguaro design, and the method used in determining the design-point performance is summarized in Section 3.0. Table 6.3 presents the design-point thermal efficiency of the molten salt receiver as a function of plant size for plants with a solar multiple of 1.0. The effective absorption, based on the receiver tube material, absorber coating, and geometry of the receiver cavity is 0.98 (Weber 1983). The reflection losses are included in the design-point thermal efficiency reported in Table 6.3. A review of Table 6.3 shows that receiver design point efficiency increases slightly as the plant size increases. It is difficult to draw firm conclusions about the reported trends in receiver efficiency because the receiver design characteristics were taken from DELSOL2, which optimizes the combination of concentrator and receiver. At small plant sizes, DELSOL2 could

TABLE 6.3. Design-Point Thermal Efficiency for a Molten Salt Cavity Receiver Subsystem (solar multiple = 1.0)

<u>Plant Size, MWe</u>	<u>Receiver Design Point Thermal Efficiency</u>
0.5	87.7
2.0	88.5
10.0	88.3
30.0	91.0
100.0	92.1

produce optimized designs for similar plants that showed substantial differences in specific details. In general, the improved performance of the receiver with increasing plant size is caused by a decrease in aperture area relative to absorber area.

In this study, the design-point receiver thermal losses can be modified by ambient temperature and wind speed. In the case of the molten salt receiver, losses were adjusted for ambient temperature but not for wind speed. The cavity design was assumed to limit the impact of wind speed variations.

6.1.2.2 Annual Average Results

The annual average receiver efficiencies are listed in Table 6.4 as a function of plant size for both plants with no storage and plants with a capacity factor of 0.4. As with the design-point efficiencies, average annual receiver efficiency increases with increasing plant size. Again, this is caused primarily by a decrease in aperture area relative to absorber area, which reduces losses at large plant sizes. Receiver efficiencies for the 0.4-capacity-factor case generally exceed receiver efficiencies for the no-storage case. This fact is also related to the increased field and receiver size required by the 0.4-capacity-factor case when compared to the no-storage case. As with increasing plant size, the larger cavity receivers have a lower ratio of aperture area to absorber area, resulting in lower thermal losses. In addition to receiver efficiency, Table 6.4 also lists field efficiency, which is the product of receiver and concentrator efficiency.

TABLE 6.4. Annual Average Efficiency of a Molten Salt Cavity Receiver

Plant Size, MWe	Receiver Efficiency		Field Efficiency	
	No Storage	0.4 Capacity Factor	No Storage	0.4 Capacity Factor
0.5	75.8	77.5	40.1	39.4
2.0	81.7	81.6	49.4	53.5
10.0	83.1	84.1	53.1	53.2
30.0	84.9	85.4	54.1	54.2
100.0	85.7	87.1	53.6	53.6

6.1.3 Transport Subsystem

The transport subsystem involves both the transport of the molten salt between the receiver and the storage tanks and the transport of the molten salt between the storage tanks and the heat engine.

6.1.3.1 Design-Point Performance Assessment Results

The transport subsystem performance was characterized using the methods described in Section 4.2.3.1. This resulted in transport subsystem design-point performance being determined for each combination of plant size and storage. An example of transport subsystem design-point efficiency results is presented in Table 6.5 for molten salt central receiver systems with a capacity factor of 0.4. The thermal efficiency indicates the impact of heat losses from the piping, and pumping power indicates the mechanical energy required to overcome viscous drag and elevation head.

The central receiver transport thermal efficiencies are high, resulting from the relatively short, large-diameter pipes that are used. Because no mechanical energy is recovered from the receiver downcomer, the tower height directly affects the pumping power required. In contrast, the storage discharge pumping requirements does not have a significant gravity head to overcome. As a result, these pumping requirements are less than 5% of the field pumping power requirements.

TABLE 6.5. Design-Point Performance for Molten Salt Transport Subsystem
(In CR-Salt System with a 0.4 Capacity Factor)

System Size, MWe	Receiver Supply		Storage Discharge	
	Thermal Efficiency	Pump Power, kWe	Thermal Efficiency	Pump Power, kWe
0.5	0.9790	20	0.9974	3.1
2.0	0.9895	71	0.9986	11.5
10.0	0.9960	331	0.9997	44.7
30.0	0.9977	1066	0.9999	117.0
100.0	0.9990	4716	0.9999	516.0

6.1.3.2 Transport Subsystem Average Annual Performance

The annual average transport figure of merit as defined in Section 4.1 is reported in Table 6.6. This characteristic of the transport system includes the impact of both thermal losses and pumping power for both the receiver supply and storage discharge piping. The figure of merit for the central receiver molten salt transport subsystem is the lowest of the central receivers, but it is still quite high. The figure of merit peaks at 30 MWe; at higher power levels the pumping losses resulting from the receiver tower height dominate, and at lower power levels the thermal losses dominate.

6.1.4 Storage Subsystem

The storage subsystem consists solely of the molten salt storage tanks and molten salt. The charging and discharge piping and equipment is included in the transport subsystem. The main loss mechanism for the storage subsystem is thermal losses from the storage tank.

6.1.4.1 Design-Point Performance

The thermal losses from the storage tanks were calculated using the methods described in Section 4.2.4 and are reported on Table 6.7 along with the dimensions of the storage tanks. The results are reported as a function of power level and hours of storage.

Obviously, tank dimensions increase with plant size and storage capacity. Large tanks have less surface area for the contained volume than small tanks, resulting in lower thermal losses for the larger tanks.

TABLE 6.6. Average Annual Figure of Merit for a Molten Salt Transport Subsystem

<u>Plant Size, MWe</u>	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	96.2	96.4
2.0	97.4	97.7
10.0	98.1	98.2
30.0	98.2	98.3
100.0	97.9	97.9

TABLE 6.7. Design-Point Performance for a Molten Salt Storage Subsystem

<u>Power Level, MWe</u>	<u>Storage Time, h</u>	<u>Height, m</u>	<u>Diameter, m</u>	<u>Loss Per Hour, %</u>
0.5	3	5.0	2.5	0.324
	6	7.0	3.5	0.116
	9	8.0	4.0	0.0672
	12	9.0	4.5	0.045
2.0	3	9.0	4.5	0.18
	6	9.0	6.0	0.067
	9	10.0	6.5	0.045
	12	10.5	7.0	0.029
10.0	3	12.0	8.0	0.10
	6	11.0	11.0	0.037
	9	12.5	12.5	0.021
	12	14.0	14.0	0.014
30.0	3	12.5	12.5	0.065
	6	16.0	16.0	0.025
	9	18.0	18.0	0.015
	12	20.0	20.0	0.010
100.0	3	19.0	19.0	0.0425
	6	15.0	30.0	0.0130
	9	17.0	34.0	0.0079
	12	19.0	38.0	0.0053

6.1.4.2 Average Annual Storage Performance

The figure of merit for storage performance is described in Section 4.1 and includes the impact of thermal losses from the tank and discharge piping pumping power. The storage figure of merit is reported on Table 6.8.

The storage figure of merit is quite high for all plant sizes but increases with plant size. This increase is due to larger plants having larger storage tanks, which in turn have reduced thermal losses.

TABLE 6.8. Average Annual Figure of Merit for a Molten Salt Storage Subsystem

<u>Plant Size, MWe</u>	<u>Figure of Merit for 0.4 Capacity Factor</u>
0.5	96.8
2.0	97.7
10.0	98.9
30.0	99.1
100.0	99.1

6.1.5 Energy Conversion Subsystem

The energy conversion subsystem in the CR-Salt concept includes the steam Rankine heat engine, heat rejection equipment, and auxiliary components.

6.1.5.1 Design Point Performance

As discussed in Section 4.2.5.2, the energy conversion subsystem is modeled as having a design-point gross efficiency corrected for part-load operation and ambient temperature. The design-point gross efficiencies are presented in Table 6.9. The design-point gross efficiencies do not include the impact of parasitics losses and are consequently higher than the 35 to 39% efficiency normally quoted for large steam power plants. The design-point efficiency increases with increasing plant size. The increase is caused by improved component efficiency (particularly for the steam turbine) at larger plant sizes.

TABLE 6.9. Gross Efficiency of a CR-Salt Energy Conversion Subsystem

<u>Plant Size, MWe</u>	<u>Energy Conversion Gross Efficiency</u>
0.5	0.423
2.0	0.385
10.0	0.357
30.0	0.328
100.0	0.316

The part-load correction for all steam turbines used in the study is shown in Figure 6.2. In this study, all turbine designs were assumed to have a 20% turn-down ratio. Below 20% load, it was assumed that the facility was shut down. The ambient temperature correction for the steam turbines is shown in Figure 6.3. As ambient temperature increases, the correction factor decreases. The decrease simulates the reduced heat engine efficiency due to higher condenser backpressure related to high cooling water temperature caused by higher ambient temperature in the evaporative cooling tower.

6.1.5.2 Average Annual Efficiency

The average annual gross energy conversion efficiency, including part-load and ambient temperature correction, is shown on Table 6.10 for plants with a range of sizes and with both no storage and a storage capacity factor of 0.4.

FIGURE 6.2. Part-Load Efficiency for Steam Turbine Energy Conversion Subsystem

FIGURE 6.3. Temperature Correction for Steam Turbine Energy Conversion Subsystem

TABLE 6.10. Average Annual Efficiency of the Energy Conversion Subsystem in a CR-Salt System

Plant Size, MWe	Efficiency	
	No Storage	0.4 Capacity Factor
0.5	29.7	29.9
2.0	30.7	31.1
10.0	33.2	34.0
30.0	35.8	36.8
100.0	39.7	40.6

As with the design-point gross efficiency, the average annual efficiency increases with increasing plant size. In addition, the plants with a capacity factor of 0.4 consistently have a higher efficiency than do these with the no-storage designs. This difference results from the no-storage plants being at part-load conditions more often than are plants with storage.

6.1.6 Parasitic Losses

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), pumping power, and fixed operating power. When the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

6.1.6.1 Design-Point Parasitics

The design-point parasitics are presented in Table 6.11 as a function of plant size. The first two columns present the energy conversion parasitics. The "b" coefficient is constant across plant size because these parasitic loads were estimated from the Saguaro plant design and scaled based on plant size. The third column presents field-related parasitic loads that are dominated by

TABLE 6.11. Design-Point Parasitic Loss Characteristics
for a CR-Salt System

Plant Size, MWe	Energy Conversion Parasitic ^(a)		Field Parasitic, kW/m ²	Fixed Parasitic, kW	Standby Parasitic, kW
	a, kW	b, kWe/kWt			
0.5	6.4	0.012	0.00128	6.0	8.2
2.0	27.1	0.012	0.00128	23.7	30.8
10.0	113.8	0.012	0.00128	113.3	141.3
30.0	316.0	0.012	0.00128	286.0	400.0
100.0	959.0	0.012	0.00128	915.0	1173.0

(a) Parasitic power = $a + b * (\text{thermal input to the heat engine})$; where "a" is a constant electrical parasitic loss whenever the turbine is operating and " $b * (\text{thermal input to the heat engine})$ " is an electrical parasitic loss which varies with the thermal input to the heat engine.

heliostat drives and consequently are constant when calculated on an area-normalized basis. Fixed parasitics are presented on the forth column, and standby parasitics are described in column five. A more detailed discussion of the definition of the various components of parasitic losses is presented in Section 4.2.6.

6.1.6.2 Average Annual Plant Parasitics

The average annual parasitic power consumption is reported on Table 6.12 as a percent of plant gross output. The results show that parasitics are relatively constant with plant size except for very small facilities where the overall drop in conversion efficiency results in relatively larger fields (with larger field parasitics) and relatively larger steam flow rates (with larger energy conversion parasitics). The slight increase in parasitic loads for the 100 MWe case shows the impact of increased pumping power required to overcome the elevate head associated with increased tower height.

6.2 SODIUM EXTERNAL CENTRAL RECEIVER WITH SODIUM STORAGE

The CR-Na concept consists of a heliostat field for the concentrator subsystem, an external sodium receiver subsystem, sodium transport subsystem, sodium storage subsystem, and a steam Rankine heat engine for the energy conversion subsystem. The performance of each subsystem is discussed below.

6.2.1 Concentrator Subsystem

The concentrator subsystem includes a surround field of heliostats with performance characteristics representative of the ARCO 148.6 m² heliostat.

TABLE 6.12. Average Annual Parasitic Loads for a CR-Salt System

<u>Plant Size MWe</u>	<u>% of Gross Output Used for Parasitic Loads</u>
0.5	15.1
2.0	13.8
10.0	12.2
30.0	12.1
100.0	12.5

6.2.1.1 Design Point Performance

The primary figure of merit for describing concentrator performance is concentrator efficiency. As discussed in Section 4.0, concentrator efficiency includes the impact of cosine losses, heliostat reflectivity losses, shadowing, blocking, atmospheric attenuation, and receiver intercept losses. Concentrator efficiency is reported in two ways. First, concentrator efficiency is shown in Figure 6.4 as a function of altitude and azimuth for a 30 MWe plant with a solar multiple of 1.0. When compared to a north field, a surround field is not as dependent on the azimuth, and on average has a lower efficiency. Second, the concentrator efficiency is presented in Table 6.13 as a function of plant size for a solar multiple of 1.0. A review of Table 6.13 shows that concentrator efficiency is severely degraded at small plant sizes. The concentrator

FIGURE 6.4. Concentrator Efficiency for a 30 MWe Surround Field in a CR-Na System

TABLE 6.13. Concentrator Average Annual Efficiency in a CR-Na System

System Size, MWe	Solar Multiple = 1.0 Azimuth = 0°, Altitude = 80°	Average Annual Efficiency	
		No Storage	0.4 Capacity Factor
0.5	42.1	50.1	48.8
2.0	57.4	56.3	55.8
10.0	65.7	56.5	58.7
30.0	71.7	59.2	60.3
100.0	67.3	60.6	60.8

efficiency actually shows a peak at plant sizes of 30 MWe and declines slightly as plant size increases to 100 MWe. This decline is caused by increased atmospheric attenuation associated with larger field sizes.

6.2.1.2 Average Annual Performance

The average annual concentrator efficiency is reported in two ways. The components of average annual concentrator efficiency are presented, followed by a discussion of the overall concentrator performance.

The components of average annual concentrator efficiency for the sodium system are shown in Table 6.14. These results are from DELSOL2 simulations and are reported for two field sizes. The large field is sized for a 100 MWe plant with a solar multiple of 2.8. The small field is sized for a 0.5 MWe power plant with a solar multiple of 1.0.

TABLE 6.14. Components of Average Annual Concentrator Efficiency in a CR-Na System

Loss Mechanism	Efficiency	
	Small Field	Large Field
Cosine Loss	0.82	0.78
Reflection Loss	0.92	0.92
Shadowing	0.97	0.95
Blocking	0.90	0.99
Atmospheric Attenuation	0.98	0.91
Intercept Losses	0.66	0.99

As with the other central receiver systems, the heliostat "out of box" reflectivity is assumed to be 0.94, and the degradation factor is assumed to be 0.9787. The degradation factor is based on bi-weekly heliostat cleaning. These two factors result in an annual average reflectivity of 0.92.

A review of Table 6.14 shows that the major loss mechanisms for the larger field size are cosine losses, shadowing, and atmosphere attenuation. The major loss mechanisms for the small field size are cosine losses, blocking, and intercept losses. A comparison of the large field and small field design-point performance clearly shows that the small field performance is substantially below that for the large field and that the major cause of the performance difference is intercept losses.

The annual average concentrator efficiencies for the no-storage design and the 0.4-capacity-factor design are listed in Table 6.13. The efficiency is only slightly higher at 100 MWe than at 30 MWe, indicating that atmospheric attenuation is beginning to dominate the losses and that few efficiency gains can be expected at larger field sizes.

6.2.2 Receiver Subsystem

The receiver subsystem includes a sodium receiver that is based on a single-pass external receiver design.

6.2.2.1 Design-Point Performance

The sodium receiver was analyzed using the method described in Section 4.0, and the resulting design-point thermal efficiency is reported in Table 6.15 as a function of plant size for plants with a solar multiple of

TABLE 6.15. Design-Point Thermal Efficiency for a Sodium Receiver
(Solar Multiple = 1.0)

<u>Plant Size, MWe</u>	<u>Receiver Design-Point Thermal Efficiency</u>
0.5	76.2
2.0	84.3
10.0	90.4
30.0	91.8
100.0	92.7

1.0. The effective absorption, based on received tube material and absorber coating, is 0.96. The reflection losses are included in the design-point thermal efficiency reported in Table 6.15. A review of Table 6.15 shows that receiver design-point efficiency increases slightly as plant size increases. It is difficult to draw firm conclusions about the reported trends in receiver efficiency because the receiver design characteristics were taken from DELSOL2, which optimizes the combination of concentrator and receiver. At small plant sizes DELSOL2 could produce optimized designs for similar plants which showed substantial differences in specific details. In general, the improved performance of the receiver with increasing plant size is caused by a decrease in absorber area relative to the incident flux.

The design-point designs show an interesting feature of the external sodium receiver design. A major advantage of a sodium receiver is that the thermal flux on the receiver tubes can be high, because of the high thermal conductivity of sodium. The design of the concentrator/receiver layout, performed by DELSOL2, was done with a peak receiver incident flux of 1.2 MWe/m^2 ; the maximum average incident flux for any design was 0.54 MWe/m^2 . However, the DELSOL2 code does not simply maximize the receiver average incident thermal flux based on the input parameters, but instead considers the trade-off between high receiver efficiency (minimal absorber area, high flux), low concentrator/receiver spillage losses (maximum absorber area), and low capital costs. As shown in Figure 6.5, only one receiver design has an average incident flux greater than $0.5 \text{ MW}_t/\text{m}^2$. In fact, the receiver average incident thermal flux for the smaller plant sizes is more than four times smaller than the maximum allowable peak incident flux. This difference is due to the large absorber surface area required to prevent excessive spillage losses. For these small sodium receivers, a large improvement could be achieved through the use of smaller heliostats, but these improvements were not considered in this study. The important point is that the high thermal flux ability associated with a sodium receiver may not be a useful advantage, depending on plant size and heliostat characteristics.

In this study, the design-point receiver thermal losses can be modified by ambient temperature and wind speed. In the case of the sodium receiver, losses

FIGURE 6.5. Sodium Receiver Thermal Flux

were adjusted for ambient temperature but not for wind speed. The high degree of uncertainty in estimating convective losses and the lack of full-scale experimental data prevent accurate wind speed adjustments.

6.2.2.2 Annual Average Results

The annual average receiver efficiencies of sodium receivers are listed in Table 6.16 as a function of plant size for plants with no storage and for plants with a capacity factor of 0.4. As with the design-point efficiencies, average annual receiver efficiency generally increases with plant size. This increase is caused primarily by an increase in receiver average flux at large plant sizes. An increase in receiver flux reduces thermal losses. In addition to receiver efficiency, field efficiency is listed in Table 6.16. Field efficiency is the product of receiver and concentrator efficiency.

TABLE 6.16. Annual Average Efficiency of a Sodium Receiver in a CR-Na System

Plant Size, MWe	Receiver Efficiency		Field Efficiency	
	No Storage	0.4 Capacity Factor	No Storage	0.4 Capacity Factor
0.5	67.1	64.5	33.6	31.5
2.0	75.9	75.5	42.7	42.1
10.0	82.7	82.6	46.7	48.5
30.0	84.4	83.9	50.0	50.6
100.0	84.2	83.4	51.0	50.7

6.2.3 Transport Subsystem

The transport subsystem in the CR-Na concept involves both the transport of the liquid sodium between the receiver and the storage tanks and the transport of the liquid sodium between the storage tanks and the heat engine.

6.2.3.1 Design-Point Performance Assessment Results

The transport system performance was characterized using the methods described in Section 4.2.3.1. Transport system design-point performance was determined for each combination of plant size and storage. An example of transport system design point efficiency results is presented in Table 6.17 for CR-Na systems with a capacity factor of 0.4. The thermal efficiency indicates the impact of heat losses from the piping, and pumping power indicates the mechanical energy required to overcome viscous drag and elevation head.

TABLE 6.17. Design-Point Performance for a Liquid Sodium Transport System (in a CR-Na System with a 0.4 Capacity Factor)

System Size, MWe	Receiver Supply		Storage Discharge	
	Thermal Efficiency	Pump Power (kWe)	Thermal Efficiency	Pump Power, kW
0.5	0.9800	8.3	0.9938	1.5
2.0	0.9885	27.0	0.9989	6.3
10.0	0.9965	140.0	0.9995	23.5
30.0	0.9984	481.0	0.9999	56.5
100.0	0.9991	2623.0	0.9999	225.0

The central receiver transport thermal efficiencies are high because of the relatively short, large-diameter pipes that are used. Because no mechanical energy is recovered from the receiver downcomer, the tower height directly affects the pumping power required. In contrast, the storage discharge pumping requirements do not have any significant gravity head to overcome. As a result, these pumping requirements are less than 5% of the field pumping power requirements.

6.2.3.2 Transport Subsystem Average Annual Performance

The annual average transport figure of merit as defined in Section 4.1 is reported in Table 6.18. This characteristic of the transport system includes the impact of both thermal losses and pumping power for both the receiver supply and storage discharge piping. The figure of merit peaks at 30 MWe; at higher power levels the pumping losses resulting from the receiver tower height dominate, and at lower power levels the thermal losses dominate.

6.2.4 Storage Subsystem

The storage subsystem consists solely of the sodium storage tanks and sodium. The charging and discharge piping and equipment is included in the transport subsystem. The main loss mechanism for the storage subsystem is thermal losses from the storage tank.

6.2.4.1 Design-Point Performance

The thermal losses from the storage tanks were calculated using the methods described in Section 4.2.4 and are reported on Table 6.19 along with the dimensions of the storage tanks. The results are reported as a function of power level and hours of storage.

TABLE 6.18. Average Annual Figure of Merit
for a Sodium Transport Subsystem

<u>Plant Size, MWe</u>	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	97.7	97.7
2.0	97.7	97.8
10.0	98.4	98.6
30.0	98.6	98.6
100.0	98.2	98.1

TABLE 6.19. Design-Point Performance for a Sodium Storage Subsystem

<u>Power Level, MWe</u>	<u>Storage Hours</u>	<u>Height, m</u>	<u>Diameter, m</u>	<u>No. of Tanks</u>	<u>Loss Per Hour, %</u>
0.5	3	2.9	6.0	1	0.72
	6	3.8	7.2	1	0.52
	9	4.2	8.3	1	0.44
	12	4.6	9.2	1	0.38
2.0	3	4.6	9.2	1	0.38
	6	5.6	11.5	1	0.28
	9	6.8	12.7	1	0.23
	12	7.1	14.3	1	0.20
10.0	3	8	15	1	0.18
	6	9.5	18.8	1	0.14
	9	10.7	21.5	1	0.11
	12	12	23.5	1	0.10
30.0	3	10.5	21.5	1	0.11
	6	13	26.5	1	0.08
	9	15	30.0	1	0.07
	12	16.5	32.7	1	0.06
100.0	3	15.5	31	1	0.07
	6	19	38.2	1	0.06
	9	17.5	34.2	2	0.06
	12	19	37.7	2	0.05

Obviously, tank dimensions increase with plant size and storage capacity. Large tanks have less surface area for the contained volume than small tanks do, resulting in lower thermal losses for the larger tanks.

6.2.4.2 Average Annual Storage Performance

The figure of merit for storage performance in the CR-Na concept is described in Section 4.1 and includes the impact of thermal losses from the tank and discharge piping pumping power. The storage figure of merit is reported on Table 6.20.

TABLE 6.20. Average Annual Figure of Merit for a Sodium Storage Subsystem

<u>Plant Size, MWe</u>	<u>Figure of Merit for 0.4 Capacity Factor</u>
0.5	90.4
2.0	95.8
10.0	97.7
30.0	98.8
100.0	99.2

The storage figure of merit is quite high for all plant sizes but increases with plant size. This increase is due to larger plants having larger storage tanks, the larger tanks in turn have reduced thermal losses.

6.2.5 Energy Conversion Subsystem

The energy conversion subsystem used with the CR-Na concept is identical to that used with the CR-Salt system described in Section 6.1.5.

6.2.5.1 Design-Point Performance

The energy conversion subsystem for the CR-Na concept is essentially the same as for the molten salt subsystem. Therefore, the results presented in Section 6.1.5.1 are applicable to this concept.

6.2.5.2 Average Annual Performance

The average annual gross energy conversion efficiency, including part-load and ambient temperature correction, is shown on Table 6.21 for plants with a

TABLE 6.21. Average Annual Efficiency of the Energy Conversion Subsystem in a CR-Na System

<u>Plant Size, MWe</u>	<u>Conversion Efficiency</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	30.0	30.1
2.0	30.8	31.5
10.0	33.2	34.5
30.0	36.0	36.8
100.0	39.9	40.6

range of sizes and with both no storage and a storage capacity factor of 0.4. As with the design-point gross efficiency, the average annual efficiency increases with plant size. In addition, the plant designs with a capacity factor of 0.4 consistently have a higher efficiency than those for the no-storage designs. This difference was caused by the no-storage plants being at part-load conditions more often than are plants with storage.

6.2.6 Parasitic Losses

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), pumping power and fixed operating power. When the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

6.2.6.1 Design-Point Parasitics

The CR-Na concept is modeled as having plant parasitic loads (other than transport pump power) similar to the parasitic loads associated with molten salt concept. Therefore, the design-point parasitic load reported in Section 6.1.6.1 is applicable to this concept.

6.2.6.2 Average Annual Plant Parasitics

The average annual parasitic power assumption is presented in Table 6.22 as a percent of plant gross output. The results show that parasitic loads represent between 11% and 13.5% of the plants gross output with the percentage of parasitic loads generally increasing slightly with decreasing plant size.

TABLE 6.22. Average Annual Parasitic Loads for a CR-Na System

<u>Plant Size (MWe)</u>	<u>% of Gross Output Used for Parasitic Loads</u>
0.5	0.1230
2.0	0.1341
10.0	0.1210
30.0	0.1153
100.0	0.1097

6.3 SODIUM EXTERNAL CENTRAL RECEIVER WITH SALT STORAGE

The CR-Na/Salt concept consists of a heliostat field for the concentrator subsystem, an external sodium receiver, sodium transport subsystem, and a steam Rankine heat engine for the energy conversion subsystem.

6.3.1 Concentrator Subsystem

The concentrator subsystem includes a surround field of heliostats with performance characteristics representative of the ARCO 148.6 m² heliostat.

6.3.1.1 Design-Point Performance

The concentrator subsystem for this concept is essentially the same as for the CR-Na concept. Therefore, the results presented in Section 6.2.1.1 are applicable to this concept.

6.3.1.2 Average Annual Performance

Although the design-point performance of the concentrator subsystem for the CR-Na/Salt concept is similar to the concentrator subsystem for the CR-Na concept, the average annual performance is slightly different because the different parasitics result in different required field sizes.

The average annual concentrator efficiency is shown in Table 6.23 as a function of plant size for both the no-storage and the 0.4-capacity-factor designs. The efficiency increases with increasing plant size, but the efficiency at 100 MWe is only slightly higher than at 30 MWe, indicating that the atmospheric attenuation is beginning to dominate the losses and that few efficiency gains can be expected at larger field sizes.

TABLE 6.23. Concentrator Average Annual Efficiency in a CR-Na/Salt System

System Size, MWe	Efficiency	
	No Storage	0.4 Capacity Factor
0.5	51.6	47.5
2.0	56.3	55.9
10.0	56.3	58.6
30.0	59.2	60.4
100.0	60.6	60.6

6.3.2 Receiver Subsystem

The receiver subsystem consists of a sodium receiver based on a single-pass external receiver design.

6.3.2.1 Design-Point Performance

The receiver subsystem for the CR-Na/Salt concept is essentially the same as for the CR-Na concept. Therefore, the results presented in Section 6.3.2.1 are applicable to this concept.

6.3.2.2 Average Annual Performance

The annual average receiver efficiencies are listed in Table 6.24 as a function of plant size for plants with no storage and for plants with a capacity factor of 0.4. As with the design-point efficiency, average annual receiver efficiency generally increases with increased plant size. Again, this increased efficiency is caused primarily by an increased receiver average flux at large plant sizes. This increased efficiency allows a relatively smaller receiver with reduced thermal losses. In addition to receiver efficiency, field efficiency is also listed in Table 6.4. Field efficiency is the product of receiver and concentrator efficiency.

6.3.3 Transport Subsystem

The transport subsystem involves both the transport of sodium between the receiver and the storage input heat exchanger and the transport of molten salt between the storage tanks and the heat engine.

TABLE 6.24. Annual Average Efficiency of a Sodium Receiver in a CR-Na/Salt System

Plant Size, MWe	<u>Receiver Efficiency</u>		<u>Field Efficiency</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	68.6	65.3	35.4	31.0
2.0	75.9	75.7	42.7	42.3
10.0	82.3	82.4	46.3	48.3
30.0	84.4	83.6	50.0	50.5
100.0	84.2	83.8	51.0	50.8

6.3.3.1 Design-Point Performance Assessment Results

The field transport piping is the same as the piping used in the CR-Na concept; therefore, the design-point results presented in Section 6.2.3.1 are applicable to this system. The storage discharge piping is similar to that used with the CR-Salt concept; therefore, the design-point results presented in Section 6.1.3.1 are applicable. The design-point performance characteristics are summarized in Table 6.25 for a design with a capacity factor of 0.4.

6.3.3.2 Transport System Average Annual Performance

The annual average transport figure of merit for the CR-Na/Salt concept, as defined in Section 4.1, is reported in Table 6.26. This characteristic of the transport system includes the impact of both thermal losses and pumping power for both the receiver supply and storage discharge piping. The average annual performance of this system is substantially better than that for the

TABLE 6.25 Design-Point Performance for a Molten Salt Transport Subsystem (in a CR-Na/Salt System with 0.4 Capacity Factor)

System Size, MWe	<u>Receiver Supply</u>		<u>Storage Discharge</u>	
	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>
0.5	0.9800	10.3	0.9974	3.1
2.0	0.9885	24.1	0.9986	11.5
10.0	0.9965	84.0	0.9997	44.7
30.0	0.9984	230.0	0.9999	117.0
100.0	0.9990	787.0	0.9999	516.0

TABLE 6.26. Average Annual Figure of Merit for a CR-Na/Salt Transport Subsystem

Plant Size, MWe	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	97.4	97.5
2.0	98.2	98.4
10.0	99.2	99.4
30.0	99.5	99.5
100.0	99.3	99.5

CR-Na concept because the sodium/salt heat exchanger can be pressurized. With pressurization of the heat exchanger, the elevation head does not have to be dissipated in an expansion valve.

6.3.4 Storage Subsystem

The storage subsystem consists solely of the molten salt storage tanks and molten salt. The charging and discharge piping and equipment is included in the transport subsystem. The main loss mechanism for the storage subsystem is thermal losses from the storage tank.

6.3.4.1 Design-Point Performance

The storage system used with this concept is identical to the molten salt storage system described in Section 6.3.4.1. Therefore, the design-point performance reported in Section 6.3.4.1 is applicable to this subsystem.

6.3.4.2 Average Annual Storage Performance

The figure of merit for storage performance is described in Section 4.1 and includes the impact of thermal losses from the tank and discharge piping pumping power. The storage figure of merit is reported in Table 6.27.

The storage figure of merit is quite high for all plant sizes but increases with plant size. This increase is due to the reduced thermal losses of the larger storage tanks in longer plants.

6.3.5 Energy Conversion Subsystem

The energy conversion subsystem used with the CR-Na/Salt concept is identical to that used with the molten salt system described in Section 6.1.5.

TABLE 6.27. Average Annual Figure of Merit for a Molten Salt Storage Subsystem

<u>Plant Size, MWe</u>	<u>0.4 Capacity Factor</u>
0.5	96.8
2.0	97.7
10.0	98.9
30.0	99.1
100.0	99.1

6.3.5.1 Design-Point Performance

The energy conversion subsystem for this concept is essentially the same as for the molten salt system. Therefore, the results presented in Section 6.1.5.1 are applicable to this concept.

6.3.5.2 Average Annual Performance

The average annual gross energy conversion efficiency, including part-load and ambient temperature correction is shown in Table 6.28 for plants with a range of sizes with no-storage and with a storage capacity factor of 0.4. As with the design-point gross efficiency, the average annual efficiency increases with increasing plant size. In addition, the plant designs with a capacity factor of 0.4 consistently have a higher efficiency than for the no-storage designs. This difference was caused by the no-storage plants being at part-load conditions more often than are plants with storage.

6.3.6 Parasitic Loss

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), pumping power, and fixed operating power. Where the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

TABLE 6.28. Average Annual Efficiency of the Energy Conversion Subsystem in a CR-Na/Salt System

<u>Plant Size, MWe</u>	<u>Efficiency</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	30.0	30.1
2.0	30.8	31.1
10.0	33.2	34.0
30.0	36.0	36.9
100.0	39.9	40.6

6.3.6.1 Design-Point Parasitics

The CR-Na/Salt concept is modeled as having plant parasitic loads (other than transport pump power) similar to the parasitic loads associated with the molten salt concept. Therefore, the design-point parasitic load reported in Section 6.1.6.1 is applicable to this concept.

6.3.6.2 Average Annual Plant Parasitics

The average annual parasitic consumption is reported in Table 6.29 as a percent of plant gross output. The results show that parasitics are relatively constant with plant size except for very small facilities where the overall drop in common efficiency results in relatively larger fields (with larger field parasitics) and relatively larger steam flow rate (with larger energy conversion parasitics).

6.4 WATER/STEAM EXTERNAL RECEIVER WITH OIL/ROCK STORAGE

The CR-W/S concept consists of a heliostat field for the concentrator subsystem, an external water/steam receiver, a water/steam transport subsystem, an oil/rock storage subsystem, and a steam Rankine heat engine. The performance of each subsystem is discussed below.

6.4.1 Concentrator Subsystem

The concentrator subsystem includes a surround field of heliostats with performance characteristics representative of ARCO 148.6 m² heliostat.

TABLE 6.29. Average Annual Parasitic Loads for CR-Na/Salt System

<u>Plant Size, MWe</u>	<u>% of Gross Output Used for Parasitic Loads</u>
0.5	13.2
2.0	11.7
10.0	10.7
30.0	9.8
100.0	9.4

6.4.1.2 Design-Point Performance

The concentrator efficiency is presented in Table 6.30 as a function of plant size for a plant with a solar multiple of 1.0 and a solar position of 0° azimuth and 80° altitude. As with other central receiver designs, the concentrator efficiency is severely degraded at small plant sizes. The concentrator efficiency actually shows a peak at plant sizes of 30 MWe and declines slightly as plant size increases to 100 MWe. This decline is caused by increased atmospheric attenuation associated with larger field sizes.

6.4.1.3 Average Annual Performance

The annual average concentrator efficiencies for the no-storage design and the 0.4-capacity-factor design are listed in Table 6.30. The efficiency peaks at a plant size of 30 MWe. At large field sizes, the atmospheric attenuation replaces receiver spillage as the dominant loss.

6.4.2 Receiver Subsystem

The water/steam receiver was analyzed using the procedure described in Section 4.0. Table 6.31 presents the receiver design-point efficiency as a function of plant size for plants with a solar multiple of 1.0. The effective absorption, based on tube material and absorber coating, is 0.96. The reflection losses are included in the design-point thermal efficiency reported in Table 6.31. A review of Table 6.31 shows that receiver design-point efficiency is nearly constant with plant size.

TABLE 6.30. Concentrator Efficiency for a CR-W/S System

System Size, MWe	Solar Multiple = 1.0 Azimuth = 0°, Altitude = 80°	Average Annual	
		No Storage	0.4 Capacity Factor
0.5	39.1	45.6	44.3
2.0	57.6	49.0	50.3
10.0	66.5	57.7	58.2
30.0	68.8	60.0	59.5
100.0	67.5	59.8	58.3

TABLE 6.31. Design-Point Thermal Efficiency of a Water/Steam Receiver

<u>Plant Size, MWe</u>	<u>Receiver Design-Point Thermal Efficiency</u>
0.5	85.8
2.0	87.7
10.0	87.7
30.0	87.8
100.0	87.7

For all power levels, the DELSOL2 designs have an average incident flux very close to 0.225 MWe/m^2 , which was a design constant. This similarity in flux results in receiver design-point thermal efficiency being nearly constant across all plant sizes.

The design-point receiver thermal losses can be modified by ambient temperature and wind speed. In the case of the water/steam receiver, losses were adjusted for ambient temperature but not for wind speed. The wind speed correction was not included because of the high degree of uncertainty in estimating convective losses and the lack of full-scale experimental data.

6.4.2.1 Annual Average Results

The annual average receiver efficiencies as a function of plant size for plants with no storage and for plants with a capacity factor of 0.4 are listed in Table 6.32. The efficiency varies only 4.2% over the range of power levels shown, as compared to the 30% change that occurs with the sodium receiver.

TABLE 6.32. Average Annual Efficiency of a Water/Steam Receiver

<u>Plant Size, MWe</u>	<u>Receiver Efficiency</u>		<u>Field Efficiency</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	76.1	75.8	34.7	33.6
2.0	77.8	77.8	38.1	39.1
10.0	77.9	78.3	45.0	45.6
30.0	78.7	78.7	47.2	46.8
100.0	78.6	79.0	47.0	46.0

This steady efficiency is caused by the optimization procedure used in DELSOL2, which produced designs with improved receiver efficiency when compared to the sodium designs but had a decreased collector efficiency. This effect is shown by comparing field efficiency (the product of receiver and collector efficiency) for a water/steam and sodium system. The comparison will show that both concepts have similar performance economies of scale.

6.4.3 Transport Subsystem

The transport subsystem for the CR-W/S concept involves both the transport of the water/steam between the receiver and the plant and the transport of oil from the storage tanks to the storage charging and discharging heat exchanges.

6.4.3.1 Design-Point Performance Assessment Results

The transport system performance was characterized using the methods described in Section 4.2.3.1. Transport system design-point performance was determined for each combination of plant size and storage. An example of transport system design-point efficiency results is presented in Table 6.33 for CR-W/S systems with a capacity factor of 0.4. The thermal efficiency indicates the impact of heat losses from the piping, and pumping power indicates the mechanical energy required to overcome viscous drag and elevation head.

The central receiver transport thermal efficiencies are high, because of the relatively short, large-diameter pipes that are used. The storage

TABLE 6.33. Design-Point Discharge for the Transport Subsystem
in a CR-W/S System (0.4 Capacity Factor)

<u>System Size, MWe</u>	<u>Receiver Supply</u>		<u>Storage Discharge</u>	
	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>
0.5	0.9934	18	0.9989	3.1
2.0	0.9967	62	0.9994	11.3
10.0	0.9980	259	0.9998	51.2
30.0	0.9988	749	0.9999	146.0
100.0	0.9990	2986	0.9999	477.0

discharge pumping requirements do not have a significant gravity head to overcome. As a result, these pumping requirements are less than 5% of the field pumping power requirements.

6.4.3.2 Transport System Average Annual Performance

The annual average transport figure of merit as defined in Section 4.1 is reported in Table 6.34. This characteristic of the transport system includes the impact of both thermal losses and pumping power for both the receiver supply and storage discharge piping. The efficiency of the CR-W/S transport system is the lowest for any of the central receivers, but it is still quite high. The efficiency peaks at 30 MWe; at higher power levels the pumping losses due to the receiver tower height dominate and at lower power levels the thermal losses dominate.

6.4.4 Storage Subsystem

The storage subsystem for the CR-W/S concept consists of the oil/rock storage media, storage tanks, storage charging, and discharge heat exchangers. The main loss mechanism for the storage subsystem is thermal loss from the storage tank.

6.4.4.1 Design-Point Performance

The thermal losses from the storage tanks were calculated using the methods described in Section 4.2.4 and are reported on Table 6.35 along with the dimensions of the storage tanks. The results are reported as a function of power level and hours of storage.

TABLE 6.34. Average Annual Figure of Merit for the Transport Subsystem in a CR-W/S System

<u>Plant Size, MWe</u>	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	98.2	98.3
2.0	98.3	98.2
10.0	98.6	98.6
30.0	98.8	98.6
100.0	98.7	98.4

TABLE 6.35. Design-Point Performance for the Oil/Rock Storage Subsystem in a CR-W/S System

<u>Power Level, MWe</u>	<u>Storage Hours</u>	<u>Height, m</u>	<u>Diameter, m</u>	<u>No. of Tanks</u>	<u>Percent Loss Per Hour</u>
0.5	3	3.7	7.4	1	0.062
	6	4.6	9.2	1	0.044
	9	5.2	10.4	1	0.036
	12	5.7	11.4	1	0.031
2.0	3	5.5	11.2	1	0.032
	6	7.0	14.0	1	0.022
	9	8.0	16.0	1	0.017
	12	8.8	17.6	1	0.015
10.0	3	9.5	19.1	1	0.013
	6	12.0	23.4	1	0.009
	9	13.4	26.8	1	0.007
	12	14.7	29.4	1	0.006
30.0	3	13.5	27.1	1	0.007
	6	16.7	33.5	1	0.005
	9	19.0	38.0	1	0.004
	12	20.0	30.2	2	0.006
100.0	3	20.0	40.2	1	0.004
	6	20.0	39.1	2	0.004
	9	20.0	38.7	3	0.004
	12	20.0	38.5	4	0.004

Obviously, tank dimensions increase with plant size and storage capacity. Larger tanks have less surface area for the contained volume than small tanks, resulting in lower thermal losses for the larger tanks. The percent loss per hour is more than an order of magnitude less for the oil/rock storage system than for the sodium or molten salt storage systems because of the lower operating temperature of the salt.

6.4.4.2 Average Annual Storage Performance

The figure of merit for storage performance is described in Section 4.1 and includes the impact of thermal losses from the tank and discharge piping pumping power. The storage figure of merit is reported on Table 6.36.

The storage figure of merit is quite high for all plant sizes but generally increases with plant size. This increase is due to the reduced thermal losses of larger storage tanks.

6.4.5 Energy Conversion Subsystem

The energy conversion subsystem for the CR-W/S concept includes the steam Rankine heat engine, heat rejection equipment, and auxiliary components.

6.4.5.1 Design-Point Performance

As discussed in Section 4.2.5, the energy conversion subsystem of the CR-W/S concept is modeled as having a design-point gross efficiency that is corrected for part-load operation and ambient temperature. The design-point gross efficiencies are presented in Table 6.37. The design-point gross efficiencies do not include the impact of parasitics losses. Two design-point gross efficiencies are reported: gross efficiency when operating from the field and gross efficiency when operating from storage. When operating from storage, the maximum oil temperature in storage limits the maximum steam temperature to 530°F. This temperature limitation results in a substantial reduction in efficiency. The design-point efficiency increases with increasing plant size. This increase is caused by improved component efficiency (particularly for the steam turbine) at larger plant sizes.

TABLE 6.36. Figure of Merit for the Oil/Rock Storage Subsystem in a CR-W/S System

<u>Plant Size, MWe</u>	<u>Figure of Merit for 0.4 Capacity Factor</u>
0.5	96.7
2.0	98.0
20.0	98.7
30.0	99.0
100.0	98.4

6.4.6.1 Design-Point Parasitics

The design-point parasitics are presented in Table 6.39 as a function of plant size. The first two columns present the energy conversion parasitics. The "b" coefficient is constant across plant size because these parasitic loads were estimated from the Saguaro plant design, modified for the water/steam configuration, and scaled based on plant size. The third column presents field-related parasitic loads, which are dominated by heliostat drives and consequently are constant when calculated on an area-normalized basis. Fixed parasitics are presented in the fourth column, and standby parasitics are described in column five. A more detailed discussion of the definition of the various components of parasitic losses is presented in Section 4.2.6.

6.5.6.2 Average Annual Plant Parasitics

The average annual parasitic power consumption is reported on Table 6.40 as a percent of plant gross output. The results show that parasitics are relatively constant with plant size except for very small facilities where the overall drop in conversion efficiency results in relatively larger fields (with larger field parasitics). When compared with other central receiver systems, the CR-W/S concept has the largest parasitic power consumption. This large consumption is caused by the lower overall conversion efficiency of this concept when compared with other central receiver systems.

TABLE 6.39. Design-Point Parasitic Loss Characteristics for a CR-W/S System

Plant Size, MWe	Energy Conversion Parasitic ^(a)		Field Parasitic, kW/m ²	Fixed Parasitic, kW	Standby Parasitic, kW
	a, kW	b, kWe/kWt			
0.5	6.5	0.013	0.00128	5.8	7.1
2.0	27.0	0.013	0.00128	22.4	27.2
10.0	140.0	0.013	0.00128	103.0	131.0
30.0	316.0	0.013	0.00128	304.0	373.0
100.0	996.0	0.013	0.00128	931.0	1163.0

(a) Parasitic power = $a + b * (\text{thermal input to the heat engine})$; where "a" is a constant electrical parasitic loss whenever the turbine is operating and " $b * (\text{thermal input to the heat engine})$ " is an electrical parasitic loss that varies with the thermal input to the heat engine.

TABLE 6.37. Gross Efficiency of the Energy Conversion Subsystem in a CR-W/S System

<u>Plant Size, MWe</u>	<u>Efficiency</u>	
	<u>Operating From the Field</u>	<u>Operating From Storage</u>
0.5	0.314	0.233
2.0	0.327	0.243
10.0	0.354	0.264
30.0	0.385	0.272
100.0	0.407	0.275

6.4.5.2 Average Annual Efficiency

The average annual gross energy conversion efficiency, including part-load and ambient temperature correction is shown on Table 6.38 for a range of plant sizes with both no storage capacity and a storage capacity factor of 0.4. As with the design-point gross efficiency, the average annual efficiency increases with plant size.

6.4.6 Parasitic Losses

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), pumping power, and fixed operating power. When the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

TABLE 6.38. Average Annual Efficiency of the Energy Conversion Subsystem in a CR-W/S System

<u>Plant Size, MWe</u>	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	29.5	26.3
2.0	30.6	27.7
10.0	33.0	30.2
30.0	34.5	32.6
100.0	38.2	33.5

TABLE 6.40. Average Annual Parasitic Loads for a CR-W/S System

<u>Plant Size, MWe</u>	<u>% of Gross Output Used for Parasitic Loads</u>
0.5	16.3
2.0	16.0
10.0	14.0
30.0	13.5
100.0	13.4

6.5 PARABOLIC DISH WITH STIRLING ENGINE CONVERSION

The dish concept consists of a field of parabolic collectors, with the receiver and Stirling cycle heat engine located at the focal point of the concentrator. A lead-acid battery provides storage. The performance of each major subsystem is discussed below.

6.5.1 Concentrator Subsystem

The concentrator subsystem consists of a field of parabolic concentrators with a design based on a 86-m² parabolic concentrator developed by MDAC.

6.5.1.1 Design-Point Performance

The design-point performance of the dish concentrator was determined using the methods discussed in Section 4.2.1.2. These methods produced estimates for the various loss mechanisms. Concentrator reflectivity was estimated to be 0.94 for a new, clear concentrator. Reflectivity degradation was based on bi-weekly cleaning and was estimated to be 0.95. The concentrator/receiver intercept factor was estimated to be 0.966. Unlike the central receiver concepts, the intercept factor does not vary with sun position. Wind effects on the intercept factor were assumed to be negligible. These component efficiencies result in a design-point concentrator efficiency of 86.3%.

6.5.1.2 Average Annual Performance

The annual average concentrator efficiency is 86.1%. The efficiency is constant for all power levels because of the modular nature of the concept.

This high efficiency is due to the excellent optics of two-axis sun tracking (no cosine loss) and the closeness of the receiver to the collector (minimal atmospheric attenuation).

6.5.2 Receiver Subsystem

The receiver subsystem consists of the receiver located at the focal point of each collector. The receiver is assumed to be similar to the receiver design developed by Advanco.

6.5.2.1 Design Point Performance

The receiver performance was determined using the method described in Section 4.2.2.2. The results of this analysis consist of receiver optical properties and thermal losses. The receiver absorption and emittance are determined by the Stirling engine heater tube material and the receiver geometry. The receiver absorption was estimated to be 0.96, and receiver emittance was estimated to be 0.945. The total receiver thermal loss was estimated to be 0.0854 kW/m² of dish aperture. Receiver thermal losses were corrected for ambient temperature variations but not for wind speed because it was assumed that the cavity design would limit the impact of wind speed variations.

6.5.2.2 Annual Average Performance

The annual average receiver efficiency as defined in Section 4.0 was 81.7%. Because of the modular nature of the design, the annual average receiver efficiency is independent of plant size.

6.5.3 Transport Subsystem

The dish concept is unique in that electric energy rather than thermal energy is transported.

6.5.3.1 Design-Point Performance

Losses in this transport subsystem represent electrical losses in transmission lines and electrical components. Table 6.41 reports the design-point transport efficiency for several field sizes. Transport efficiency decreases with increasing plant size because larger fields have long transmission lines with higher losses.

TABLE 6.41. Efficiency of the Transport Subsystem in a Dish System

<u>Field Size, m²</u>	<u>Transport Efficiency</u>
2.23	0.974
13.5	0.974
56.2	0.972
225.0	0.964
1125.0	0.958

6.5.3.2 Average Annual Performance

The average annual performance is presented in Table 6.42 as a function of plant size for plants with no storage capacity and for plants with a storage capacity factor of 0.4. These results show the same trends as did the design-point results. Transport efficiency decreases with increasing plant size. This decrease in efficiency results from the higher losses associated with larger fields and longer transmission lines in larger plants. The 0.4-capacity-factor designs have losses either equal to or greater than the no-storage case because the 0.4 capacity factor designs will have larger collector fields.

6.5.4 Storage Subsystem

The dish concept is unique in that the storage subsystem stores electrical energy (in batteries) rather than thermal energy.

TABLE 6.42. Average Annual Figure of Merit for the Transport Subsystem of a Dish System

<u>Plant Size, MWe</u>	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	97.4	97.4
2.0	97.4	97.4
10.0	97.2	97.1
30.0	96.7	96.5
100.0	96.2	95.9

6.5.4.1 Design-Point Performance

The lead-acid battery storage subsystem was evaluated using the methods discussed in Section 4.0. The design-point performance can be summarized as an input and output efficiency, battery cycle efficiency, and self-discharge rate. The input and output efficiency was assumed to be equal to the transformer/inverter efficiency (0.95). The battery cycle efficiency varies between 50% and 70% depending on charge and discharge rates. The self-discharge rate was estimated to be 0.003% of the energy in storage per hour.

6.5.4.2 Annual Average Performance

The annual average storage figure of merit for the battery storage system was estimated to be 58.4%. This estimate includes input/output losses, battery cycle inefficiencies, and battery self discharge.

6.5.5 Energy Conversion Subsystem

The energy conversion subsystem for the dish concept consists of a 25 MWe Stirling cycle heat engine located at the focal point of the dish concentrator.

6.5.5.1 Design-Point Performance

Using the analysis procedure described in Section 4.0, the gross energy conversion efficiency of the dish/Stirling engine was determined to be 39.5%. The part-load correction curve is shown in Figure 6.6. The ambient air temperature correction is a function of both the air temperature and the cooling fan set point of 16°C, as shown in Figure 6.7. When the ambient temperature is below 16°C, the fan parasitic energy requirement is greater than the increase in engine output because the lower heat sink temperature. The fan is not used in that case. Conversely, above 16°C the fan parasitic energy requirement is lower than the increase in engine output, so the fan is used.

6.5.5.2 Average Annual Performance

The annual average gross energy conversion efficiency for the Stirling heat engine was estimated to be 35.4%. Because of the modular nature of the design, energy conversion efficiency is constant for all plant sizes.

FIGURE 6.6. Part-Load Correction for the Energy Conversion Subsystem in a Dish System

FIGURE 6.7. Ambient Temperature Correction for the Energy Conversion Subsystem in a Dish System

6.5.6 Parasitic Losses

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), and fixed operating power. When the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

6.5.6.1 Design-Point Parasitics

The design-point parasitics are presented in Table 6.43 as a function of plant size. The first two columns present the energy conversion parasitics. The third column presents field-related parasitic loads that are dominated by dish drives and consequently are constant when calculated on an area-normalized basis. Fixed parasitics are presented in the fourth column, and standby parasitics are described in column five. A more detailed discussion of the definition of the various components of parasitic losses is presented in Section 4.2.6.

TABLE 6.43. Design-Point Parasitic Loss Characteristics for a Dish System

Plant Size, MWe	Energy Conversion Parasitic ^(a)		Field Parasitics kW/cm ²	Fixed Parasitic kW	Standby Parasitics ^(b)	
	a, kW	b, kWe/kWt			c, kW	d, kW/m ²
0.5	0.0	0.0033	0.0046	2.48	1.45	0.0067
2.0	0.0	0.0033	0.0046	9.9	5.8	0.007
10.0	0.0	0.0083	0.0046	49.5	29.0	0.007
30.0	0.0	0.0033	0.0046	148.5	87.0	0.007
100.0	0.0	0.0033	0.0046	495.0	290.0	0.007

(a) Parasitic power = $a + b * (\text{thermal input to the heat engine})$; where "a" is a constant electrical parasitic loss whenever the turbine is operating and " $b * (\text{thermal input to the heat engine})$ " is an electrical parasitic loss that varies with the thermal input to the heat engine.

(b) Parasitic power = $c + d * (\text{concentrator area})$.

6.5.6.2 Average Annual Plant Parasitics

The average annual parasitic power consumption is reported in Table 6.44 as a percent of plant gross output. The results show that parasitics are essentially constant with plant size, which would be expected because of the modular nature of this concept.

6.6 PARABOLIC TROUGH WITH OIL/ROCK STORAGE

The trough concept consists of a field of trough collectors, an oil transport subsystem, an oil/rock thermal energy storage subsystem, and a steam Rankine heat engine for the energy conversion subsystem. The performance of each subsystem is discussed below.

6.6.1 Concentrator Subsystem

The concentrator subsystem consists of a field of trough concentrators with performance similar to the Acurex Corporation 78-m² glass-metal trough collector.

6.6.1.1 Design-Point Performance

The design-point performance of the trough collector was characterized by loss mechanisms that depend on solar position (cosine loss, blocking, and shadowing) and by loss mechanisms that are constant with solar position (reflectivity and intercept losses). The variable loss mechanisms are combined into one geometric optical efficiency. Table 6.45 presents geometric optical efficiency as a function of azimuth and altitude. The concentrator "out of the

TABLE 6.44. Average Annual Parasitic Loads for a Dish System

<u>Plant Size, MWe</u>	<u>% of Gross Output Used for Parasitic Loads</u>
0.5	16.2
2.0	16.1
10.0	16.0
30.0	16.0
100.0	16.0

TABLE 6.45. Geometric Optical Efficiency of a Trough System

Altitude	Azimuth						
	0	20	40	60	80	100	120
0.00	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
5.00	0.0000	0.1100	0.1490	0.1630	0.1700	0.1700	0.1630
15.00	0.1190	0.2980	0.4490	0.4840	0.5040	0.5040	0.4840
25.00	0.3020	0.4160	0.6370	0.7940	0.8240	0.8240	0.7940
35.00	0.4710	0.5440	0.7050	0.8630	0.9670	0.9670	0.8630
45.00	0.6230	0.6690	0.7770	0.8920	0.9710	0.9710	0.8920
55.00	0.7520	0.7790	0.8460	0.9210	0.9760	0.9760	0.9210
65.00	0.8560	0.8700	0.9060	0.9480	0.9800	0.9800	0.9480
75.00	0.9320	0.9370	0.9520	0.9700	0.9850	0.9850	0.9700
80.00	0.9590	0.9620	0.9690	0.9780	0.9860	0.9860	0.9780

box" reflectivity for the trough concentrator was estimated to be 0.94 and the average concentrator reflectivity degradation is 0.95. The concentrator intercept factor was estimated to be 0.98.

6.6.1.2 Average Annual Performance

The average annual concentrator efficiency was determined to be 63.9%. Because of the modular nature of the trough collector, the concentrator efficiency is not a function of plant size.

6.6.2 Receiver Subsystem

The receiver used by the trough concept consists of a steel absorber tube with a surrounding glass tube.

6.6.2.1 Design-Point Performance

The receiver was coated with a selective surface that produced an absorptivity of 0.94. The cover glass has a transmittance of 0.97 which results in an overall receiver absorption of 0.91. The receiver thermal losses were estimated to be 0.157 kW/m^2 of concentrator aperture. The cover glass effectively reduces the impact of wind on receiver losses so that the effects of wind speed or ambient air temperature on thermal losses could be neglected.

6.6.2.2 Average Annual Performance

The average annual receiver efficiency was estimated to be 60.1%. Because of the modular nature of the trough concept, receiver efficiency is constant with plant size. The field efficiency (combined concentrator/receiver efficiency) was 38.4% and is constant with field size also.

6.6.3 Transport Subsystem

The transport subsystem in the trough concept involves both the transport of oil between the receivers and the storage tanks and the transport of oil between the storage tanks and the heat engine.

6.6.3.1 Design-Point Performance

The transport system performance was characterized using the methods described in Section 4.2.3.1. Transport system design-point performance was determined for each combination of plant size and storage. An example of transport system design-point efficiency results is presented in Table 6.46 for trough systems with a capacity factor of 0.4. The thermal efficiency indicates the impact of heat losses from the piping, and pumping power indicates the mechanical energy required to overcome viscous drag.

The thermal efficiency of the field transport piping decreases with increasing plant size. Larger plants have larger collector fields, involving more piping per m^2 of collector area than for smaller fields. Similarly, field pumping power increases with increasing plant size.

TABLE 6.46. Design-Point Performance for the Transport Subsystem of a Trough System (0.4 Capacity Factor)

System Size, MWe	<u>Receiver Supply</u>		<u>Storage Discharge</u>	
	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>	<u>Thermal Efficiency</u>	<u>Pump Power, kWe</u>
0.5	0.928	2.8	0.995	3.9
2.0	0.938	29.0	0.996	17.5
10.0	0.926	225	0.999	78.1
30.0	0.926	2,282	1.0	232.0
100.0	0.918	11,876	1.0	770.0

6.6.3.2 Average Annual Performance

The annual average transport figure of merit as defined in Section 4.1 is reported in Table 6.47. This characteristic of the transport system includes the impact of both thermal losses and pumping power for both the receiver supply piping.

The figure of merit peaks at 2 MWe, then steadily decreases at higher power levels. This optimum is due to the trade-off between losses increasing with increasing pipe length and losses decreasing with increasing pipe diameter.

6.6.4 Storage Subsystem

The storage subsystem in the trough concept consists of the oil/rock storage tanks and oil. The charging and discharge piping and equipment is included in the transport subsystem. The main loss mechanism for the storage subsystem is thermal loss from the storage tank.

6.6.4.1 Design-Point Performance

The thermal losses from the storage tanks were calculated using the methods described in Section 4.2.4 and are reported in Table 6.48 along with the dimensions of the storage tanks. The results are reported as a function of power level and hours of storage.

Obviously, tank dimensions increase with plant size and storage capacity. Larger tanks have less surface area in proportion to the contained volume than do small tanks, resulting in lower thermal losses for the larger tanks.

TABLE 6.47. Average Annual Figure of Merit for the Transport System in a Trough System

<u>Plant Size, MWe</u>	<u>Figure of Merit</u>	
	<u>No Storage</u>	<u>0.4 Capacity Factor</u>
0.5	92.5	92.6
2.0	94.0	93.8
10.0	91.6	91.6
30.0	90.1	90.0
100.0	87.7	87.6

TABLE 6.48. Design-Point Performance for the Oil/Rock Storage Subsystem in a Trough System

<u>Power Level, MWe</u>	<u>Storage Hours</u>	<u>Height, m</u>	<u>Diameter, m</u>	<u>No. of Tanks</u>	<u>Percent Loss Per Hour</u>
0.5	3	4	8.4	1	0.052
	6	5.5	9.9	1	0.037
	9	6	11.6	1	0.029
	12	6.5	12.8	1	0.025
2.0	3	6.5	12.7	1	0.025
	6	8	15.9	1	0.018
	9	9	18.2	1	0.014
	12	10	19.9	1	0.012
10.0	3	10.8	21	1	0.011
	6	13.2	26.4	1	0.008
	9	15	30.1	1	0.006
	12	16.5	33	1	0.005
30.0	3	15	30.2	1	0.006
	6	20	36.3	1	0.004
	9	20	31.2	2	0.003
	12	20	35.9	2	0.004
100.0	3	20	33.5	2	0.005
	6	20	37.8	3	0.004
	9	20	39.8	4	0.004
	12	20	37.4	6	0.004

6.6.4.2 Average Annual Storage Performance

The figure of merit for storage performance is described in Section 4.1 and includes the impact of thermal losses from the tank and discharge piping pumping power. The storage figure of merit is reported in Table 6.49.

The storage figure of merit is quite high for all plant sizes and does not increase with plant size. This fact is due to the large storage tanks with low thermal losses in all plants.

TABLE 6.49. Average Annual Storage Figure of Merit for the Storage Subsystem in a Trough System

<u>Plant Size, MWe</u>	<u>0.4 Capacity Factor</u>
0.5	99.0
2.0	99.0
20.0	99.1
30.0	99.1
100.0	99.0

6.6.5 Energy Conversion Subsystem

The energy conversion subsystem includes the steam Rankine heat engine, heat rejection equipment, and auxiliary components.

6.6.5.1 Design-Point Performance

As discussed in Section 4.2.5, the energy conversion subsystem is modeled as having a design-point gross efficiency corrected for part-load operation and ambient temperature. The design-point gross efficiencies are presented in Table 6.50. The design-point gross efficiencies do not include the impact of parasitics losses.

The part-load correction is shown in Figure 6.8. All turbine designs were assumed to have a 20% turn-down ratio. Below 20% load, the facility was assumed to be shut down. The ambient temperature correction is shown in Figure 6.9. As ambient temperature increases, the correction factor decreases,

TABLE 6.50. Gross Efficiency of the Energy Conversion Subsystem in a Trough System

<u>Plant Size, MWe</u>	<u>Energy Conversion Gross Efficiency</u>
0.5	0.211
2.0	0.239
10.0	0.264
30.0	0.266
100.0	0.272

FIGURE 6.8. Part-Load Correction for the Energy Conversion Subsystem in a Trough System (same curve is valid for all power levels)

simulating the reduced heat engine efficiency caused by higher condenser backpressure. High condenser backpressure is related to high cooling water temperature, which is caused by higher ambient temperature in the evaporative cooling tower.

6.6.5.2 Average Annual Efficiency

The average annual gross energy conversion efficiency, including part-load and ambient temperature correction, is shown on Table 6.51 for a range of plant sizes with both no storage capacity and a capacity factor of 0.4. As with the design-point gross efficiency, the average annual efficiency increases with increasing plant size. In addition, the plant designs with a storage capacity factor of 0.4 consistently have a higher efficiency than the no-storage

FIGURE 6.9. Ambient Temperature Correction for the Energy Conversion Subsystem in a Trough System

TABLE 6.51. Average Annual Efficiency of the Energy Conversion Subsystem in a Trough System

Plant Size, MWe	Efficiency	
	No Storage	0.4 Capacity Factor
0.5	20.5	20.5
2.0	23.0	23.4
10.0	25.1	25.9
30.0	25.3	26.1
100.0	25.9	26.7

designs. This difference is a result of the no-storage plants being at part-load conditions more often than are plants with storage.

6.6.6 Parasitic Losses

The parasitic losses consist of energy conversion subsystem parasitics (which are a function of plant output), field parasitics (which are a function of field size), pumping power, and fixed operating power. When the plant is not in operation, parasitic loads are modeled as consisting solely of standby loads.

6.6.6.1 Design-Point Parasitics

The design-point parasitics are presented in Table 6.52 as a function of plant size. The first two columns present the energy conversion parasitics. The "b" coefficient is constant for all plant sizes because these parasitic loads were estimated from the Saguaro plant design, modified for the trough configuration, and scaled based on plant size. The third column presents field-related parasitic loads that are dominated by trough drives and consequently constant when calculated on an area-normalized basis. Fixed parasitics are presented on the forth column and standby parasitics are

TABLE 6.52. Design-Point Parasitic Loss Characteristics for a Trough System

Plant Size, MWe	Energy Conversion Parasitic ^(a)		Field Parasitic, kW/m ²	Fixed Parasitic, kW	Standby Parasitic, kW
	a, kW	b, kWe/kWt			
0.5	4.4	0.0062	0.0005	6.4	9.7
2.0	16.0	0.0062	0.0005	24.1	31.0
10.0	73.0	0.0062	0.0005	116.0	163.0
30.0	216.0	0.0062	0.0005	345.0	484.0
100.0	705.0	0.0062	0.0005	1080.0	1559.0

(a) Parasitic power = $a + b * (\text{thermal input to the heat engine})$; where "a" is a constant electrical parasitic loss whenever the turbine is operating and " $b * (\text{thermal input to the heat engine})$ " is an electrical parasitic loss that varies with the thermal input to the heat engine.

described in column five. A more detailed discussion of the definition of the various components of parasitic losses is presented in Section 4.2.6.

6.6.6.2 Average Annual Plant Parasitics

The average annual parasitic power consumption is reported in Table 6.53 as a percent of plant gross output. The results show that parasitics tend to increase with increasing plant size. This increase is caused by increasing field transport pumping power associated with larger plant sizes.

TABLE 6.53. Average Annual Plant Parasitics for a Trough System

<u>Plant Size MWe</u>	<u>% of Gross Output Used for Parasitic Load</u>
0.5	10.4
2.0	9.3
10.0	10.0
30.0	12.7
100.0	15.5

7.0 COMPONENT COST RESULTS

This section of the report presents and discusses the results of the component cost analysis. Section 5.1 described the cost estimating approach used to develop the estimates presented here. This section follows the same general outline. Cost results are presented for concentrators, receivers, transport, storage, energy conversion, balance-of-plant, and O&M.

7.1 CONCENTRATOR COST RESULTS

Manufacturing cost estimates were developed for heliostat, dish, and trough concentrators. The design and cost bases for the analysis were ARCO's 150-m² heliostat, MDAC's 85-m² dish, and the Model 3011 trough by Acurex. Primary design information was obtained by visual inspection of the ARCO and MDAC concentrators. A previous manufacturing cost analysis had been completed for the trough (Reese et al. 1982). Supporting information was taken from second generation heliostat reports by ARCO, MDAC, Sandia-Livermore, and PNL.

Costs were estimated via the approach described in Section 5.1.1. Subcomponent costs are compared in Table 7.1. Part of the difference in costs per square meter of aperture area is due to the difference in the ratio of surface

TABLE 7.1. Concentrator Subcomponent Costs

<u>Subcomponent</u>	<u>Costs, 1984\$/m² Aperture</u>		
	<u>Heliostat</u>	<u>Dish</u>	<u>Trough</u>
mirror/support	56	61	85
foundation	12	9	16
transportation	3	7	4
drive	10	17	19
controls/wiring	6	9	10
field installation	16	16	13
<u>Total</u>	103	119	147
<u>Rounded Total</u>	100	120	150
<u>Long-term Goal</u>	100	140	110

area to aperture area for each concentrator. Table 7.2 shows that the dish and trough have 5% and 14%, respectively, more surface area per aperture area than the heliostat. Controls, wiring, and drive costs also contribute to the difference in unit cost among the three concentrators. Each of these three sub-components represents a fixed cost that is spread over a much larger aperture area for the heliostat. The normalized mirror/support cost shown in Table 7.2 for the dish is slightly higher than the cost for the heliostat, which seems reasonable considering the complexities of supporting a curved surface over a large area. However, the cost disadvantage of the trough for this subcomponent was much higher than expected. The cost problem for the trough can be traced to what may be an overly rigorous support structure on the Acurex trough. A preliminary cost analysis of the LUZ trough indicated that material costs for its support may as much as \$20 to \$30/m² less expensive.

7.2 RECEIVER COST RESULTS

Receiver costs were estimated for five different technologies: molten salt, sodium, and water/steam central receivers and dish and trough distributed receivers. A summary of receiver units costs as a function of concentrator field size is presented in Table 7.3. The figures show that unit costs are constant for the distributed (dish and trough) systems and decline with increasing field size for centralized systems. Overall, unit receiver costs for a 1,000,000-m² field are quite close, considering the extreme differences

TABLE 7.2. Normalized Concentrator Cost Data

<u>Attribute</u>	<u>Heliostat</u>	<u>Dish</u>	<u>Trough</u>
surface area per unit	148.64 m ²	91.42 m ²	89.19 m ²
aperture area per unit	148.64 m ²	86.86 m ²	78.04 m ²
surface area/aperture area	1	1.05	1.14
total cost/surface area	103	113	129
normalized cost/surface area	1	1.10	1.25
total mirror and support cost/ surface area	56	58	75
mirror and support normalized cost/surface area	1	1.04	1.33

TABLE 7.3. Receiver Costs for Various Field Sizes

Technology	Costs per m ² of Concentrator Aperture Area, 1984\$				
	10,000 m ²	30,000 m ²	100,000 m ²	300,000 m ²	1,000,000 m ²
CR-Salt	112	72	50	34	22
CR-NA	77	45	26	19	14
CR-W/S	82	63	53	37	28
dish	21	1	21	21	21
trough	14	14	14	14	14

Note: Central receiver costs do not include the tower.

in design represented by dish, trough, and central receiver systems. Subtle differences in the cost of the three central receivers are also identified in Table 7.3. Sodium receivers have the lowest unit cost for all field sizes, molten salt receivers are the most expensive at small field sizes, and water/steam receivers have the highest cost at large field sizes.

Reporting receiver costs on the basis of \$/m² of concentrator aperture area combines two factors: 1) the cost of the receiver per square foot of absorber area and 2) the number of square feet of absorber area required for a given concentrator field size. The former figure reflects the cost of the receiver as a function of the physical dimensions of the receiver, and the latter figure captures the effects of variable flux limits and overall thermo-optical efficiency. Separating these two factors helps explain the cost trends among the central receivers because both the cost per square foot of absorber area and the absorber area required per square meter of concentrator area vary.

Cost estimating equations for the three central receiver systems are shown below. For any given size of receiver absorber, the cost of a water/steam receiver is greater than the cost of a molten salt receiver, and both are more costly than the sodium receiver. The development of these receiver cost estimating models is discussed in more detail in the sections that follow.

$$\text{CR-Salt: Cost} = \$11,661 (\text{absorber area, ft}^2)^{0.75}$$

$$\text{CR-NA: Cost} = \$10,368 (\text{absorber area, ft}^2)^{0.75}$$

$$\text{CR-W/S: Cost} = \$12,240 (\text{absorber area, ft}^2)^{0.75}$$

Table 7.4 shows how the absorber area required per unit of concentrator area varies with field size and technology. The most significant trend is the advantage sodium receivers have because of (principally) their much higher flux capabilities. Except at small field sizes, molten salt and water/steam receivers require roughly twice as much absorber area as does a sodium receiver for a given field size. In summary, differences in the amount of absorber area required for a given field size contribute much more to differences in the receiver cost per square meter of concentrator area than does the unit cost of the absorber per square foot.

7.2.1 Central Receiver Cost Details

Cost modeling equations were developed by first estimating the cost of molten salt, sodium, and water/steam receivers at a single design point and then scaling this cost with absorber size. As discussed in Section 5.1.2, the design-point estimates were established by considering expected relative cost differences resulting from fundamental design differences and the absolute costs given in any single report. Tables 7.5, 7.6, and 7.7 identify the reports used as references for the central receiver cost analysis and some of the key design and cost data.

As indicated in Table 7.5, estimated unit costs for molten salt absorbers varied by a factor of five. The three lower estimates were developed by the same subcontractor. An estimate of \$800/ft² for Incoloy® absorber panels was derived by considering the three lower figures as one estimate and averaging it with the other two.

TABLE 7.4. Central Receiver Absorber Area Requirements

<u>Technology</u>	<u>Absorber Area (ft² of Absorber Area/103 m² Concentrator Area)</u>				
	<u>10,000 m²</u>	<u>30,000 m²</u>	<u>100,000 m²</u>	<u>300,000 m²</u>	<u>1,000,000 m²</u>
CR-Salt	39	33	32	30	26
CR-NA	31	21	15	14	15
CR-W/S	27	28	30	30	29

(a) Incoloy® is a trade name for Nickel-Iron-Chromium alloys, Huntington Alloys Inc., Huntington, West Virginia.

TABLE 7.5. CR-Salt Receiver Design and Cost Data

Source	Thermal Power (MWt)	Absorber Area (ft ²)	Costs, 1984\$/ft ² Absorber Area				
			Total	Absorber	Structure	Circulation	Instrumentation
Weber 1980	316	11,676	592	300	162	109	21
Weber 1982	181	7,301	782	353	272	123	34
Weber 1983	190	8,989	1715	1449	210	a	56
Joy et al. 1981	120	9,150	420	283	118	a	19
Easton and Endicott 1982	110	4,520	1376	725	370	245	36
South California Edison 1982	312	14,300	797	a	a	a	a

(a) The costs for these components were not explicitly specified.

TABLE 7.6. CR-Na Receiver Design and Cost Data

Source	Thermal Power (MWt)	Absorber Area (ft ²)	Costs, 1984\$/ft ² Absorber Area				
			Total	Absorber	Structure	Circulation	Instrumentation
Rockwell 1979	260	8,726	1259	671	49	241	298
Rockwell 1979	723	17,664	1374	622	45	430	277
Rockwell 1980	145	7,196	557	237	42	236	42
Brower et al 1979	240	8,656	1306	359	46	751	150
Bechtel 1985	258	6,852	1343	a	a	a	a
Bechtel 1985	107	2,600	1000	a	a	a	a

(a) The costs for these components were not explicitly specified.

TABLE 7.7. CR-W/S Receiver Design and Cost Data

Source	Thermal Power (MWt)	Absorber Area (ft ²)	Costs, 1984\$/ft ² Absorber Area				
			Total	Absorber	Structure	Circulation	Instrumentation
Black and Veatch 1981	37	1,254	4597	2415	1237	281	664
Black and Veatch 1980	73	3,267	3857	2218	1222	191	226
El Paso Electric 1980	92	4,460	2556	a	a	a	a
El Paso Electric 1982	141	4,235	3100	a	a	a	a
DeLaquil 1980	338	12,820	1505	1320 ^b	185 ^c	a	a
Darnell et al. 1985	388	20,000	2075	a	a	a	a

(a) The costs for these components were not explicitly specified.

(b) Includes circulation costs.

(c) Includes instrumentation and control costs.

Sodium receivers employ SS-316 tubes, which are less expensive than the Incoloy tubes used by the molten salt receiver. Absorber costs found in the literature (and listed in Table 7.6) reflected this difference. A unit cost of \$650/ft² was established for sodium absorbers, based on data from Rockwell (1979). Data from the Rockwell (1980) and Brower et al. (1979) reports were discounted because of various inconsistencies found in the data.

The water/steam receiver has carbon steel preheater panels, low-alloy steel boiler panels, and Incoloy superheater panels. The Incoloy panels were assumed to cost the same as the panels in the molten salt receiver. Low-alloy panels should cost less than SS-316 panels and were estimated to cost \$500/ft². Finally, the carbon steel panels should be the least expensive. They were estimated to cost \$400/ft². The screened tube design of the water/steam receiver actually requires 800 ft² of Incoloy, 200 ft² of carbon steel, and 340 ft² of low-alloy steel per 1000 ft² of projected surface area. Thus, the average cost is actually the highest of the three central receivers at \$900/ft².

Structural costs are directly related to receiver weight and external dimensions. For receivers with the same absorber area, the principal difference is the choice between cavity or external design. A cavity design (employed by the molten salt receiver) has greater structural requirements than an external design (employed by the sodium and water/steam receiver designs). Another differentiating factor is the 1/3 additional absorber area required for water/steam receivers for the same projected area. Relative structural costs between the cavity and two external receivers were established by comparing the receiver weight required per square foot of absorber area for the receiver designs referenced in Tables 7.5, 7.6, and 7.7. The result of this analysis yielded relative costs for sodium, water/steam, and molten salt receivers of 1, 1.3, and 2.3. Absolute structural costs were tied to the molten salt design specified in Weber (1983).

Relative circulation costs are affected by volumetric flow requirements, fluid design velocity, and fluid pressure. These factors all bear on the sizing of headers and manifolds. Flow requirements per MWt and for a 9000-ft² receiver are given in Table 7.8.

TABLE 7.8. Central Receiver Flow Requirements

<u>Technology</u>	<u>Flow Requirements per MWt</u>		<u>Requirements for 9000 ft² Absorber</u>	
	<u>lb/h</u>	<u>ft³/h</u>	<u>Thermal Power, MWt</u>	<u>Flow, ft³/h</u>
CR-Na	31,300	593	265	157,000
CR-Salt	18,000	159	188	30,000
CR-W/S	3,330	1780	219	390,000

The flow requirements calculated in Table 7.8 must be adjusted for relative design velocities to estimate the relative pipe size. Molten salt and sodium, both liquids, should have approximately the same design velocity. The steam velocity was assumed to be an order of magnitude higher, based on standard rule-of-thumb design guidelines. Adjusting for design velocity gives relative pipe diameters of 1, 5.2, and 1.3 for molten salt, sodium, and steam piping systems. Piping costs tend to scale with pipe diameter by a 0.6 exponent. Thus, the final expected relative circulation costs for molten salt, sodium, and steam are 1, 2.7, and 1.3, respectively.

Actual relative circulation costs compared very favorably with the predicted values. Estimates in the literature for molten salt and sodium system circulation costs averaged about \$100 and \$240 per ft² of absorber area, respectively, for absorber sizes around 9000 ft². No estimates of circulation costs were available for water/steam systems with absorber areas greater than 3300 ft². Unit circulation costs were estimated in two reports to be \$281/ft² at 1250 ft² and \$191/ft² at 3270 ft² of absorber area. The unit circulation cost for a 9000 ft² absorber was estimated to be \$150/ft².

Instrumentation and control costs varied by an order of magnitude among the different reports. Trends with receiver size were nebulous, and little justification seems to exist for differences among the technologies. Instrumentation and control costs were estimated to be \$750,000 for each central receiver at the 9000 ft² design point based on a simple average of the individual estimates from each report.

A summary of central receiver costs by subcomponent is shown in Table 7.9 for receivers with an absorber area of 9000 ft². Costs for receivers at other sizes were estimated by applying a scaling factor of 0.75, i.e., costs were

TABLE 7.9. Central Receiver Design-Point Costs

Component	Cost for a 9000 ft ² Absorber, 1984\$		
	CR-Salt	CR-Na	CR-W/S
Absorber	7,200,000	5,850,000	8,100,000
Structure	1,900,000	830,000	1,110,000
Circulation	925,000	2,150,000	1,350,000
Instrumentation	750,000	750,000	750,000
Total	10,775,000	9,580,000	11,310,000

assumed to be proportional to absorber area raised to the 0.75 power. The scaling factor of 0.75 was derived from examining the cost trends of similarly designed but different-sized solar receivers and historically proven factors for similar equipment such as boilers and furnaces.

Table 7.10 lists the data sources and summary design and cost information used for analysis of central receiver tower costs. Data were collected for steel and concrete towers for tower heights ranging from 120 to 1000 ft. Cost data from the reports were normalized to include direct costs for the tower, its foundation, and tower accessories in 1984 dollars. Separate tower cost estimating equations (see bottom of Table 7.10) were developed for towers less than 300 feet and towers over 300 feet.

7.2.2 Dish Receiver Cost Details

The receiver and energy conversion components are combined in a single power conversion unit for the dish. Three sources were found with estimated mass production costs for a model 4-95 Solar II Stirling engine. The sources and their cost estimates (adjusted to 1984\$) are shown in Table 7.11.

The data from each source in Table 7.11 were fit to an equation to estimate the cost at a production level equivalent to $2.5 \times 10^6 \text{ m}^2$ of concentrator aperture area or, for the dish, 28,782 units per year. An additional 10% was added to the United Stirling and JPL costs to allow for installation; the Vanguard estimate already included installation. The resulting three power conversion unit cost estimates were remarkably similar as shown in Table 7.12. The average cost of \$6,034 was rounded to \$6000 even. According to United Stirling, approximately 30% of the power conversion unit could be considered the receiver and the balance would fall into energy conversion. With a dish

TABLE 7.10. Central Receiver Tower Design and Cost Data

Source	Type	Height, ft	Cost, 10 ⁶ \$
Weber 1980	concrete	508	2.47
Weber 1982	concrete	394	2.24
Weber 1983	concrete	479	3.25
Easton and Endicott 1982	steel	394	1.49
Southern California Edison 1982	concrete	585	4.84
Joy et al. 1981	concrete	451	2.56
Darnell et al. 1985	concrete	555	5.20
Darnell et al. 1985	concrete	755	11.25
Rockwell 1980	concrete	464	3.07
Brower et al 1979	concrete	599	7.51
Rockwell 1979	concrete	516	3.26
Rockwell 1979	concrete	328	0.92
Rockwell 1979	concrete	656	5.53
Bechtel National 1985	concrete	528	3.90
Black and Veatch 1981	steel	244	0.46
El Paso Electric 1980	concrete	417	2.59
Darnell et al. 1985	concrete	495	3.70
Darnell et al. 1985	concrete	602	4.90
Bechtel 1982a	steel	235	0.90
Black and Veatch 1980	concrete	359	1.28
Stearns-Roger 1979	concrete	300	1.50
Stearns-Roger 1979	concrete	475	2.50
Stearns-Roger 1979	concrete	850	6.40
Stearns-Roger 1979	concrete	1000	13.50
Stearns-Roger 1979	steel	120	0.48
Stearns-Roger 1979	steel	210	0.60
Stearns-Roger 1979	steel	300	1.00

For towers less than 300 feet, but greater than 100 feet:
Cost = \$226,080 Exp(0.005205 * height (ft))

For towers greater than 300 feet, but less than 1000 feet:
Cost = \$4.93 * (height [ft])^{2.158}

TABLE 7.11. Dish Power Conversion Unit Cost Data

<u>Source</u>	<u>Production Quantity</u>	<u>Cost/Unit</u>
United Stirling ^(a)	1	\$50,000
	2,000	\$20,000
	25,000	\$ 5,500
JPL (Fortgang and Mayers 1980)	1,000	\$17,576
	25,000	\$ 5,641
	100,000	\$ 2,946
	400,000	\$ 2,539
Vanguard (Washom 1984)	1	\$57,878
	100	\$28,941
	1,000	\$15,463
	10,000	\$ 8,700

(a) Telephone conversation with Worth Percival, United Sterling.

TABLE 7.12. Dish Power Conversion Unit Cost Estimates (Costs estimated in 1984\$ for 28,782 units per year)

<u>Source</u>	<u>Cost, \$/Unit</u>
United Stirling ^(a)	5,630
JPL (Fortgang and Mayers 1980)	5,926
Vanguard (Washom 1984)	6,546
Average	6,034
Rounded Average	6,000

(a) Telephone conversation with Worth Percival,
United Stirling

concentrator aperture area of 86.86 m^2 and a Stirling engine power rating of 25 kWe, the unit production costs of the dish receiver and heat engine were estimated to be \$21/m² and \$170/kWe, respectively.

Even though the three estimates of the dish power conversion unit cost were found to be in close agreement, some caution is advised. None of the power conversion unit manufacturing cost estimates had been completed in near

the detail that has been done for heliostats, for example. Yet, the viability of the dish concept hinges on the cost and performance of this component.

7.2.3 Trough Receiver Cost Details

The trough receiver cost analysis was done in concert with the trough concentrator manufacturing cost analysis. Manufacturing cost analyses completed by Acurex (Reese 1982) and the Central Solar Energy Research Corporation (1981) served as the primary bases for material, labor, and factory capital requirements and costs. A summary of the trough receiver cost estimate is shown in Table 7.13. The cost of \$14/m² does not include the flexhose required to link one trough drive module to the next or to link up with the stationary transport piping. The cost of flexhose was included within the transport component. If flexhose were included in the receiver capital account, the receiver cost would approximately double.

7.3 TRANSPORT COST RESULTS

Transport costs were estimated from designs developed at PNL; material and installation costs were acquired from vendors, cost estimating manuals, and PNL cost data bases. Unit costs are presented in Table 7.14 as a function of field size for each of the six systems evaluated.

Overall, centralized transport is estimated to be less expensive than distributed transport. Distributed thermal transport was estimated to be 2 to 4 times more costly than centralized thermal transport, and distributed electrical transport is about the same as central thermal.

TABLE 7.13. Trough Receiver Component Costs (Costs in 1984\$ per m² of concentrator aperture area)

<u>Component</u>	<u>Cost, \$/m²</u>
Factory	
direct materials	7.50
other factory	4.00
Transportation	0.50
Installation	2.00
Total	14.00

TABLE 7.14. Transport Costs for Various Field Sizes

Technology	Cost per m ² of Concentrator Aperature Area 1984\$				
	10 ⁴ m ² field	3 x 10 ⁴ m ² field	3 x 10 ⁵ m ² field	3 x 10 ⁵ m ² field	10 ⁶ m ² field
CR-Salt	18	12	11	11	13
CR-Na	18	13	11	13	14
CR-Na/Salt	27	18	15	15	14
CR-W/S	13	9	8	8	9
Trough	36	35	40	45	56
Dish	16	16	16	16	17

Central thermal transport costs (per m²) initially decline with increasing field size and begin to rise as field size approaches one million square meters. The driving forces behind this trend are piping economies of scale and pumping diseconomies of scale (as the pumping head up the tower grows). The former dominates for relatively small piping systems, and the latter becomes more important for larger systems.

Increasing the field size for trough systems requires that headers and mains be increased in both length and diameter and that laterals be added to directly feed the collectors. Both piping and pumping capital costs increase at a rate faster than the increase in field size. Hence, trough system transport costs per unit increase with increasing scale. Dish (electric) transport costs also rise with field size, but the rate of increase is almost negligible.

Transport costs for the water/steam system are lower than the other central receiver systems because this system has less ancillary piping, such as drain-down lines. The CR-Na/salt system benefits from a closed sodium loop that allows recovery of the pressure head going up the tower. This advantage is counterbalanced by the extra cost of the sodium/salt heat exchanger, however. The CR-Salt and CR-Na systems have very similar total transport costs, but quite different cost subcomponents. Pumps are more expensive for the taller towered molten salt systems, but piping is less expensive because there is less fluid to pump per MWt.

Transport costs estimated for other central receiver analyses are listed in Table 7.15. Cost estimates ranged from \$4 to 67/m² and averaged \$21/m² for

TABLE 7.15. Central Receiver Transport Design and Cost (Costs in 1984\$/m² of Concentrator Aperture Area)

Source	Transport/Fluid	Plant Type	Unit Cost
Rockwell 1983	sodium	stand alone	66
Bechtel National 1985	salt	stand alone	8
Bechtel National 1985	salt	stand alone	8
Bechtel National 1985	sodium	stand alone	5
Bechtel National 1985	sodium	stand alone	4
Rockwell 1980	sodium	repowering	40
Darnell et al. 1985	water/steam	stand alone	4
El Paso Electric 1982	water/steam	repowering	19
Southern California Edison 1982	salt	stand alone	13
Weber 1980	salt	repowering	10
Weber 1983	salt	repowering	48
Rockwell 1979	sodium	stand alone	8
Rockwell 1979	sodium	stand alone	11
Brower et al. 1979	sodium	stand alone	10
Weber 1982	salt	repowering	22
Easton and Endicott 1982	salt	repowering	38
Black and Veatch 1982	water/steam	repowering	17
Black and Veatch 1981	water/steam	repowering	9
Bechtel 1982a	water/steam	repowering	67
Overall average			21
Stand-alone average			14
Repowering average			30

all of the sources reviewed. If stand-alone plants are considered by themselves, the average cost drops to \$14/m². The average cost of repowering plants, on the other hand, was \$30/m². This differential meets with expectations since the transport path from the base of the tower to the energy conversion unit is often more lengthy in a repowering situation.

7.4 ENERGY STORAGE COST RESULTS

Energy storage costs were estimated for molten salt, sodium, and oil/rock thermal systems and a lead/acid battery electrical system. Subcomponent costs were derived from a variety of sources, including system design reports with an analysis of the storage component and other reports that evaluated several storage technologies. The principal sources of subcomponent cost data are identified in Table 7.16. Every attempt was made to use the same unit cost for subcomponents common to more than one technology. For example, all carbon steel tank costs were based on the same cost estimating equation.

Unit storage costs are shown in Table 7.17 as a function of storage capacity. All technologies benefit from economies of scale, especially the thermal systems. Economies-of-scale are captured in the containment equipment of thermal systems, but are limited to the ancillary equipment for batteries. Differences in economies of scale among thermal systems stem from the relative portion of costs in containment (which has declining unit costs with capacity) and media (which has a constant unit cost with capacity). Media costs for sodium, nitrate salt, and oil/rock storage systems are \$11.14, \$4.15, and \$3.44

TABLE 7.16. Data Sources for Energy Storage Cost Analysis

<u>Source</u>	<u>Storage Technology</u>
Brower et al. 1979	sodium
Rockwell 1979	sodium
Weber 1980	salt
Weber 1982	salt
Weber 1983	salt
Honeywell 1977	oil/rock
Stearns-Roger 1981	several
Babcock and Wilcox 1981	several
Krauthamer and Frank 1980	batteries
Bechtel Group 1982	batteries
Stolte 1982	batteries
Ramsay and Edgecomb 1982	batteries
Abraham 1985	batteries

TABLE 7.17. Storage Cost Summary (1984\$/kWht (or kWhe) of storage capacity)

<u>Technology</u>	<u>Storage Capacity, MWht</u>					
	<u>10</u>	<u>30</u>	<u>100</u>	<u>300</u>	<u>1000</u>	<u>3000</u>
sodium	71	40	29	22	19	18
salt	88	43	25	18	13	12
oil/rock/trough	62	33	18	10	8	8
oil/rock/CR ^(a)	79	49	26	17	14	15

	<u>Storage Capacity, MWhe</u>					
	<u>3</u>	<u>10</u>	<u>30</u>	<u>100</u>	<u>300</u>	<u>1000</u>
lead-acid battery	248	213	181	137	130	129

(a) Includes charging and discharging heat exchangers

per kWht, respectively. Economies of scale are strongest for oil/rock storage, which has the cheapest storage media, and weakest (among the thermal systems) for sodium, which has the most expensive storage media.

Containment materials and storage system design are also critical factors affecting the relative costs of the different storage systems. Some differences have an obvious effect. For example, carbon steel tanks used for oil/rock systems are less expensive than the stainless steel tanks required for sodium storage. Less obvious is the affect of alternative design conditions such as the media cycle temperature. Salt systems are commonly cycled from 550 to 1050°F and sodium systems from 550 to 1100°F. Oil/rock designs are more variable, however. In one report (Stearns-Roger 1981), the direct cost of an oil/rock storage system was estimated to be \$26/kWht at 1300 MWht based on a media cycle temperature of 45°F. This figure is equivalent to the PNL-estimated cost of \$8/kWht based on a media cycle temperature of 148°F. The physical equipment and total cost for the two analyses is not different; only the utilization and thermal capacity has changed.

An examination of the unit media costs and tank sizes for each of the three thermal systems (see Table 7.18) provides additional understanding of the relative storage costs. Based on the data in Table 7.18, one might expect salt storage systems to be the least expensive because of its relatively low media

TABLE 7.18. Thermal Storage Unit Media Costs and Tank Size Requirements

<u>Technology</u>	<u>Media Cost, \$/kWht</u>	<u>Tank Size, ft³/kWht</u>
salt	4.15	0.173
sodium	11.14	0.391
oil/rock	3.44	0.702

cost and significantly smaller tank size requirements. There are several mitigating factors, however. First, doubling the physical size of the containment does not double its cost. Because of the economies of scale noted above, a doubling of tank size (measured in gallons, for example) might only increase the total cost by 50%. Differences in design and material requirements also mitigated the size advantage enjoyed by molten salt. Oil/rock tanks, although the largest for a given thermal capacity, require less insulation, less expensive insulation, and use tanks made of carbon steel. Sodium storage employs an externally insulated stainless steel tank, while the molten salt system employs the most expensive containment of all, an internally and externally insulated carbon steel tank with an Incoloy liner.

Storage costs estimated for other central receiver analyses are listed in Table 7.19. PNL-estimated costs fall in the middle of the range of costs estimated for salt systems and toward the lower end of the range of costs estimated for sodium systems.

7.5 ENERGY CONVERSION COST RESULTS

Unique cost estimates were developed for five different energy conversion systems. The central receiver systems and the trough system employ steam Rankine technology and are discussed here. The Stirling engine for the dish system is integrated with the receiver into a power conversion unit. The dish receiver costs were presented in Section 7.2.2.

For cost analysis purposes, the energy conversion system was broken into steam generator, turbine-generator, and balance-of-turbine cycle components. The latter category includes the condenser, cooling tower, feedwater heaters,

TABLE 7.19. Central Receiver Storage Design and Cost

<u>Source</u>	<u>Technology</u>	<u>Capacity, MWht</u>	<u>Cost, 1984\$/kWht (a)</u>
Rockwell 1983	sodium	125	36
Bechtel National 1985	salt	315	27
Bechtel National 1985	salt	1258	20
Rockwell 1980	sodium	684	18
Southern California Edison 1982	salt	2857	10
Weber 1980	salt	1166	8
Weber 1983	salt	688	21
Rockwell 1979	sodium	805	21
Rockwell 1979	sodium	2350	19
Brower et al. 1979	sodium	766	37
Weber 1982	salt	688	10

and other ancillary equipment. Because steam conditions for each of the central receiver systems are nearly identical, the cost-differentiating aspects of the turbine generator and ancillary equipment are close to nil. Significant differences do exist in the design and cost of the steam generators. Costs were estimated from preliminary design data developed for preheaters, boilers, superheaters, and 100 MWe central receiver system reheaters. Steam generator costs are summarized in Table 7.20 for each of the Rankine systems.

Differences in steam generator costs at a given power level reflect the heat transfer capabilities of the different working fluids, energy conversion efficiency, steam conditions, and design approach temperatures. In general, the heat transfer coefficient for sodium is greater than for salt. Oil has the lowest coefficient. Heat exchanger size and cost are inversely proportional to the working fluid's heat transfer capabilities. The poor conversion efficiency of the trough (and for steam generated from oil for the water/steam central receiver) adds to its costs by requiring more energy to be transferred for a given power level and, hence, a larger, more costly steam generator. Note that the steam generators for the water/steam central receiver system have been sized to produce less than the rated power of the turbine generator at each

TABLE 7.20. Rankine System Steam Generator Costs for Various Power Levels

System	Cost, thousands of 1984\$				
	0.5 MWe	2.0 MWe	10.0 MWe	30.0 MWe	100.0 MWe
CR-W/S ^(a)	122	266	885	2473	10,750
CR-Salt ^(b)	122	316	947	2286	7,320
CR-Na	108	253	694	1597	5,046
Trough	143	331	1192	3367	10,900

- (a) The water/steam central receiver requires a steam generator when storage is used and a reheater at 100 MWe.
- (b) These steam generator costs apply to the molten salt receiver/molten salt storage and sodium receiver/molten salt storage systems.

design point. These generators are therefore smaller and less costly than the steam generators for the trough system. At 100 MWe, the inclusion of a reheater for the central receiver system increases the total steam generator cost to nearly that for the trough system steam generator. Tighter approach temperatures for the oil steam generators allow slightly higher steam temperatures and better conversion efficiency, but they also contribute to bigger and more expensive heat exchangers. Sodium systems benefit from a higher approach temperature than either the molten salt or oil systems. Oil steam generators operate at lower pressure (and therefore with less expensive materials), but this savings is negated by the factors discussed above.

Turbine generator cost data were acquired from several sources, including vendor inquiries and past solar thermal systems analyses. Overall costs vary slightly at a given power level within the range of steam conditions considered. Lowering the temperature and pressure from a given design point creates two opposing cost drivers for steam turbines. On the one hand, lower temperatures and pressures generally reduce the use of expensive alloys and thick-walled pressure bearing components. On the other hand, lower-temperature and -pressure steam has a much lower energy density. Combined with the lower conversion efficiency, the steam requires a much higher volumetric throughput

handling machine. The two effects combine to yield costs for the trough turbine generator that are approximately 10% higher than those for the central receivers. For the water steam turbine generator at 100 MWe, the reduced temperature requirements yield a cost reduction of about 1%.

Empirical evidence from several studies (Rockwell 1979, Brower et al. 1979, and DOE 1985) indicated that the cost of balance-of-system components for turbine generators was approximately equal to the turbine generator alone. This assumption was applied to all of the systems at each power level. Total energy conversion costs are shown as a function of power level for the four different Rankine systems in Table 7.21.

Turbine generator costs estimated for other central receiver analyses are quite similar to the PNL-estimated costs shown in Table 7.22. Costs estimated by others for steam generators, however, show much more variability, as indicated in Table 7.23.

TABLE 7.21. Rankine Energy Conversion Cost Summary (1984\$ per kWe)

System	Cost per KWe, 1984\$				
	0.5 MWe	2.0 MWe	10.0 MWe	30.0 MWe	100.0 MWe
CR-W/S	1604	1215	689	483	414
CR-Salt	1602	1240	695	477	383
CR-Na	1576	1205	669	453	360
trough	1742	1320	761	540	447

TABLE 7.22. Central Receiver Turbine Generator Costs

Source	Size, MWe	Unit Cost 1984\$, kWe
Rockwell 1979	100	161
Brower et al. 1979	100	156
Rockwell 1983	30	200
Southern California		
Edison 1982	100	160
PNL	100	155
PNL	30	200

TABLE 7.23. Central Receiver Steam Generator Costs

<u>Source</u>	<u>Fluid</u>	<u>Size, MWe</u>	<u>Unit Cost, 1984\$/kWe</u>
Rockwell 1979	sodium	100	61
Rockwell 1979	sodium	281	42
Rockwell 1980	sodium	60	142
Brower et al. 1979	sodium	100	127
Rockwell 1983	sodium	30	240
Bechtel National 1985	salt	50	208
Joy et al. 1981	salt	20	250
Southern California			
Edison 1982	salt	100	70
Weber 1980	salt	120	42
Weber 1983	salt	66	153

7.6 BALANCE-OF-PLANT COST RESULTS

The balance-of-plant category is a direct capital cost account that includes support equipment necessary for plant operation that is not included in the capital cost components described above. The balance-of-plant category was broken into the following six subcomponents for this study: land and site preparation, master control, structures, power conditioning, spare parts, and service facilities. Costs for four of these subcomponents (all but master control and power conditioning) are aggregated from individual pieces of equipment or site requirements.

Several balance-of-plant subcomponents yield different costs among the solar technologies evaluated, while other subcomponent costs remain constant from one system to the next. Structures and power conditioning costs vary with plant size but are identical for all of the systems. Trough site preparation requires a finer grade to ensure a level plane, so the cost per square meter for this subcomponent is slightly higher. Master control costs are the same for each of the Rankine systems; dish system master control is less expensive. Spare parts are estimated as a fraction of initial capital, with the fraction varying depending on the centralized or distributed nature of the component.

Centralized components (e.g., central receivers, Rankine energy conversion) were assumed to require a higher fraction of initial spares because their breakdown would be more critical than a failure in a distributed component (e.g., concentrator, dish or trough receiver). Finally, some differentiation exists for service facilities, which includes the Rankine system make-up water treatment and maintenance vehicles and other items (communication, security, sanitary water) identical to all systems. The dish system does not require make-up water treatment, of course, but does require more service vehicles for maintaining the Stirling engines. Trough maintenance does not require a "cherry picker" for elevated access, so its service facilities costs are slightly lower.

Unit balance-of-plant costs are summarized in Table 7.24 for each of the six systems over a range of field sizes. Overall, each technology shows economy of scale with field size. At each field size, central receiver systems are more expensive than the dish system. The central receiver systems and the dish system are all more expensive than the trough system. Differences among the technologies are highlighted in Table 7.25, where unit costs are shown for

TABLE 7.24. Balance-of-Plant Costs for Various Field Sizes

System	Cost Per Square Meter of Concentrator Aperature Area, 1984\$				
	10,000 m ²	30,000 m ²	100,000 m ²	300,000 m ²	1,000,000 m ²
CR-Salt	82	57	47	38	33
CR-Na	80	56	46	37	32
CR-Na/Salt	80	56	46	37	32
CR-W/S	83	58	47	38	33
dish	76	50	40	30	26
trough	55	38	31	23	19

TABLE 7.25. Balance-of-Plant Subcomponent Costs

System	Cost Per Square Meter of Concentrator Aperature Area, 1984\$					
	Land	Structures	Master Control	Power Condition	Service Facilities	Spare Parts
CR-Salt	24.8	4.1	6.3	3.9	1.8	5.7
CR-Na	24.8	4.1	6.3	3.9	1.8	4.5
CR-Na/Salt	24.8	4.1	6.3	3.9	1.8	4.6
CR-W/S	24.8	4.1	6.3	3.9	1.8	6.0
dish	16.4	4.1	2.1	3.9	2.1	4.7
trough	10.5	4.1	6.3	3.9	1.6	5.5

a 10 MWe plant with a 100,000 m² field. Central receiver balance-of-plant costs are identical for each system, except that the cost of spare parts was estimated as a fraction of initial capital. The biggest difference between central receiver, dish, and trough technologies lies with land and site preparation. The relative land costs of the three systems primarily reflect their difference in ground cover ratio or concentrator packing density. Additional differences are attributable to master control, service facilities, and spare parts.

An alternative view of balance-of-plant costs is seen by comparing unit costs as a function of power level with capacity factor fixed (at 0.4) and field size allowed to vary. This format is shown in Table 7.26. Most of the trends discussed above are still apparent, but the impact of variable field and conversion efficiency on unit balance-of-plant cost is incorporated. The less efficient systems often have lower unit costs because some of the fixed portions of the balance-of-plant costs are spread over a larger field size.

Balance-of-plant costs estimated for other central receiver analyses are shown in Table 7.27. Unit cost estimates ranged from \$10 to 65/m² for

TABLE 7.26. Balance-of-Plant Costs for Various Power Levels

System	Cost Per Square Meter of Concentrator Aperture Area, 1984\$				
	0.5 MWe	2.0 MWe	10.0 MWe	30.0 MWe	100.0 MWe
CR-Salt	101 (7)(a)	71 (20)	52 (81)	43 (222)	36 (696)
CR-Na	88 (8)	65 (23)	49 (88)	41 (239)	34 (728)
CR-Na/Salt	85 (9)	66 (22)	49 (88)	42 (224)	36 (685)
CR-W/S	87 (8)	59 (27)	46 (102)	41 (275)	33 (904)
dish	132 (3)	66 (13)	42 (68)	33 (206)	28 (694)
trough	55 (10)	34 (35)	24 (160)	19 (486)	17 (1652)

(a) Numbers in parentheses are field sizes given in thousands of square meters.

TABLE 7.27. Central Receiver Balance-of-Plant Costs

<u>Source</u>	<u>Working Fluid</u>	<u>Cost, 1984\$/m²</u>	<u>Field Size 10,000 m²</u>
Rockwell 1983	sodium	48	180
Bechtel National 1985	salt	37	300
Bechtel National 1985	salt	24	630
Bechtel National 1985	sodium	38	310
Bechtel National 1985	sodium	24	640
Rockwell 1980	sodium	20	390
Darnell et al. 1985	water/steam	14	700
El Paso Electric 1980	water/steam	62	210
Joy et al. 1981	salt	33	190
Southern California Edison 1982	salt	41	880
Weber 1980	salt	18	515
Weber 1983	salt	48	284
Rockwell 1979	sodium	19	692
Rockwell 1979	sodium	10	1990
Brower et al. 1979	sodium	20	1123
Weber 1982	salt	24	287
Easton and Endicott 1982	salt	33	146
Black and Veatch 1981	water/steam	35	111
Black and Veatch 1981	water/steam	65	56
Bechtel 1982a	water/steam	37	54

heliostat field sizes ranging from about 50,000 to 2,000,000 m². Average figures for unit cost and field size were \$33/m² and 500,000 m², respectively. Costs estimated by PNL fall in the middle of this range for concentrator fields of comparable size.

7.7 OPERATION AND MAINTENANCE COST RESULTS

Operation and maintenance cost estimates were aggregated from operating labor, maintenance labor and materials, and overhead labor subcomponents. Operation and maintenance costs as a function of concentrator field size and technology are summarized in Table 7.28.

Several significant trends are evident in Table 7.28. Trough and central receiver systems show strong economies of scale, but dish O&M costs are

TABLE 7.28. Summary of Operation and Maintenance Costs

System	Cost Per Square Meter of Concentrator Aperature Area, 1984\$				
	10,000 m ²	30,000 m ²	100,000 m ²	300,000 m ²	1,000,000 m ²
CR-Salt	48	19	9	6	5
CR-Na	48	19	9	6	5
CR-Na/Salt	48	19	9	6	5
CR-W/S	48	19	9	6	5
trough	50	20	10	7	6
dish	27	24	22	22	22

relatively constant. Dish O&M is the most expensive for all but the smallest field sizes. Trough O&M cost consistently parallels central receiver O&M cost at a slightly higher level.

Operating labor requirements vary little with plant size and can be thought of as fixed costs. The unit cost (\$/m²) declines as the plant (field) size increases. This relationship accounts for most of the economies of scale seen in Table 7.28 for trough and central receiver systems. As will be discussed in more detail below, operating labor cost for dish systems is much less than for either trough or central receiver systems, which partly explains why dish system O&M shows little economy of scale.

Maintenance labor and materials also contribute to economies of scale in an indirect manner. Maintenance labor and material costs are estimated as a fraction of initial capital cost. Components with capital cost economies of scale resulting from field size (centralized) also show O&M economies of scale.

Estimates of the required staffing for operating labor may contain the most uncertainty of any cost subcomponent. As shown in Table 7.29, estimates of staffing found in published reports vary substantially from the actual requirements of currently operating facilities. Very few estimates exist for system sizes less than 10 MWe, and essentially no estimates were found for trough and dish systems.

In general, two operators per shift was seen as the minimum for steam Rankine systems. Legal requirements exist in some localities for operational

TABLE 7.29. Estimated Operating Labor Requirements
Other Sources and Operations

<u>Source</u>	<u>Plant Size, MWe</u>	<u>FTE Staff</u>
Rockwell 1983	30	16
Southern Cal Edison 1982	100	41
El Paso Electric 1980	60	4
Joy et al. 1981	40	12
Weber 1980	120	10
Brower et al. 1979	80	50
Weber 1982	60	13
Black and Veatch 1981	12	1
Black and Veatch 1980	24	2.5
Bechtel 1982a	11	4
Barstow ^(a)	10	16 (est)
LUZ ^(a)	13	5 (est)
ARCO (Taft) ^(a)	0.3	1 (est)

(a) Information acquired in site visit.

attendance to monitor the boiler and turbine. Increasing the number of operators for larger plant sizes will probably be influenced by system economics as much as by technical requirements. Operating labor costs can represent a significant fraction of the LEC for small facilities. Companies may choose to operate with a skeleton crew and live with the consequences of periodic malfunctions. Larger facilities are more likely to be able to afford the investment in extra staff to watch over the more expensive equipment.

Dish systems, not having a boiler or central energy conversion unit, were assumed to be much more amenable to fully automated operations. Dish system operational labor was restricted to evening and weekend security. A summary of operational labor assumptions is presented in Table 7.30.

Total operating labor costs were calculated by multiplying the number of full time equivalent positions by the number of working hours in a year and the burdened hourly labor rate. Two operational positions were considered:

TABLE 7.30. Operating Labor Staffing Levels for Various Power Levels

<u>System</u>	<u>Number of Staff Positions (full time equivalents)</u>				
	<u>0.5 MWe</u>	<u>2.0 MWe</u>	<u>10.0 MWe</u>	<u>30.0 MWe</u>	<u>100.0 MWe</u>
CR	7.51	7.5	7.5	10.5	13.5
Trough	7.5	7.5	7.5	7.5	10.5
Dish	1.5	1.5	1.5	1.5	1.5

operators and security. A review of labor charges from several solar thermal design reports yielded average burdened labor rates of \$22.50/h and \$18.50/h for operators and security personnel, respectively.

As noted above, maintenance labor and materials costs were estimated as fractions of initial capital cost. The fractions employed for each of the components are listed in Table 7.31. Annual maintenance for most items runs from

TABLE 7.31. Maintenance Labor and Material Costs Expressed as Fractions of Initial Capital Cost for Central Receiver Systems

<u>Component</u>	<u>Fraction of Initial Cost</u>
heliostat	0.0164
dish concentrator	0.01825
trough concentrator	0.0158
central receivers	0.02-0.04
dish receiver	0.19
trough receiver	0.04
oil transport	0.024
other thermal transport	0.016
electric transport	0.0075-0.0095
oil/rock storage	0.024
other thermal storage	0.016
battery storage	0.0075-0.0095
Rankine conversion	0.01
Stirling conversion	0.23
balance of plant	0.014-0.016

1 to 4% of capital cost. The notable exceptions are maintenance for the dish receiver and for the heat engine. The Stirling cycle energy conversion unit is notoriously unreliable compared to the other components and requires extensive and frequent maintenance to keep it operational. The level of annual maintenance expenditures indicated in Table 7.31 is expected to allow the Stirling power conversion unit to last indefinitely, rather than estimating a lower level of annual maintenance and periodic replacement of the entire unit.

The high maintenance charges for the dish receiver and heat engine is the reason why the total O&M estimate for dishes exceeds that for other systems at most field sizes (see Table 7.28). Few economies of scale exist for dish system components because operating labor never represents a large part of the total O&M cost and maintenance labor for most of the dish system components is directly proportional to field size.

Uncertainty in O&M costs was also recognized by the wide variance in costs estimated by others for central receiver systems (see Table 7.31). PNL-estimated costs (for systems of comparable field size) fall in the middle of the range identified in Table 7.32.

TABLE 7.32. Central Receiver Operation and Maintenance Costs

<u>Source</u>	<u>Cost, 1984 \$/m²</u>	<u>Field Size, 10,000 m²</u>
APS 1983	6.40	280
APS 1980	3.00	515
Exxon (Easton and Endicott 1982)	15.90	190
Newman (El Paso Electric 1980)	14.90	210
Solar 100 (Southern Cal Edison 1982)	8.00	880
Bechtel/EPRI	12.60	680
Paint Creek (Rockwell 1980)	1.70	390
Bechtel/PG&E	13.70	300
Bechtel/PG&E	8.30	630
Carrisa Plains (Rockwell 1983)	17.30	180
Rockwell 1979	2.90	690
Rockwell 1979	2.00	2350
General Electric (Brower et al. 1979)	6.10	1120
APS 1982	5.40	290
Northeastern (Black and Veatch 1980)	2.70	110
Pioneer Mill (Bechtel 1982)	9.70	54
Cimarron (Black and Veatch 1981)	1.00	55
Average	7.70	525
Range	1-17	54-2350

8.0 SYSTEM RESULTS

This chapter presents overall system results for each concept at various plant sizes and capacity factors. Definitions of plant design points and the way design points were selected are discussed in Section 8.1. Section 8.2 describes annual system efficiencies, and plant costs (both capital and O&M) are examined in Section 8.3. Levelized energy costs are presented and discussed in Section 8.4. Comparison of the results of this study to long-term goals established for solar thermal technology is discussed in Section 8.5.

8.1 PLANT DESIGN POINTS

Design-point information for a solar thermal plant specifies the physical features of the plant (such as concentrator field size), the output characteristics of the plant (such as the plant capacity), and the meteorological assumptions on which the output characteristics are based. Many studies have applied different definitions to plant design-point specifications. The purpose of this section is to explain the design-point definitions used in this report so that the system results can be properly interpreted.

The plant power rating is defined as the maximum electrical output the plant will supply to the utility grid. The plant power rating is therefore not the same as the nominal capacity of the heat engine. For central engine plants (all concepts except the dish system), the heat engine capacity is slightly greater than the plant power rating, so the plant can supply full rated power to the grid and also supply the parasitic power requirements for the plant. For the dish concept, which uses distributed engines, the installed heat engine capacity varies proportionally with collector field size. As the collector field size is varied by SOLSTEP during the plant optimization, the heat engine capacity is also varied. The result is that the total installed capacity of the heat engines may be substantially different from the plant power rating for dish systems, depending on results of the optimization.

The plant capacity factor is a dimensionless variable that provides a representation of the total energy produced by the solar plant during the year. The definition of capacity factor used in this study is:

$$\text{Capacity Factor} = \frac{\text{Annual Plant Electric Output (kWh)}}{\text{Plant Rating (kW)} * \text{Hours in Analysis Period (h)}} \quad (8.1)$$

Capacity factors reported include allowances for scheduled and unscheduled maintenance, both of which reduce the capacity factor that would be predicted based on 100 percent availability.

As discussed in Chapter 4.0, the optimal balance between the collector field size and storage capacity for a given plant rating and plant capacity factor is determined by the analysis code SOLSTEP. Rather than basing the plant design on an arbitrary design point (such as producing rated power at a specific insolation level), SOLSTEP analyzes a number of alternative field sizes and storage capacities to determine the LEC for each combination. The code then selects the plant designs that result in the minimum LEC for each plant capacity factor specified by the user.

In addition to selecting optimal plant designs for specific plant capacity factors, SOLSTEP also reports two special case design points. One of these cases is the no-storage plant design, which is the plant design without energy storage that resulted in the lowest LEC during the analysis. The no-storage design is not based on a specific solar multiple, but is selected on the basis of overall plant economics. The capacity factor that corresponds to no-storage designs varies among concepts because the economic and performance characteristics of each concept are different. The capacity factor for no-storage designs can also vary for a single concept at different plant power ratings since the performance and cost characteristics being optimized for the small plants can vary significantly from the characteristics for large plants.

The second special case design point reported by SOLSTEP is the optimal LEC case. This design point is the plant design with the lowest LEC, regardless of the plant capacity factor. As with the no-storage design points, the capacity factor corresponding to the optimal LEC case varies significantly among the concepts and can also vary somewhat with power level within an individual concept. The optimal LEC case is not necessarily a global optimum, but only the lowest energy cost for plant sizes between 0.5 to 100 MWe and 0.25 to 0.6 capacity factor (the scope of this study).

No single LEC result provides a complete picture of the economic performance of one concept relative to another, therefore care must be used in interpreting the results of this study. The concept with the lowest LEC at large plant sizes and large capacity factors may not be competitive with other concepts at smaller plant sizes. Ultimately, the application being considered (small or large plants, peaking or intermediate-load) will determine the point at which to compare LEC among concepts.

All of the component and system performance results in this report are based on annual simulations (i.e. performance simulated over an entire year) unless specifically stated otherwise. Weather data for Barstow, California, in 1976 was used in the simulation.

8.2 SYSTEM PERFORMANCE

This section discusses overall system performance results. Section 8.2.1 presents an overview of system performance for the six concepts. Section 8.2.2 includes a more detailed description of system performance presented as efficiency "water fall" bar charts.

8.2.1 Overall System Efficiency

This section presents plant annual efficiency results. The plant annual efficiency calculated by SOLSTEP is an operating efficiency. This calculation assumes that the plant is available for operating during every time step in the analysis period. Allowances for plant down-time due to scheduled or unscheduled maintenance are accounted for in the LEC, but are not included in the calculation of annual efficiency. The annual efficiency is defined as:

$$\text{Annual System Efficiency} = \frac{AI * CA}{AEO} \quad (8.2)$$

where AI = Annual direct normal insolation, kWh/m²

CA = Concentrator aperture area, m²

AEO = Plant annual energy output assuming 100% availability, kWh

Annual system efficiencies for no-storage cases for all the six concepts are summarized in Figure 8.1. For no-storage plants, the dish concept consistently has the highest system efficiency for all plant sizes. The trough system consistently has the lowest system efficiency, with the central receiver concepts being between the dish and trough systems.

One of the most interesting aspects of the system efficiency results is the decrease in the system efficiency for the dish concept at the 0.5 MWe size. If the dish system were designed with consideration of efficiency only, the plant efficiency would be effectively constant at all power levels. The decrease shown in Figure 8.1 occurs not because of design aspects that limit the efficiency but because of economic tradeoffs in the system design. High fixed costs in plant construction at the 0.5 MWe size cause the most cost effective dish system to be one with a larger relative (in terms of m^2/MWe) collector field size than for the larger dish plants. Although the larger

FIGURE 8.1. System Efficiency of No-Storage Plants

relative collector field size of the 0.5 MWe plant gives it a larger capacity factor, the annual efficiency is lower than that of the larger plants because there are considerable periods during the year when the plant cannot dispatch all energy to the grid.

Of the various central receiver concepts shown in Figure 8.1, the CR-Salt, CR-Na, and CR-Na/Salt systems all have similar performance, but the CR-W/S systems are somewhat less efficient. The performances of the central receiver concepts show a substantial improvement in system efficiency with increasing plant size. This improvement is particularly pronounced between the 0.5 MWe design and the 30 MWe. The primary cause of the improvement in performance is improved concentrator, receiver, and heat engine efficiency. Concentrator and receiver performance improves very rapidly between 0.5 MWe and 10 MWe, largely because of reduced intercept losses when receiver size is increased. Low-heat engine efficiency, inherent in the very small steam Rankine heat engines, also improves with size. Increases in plant size beyond 30 MWe do not produce the same dramatic improvement in system efficiency. Increased receiver size and steam turbine size do not result in great component efficiency improvements.

System efficiency results for systems using energy storage to reach a capacity factor of 0.4 are shown in Figure 8.2. Compared to the no-storage cases, the results show a substantial reduction in the system efficiency of the dish concept (from 21% to 18% at 100 MWe), but adding storage improves the system efficiency of the CR-Salt, CR-Na, and CR-Na/Salt systems (from 15% to 18% at 100 MWe). This difference is caused by the poor round-trip storage efficiency of battery storage compared to the good round-trip efficiency of large thermal storage. The increase in central receiver system efficiency when increasing the capacity factor comes primarily from improving the energy input timing to the heat engine. Thermal storage allows significant decreases in mismatch losses (by storing rather than wasting peak insolation) and smaller reductions in heat engine startup losses and standby losses (by reducing the amount of shutdowns occurring during operating times).

Including energy storage for the CR-W/S concept does not result in a significant change because the derating of the plant when operating from storage offsets the advantages of adding storage. As with the no-storage

FIGURE 8.2. System Efficiency of 0.4-Capacity-Factor Plants

cases, the efficiency of the central receiver concepts improves dramatically when plant size increases from 0.5 MWe to 30 MWe. Above 30 MWe, there is only a small increase in system efficiency.

8.2.2 Annual Energy Loss Mechanisms

The total energy produced by the plant that reaches the grid is the result of individual component efficiencies (annual component efficiencies are discussed in Chapter 5.0), general plant energy losses, and losses resulting from plant availability. These losses can be summarized in an efficiency "water fall" diagram. Before discussing the water fall diagrams for individual concepts, each of the major energy loss mechanisms associated with a solar power plant will be defined. A more detailed discussion of the energy loss mechanisms is found in Volume II, Appendix C.

Concentrator - Losses included in the concentrator annual efficiency include all losses from the time immediately before the insolation impinges on the concentrator surface to the time immediately before the concentrated energy impinges on the receiver (either the receiver cover glass or the receiver heat transfer surface). Loss mechanisms included in the concentrator efficiency include geometric optics (cosine losses, blocking, shadowing, and atmospheric attenuation), glass reflectivity, reflectivity degradation (due to dirt and long term permanent losses), solar tracking errors, and receiver spillage.

Receiver - Losses included in the receiver annual efficiency consist of all losses that occur from the time flux impinges on the receiver control volume (the receiver aperture, cover glass or heat transfer surface) until the thermal energy is transferred away from the receiver in a working fluid.

Transport - Thermal transport subsystems include losses from the time thermal energy is absorbed in the working fluid at the receiver to the time the thermal energy is delivered to either the storage subsystem or energy conversion subsystem.

Conversion Mismatch - Conversion mismatch losses account for any thermal energy that exceeds the maximum input rate to the heat engine and therefore cannot be used to produce electrical energy. For systems without thermal energy storage, conversion mismatch losses occur any time collected energy from the field would exceed the current (based on ambient conditions) full load input requirements of the heat engine. Systems with thermal energy storage can dispatch energy to storage when power from the field exceeds the maximum heat engine input and thus have more flexibility in reducing conversion mismatch losses. For systems with thermal energy storage, conversion mismatch losses would occur only after storage had been fully charged and the thermal input requirements of the heat engine were exceeded. Conversion mismatch losses could be avoided entirely by selecting a heat engine with a maximum thermal input equal to the maximum thermal output from the collector field; however, such an approach does not represent a cost effective plant design strategy.

Storage - Performance of thermal storage subsystems includes thermal losses from the time thermal energy is delivered to the input heat exchanger (or storage tank itself if an input heat exchanger is not required) to the time energy is removed through the output heat exchanger.

Heat Engine Startup - This category of losses includes all the thermal energy used to initially warm the heat engine, ramp it to full speed, and bring it into synchronization with the grid.

Conversion - Energy conversion efficiency includes all losses related to the thermodynamic performance of the heat engine, heat rejection equipment, and the generator. The heat engine efficiency reported in this report is an operating efficiency calculated based on the actual energy delivered to the heat engine during operating periods; losses during startup and losses due to conversion mismatch are accounted for in the above categories. Conversion losses do not include parasitic power requirements for the heat engine. Such losses are included with plant parasitics.

Plant Parasitics - These losses include the electric power requirements for the solar plant both during operational periods and during periods when the plant is shut down. These power requirements include heat engine parasitics, pumping power, and heat tracing requirements for energy transport and energy storage, miscellaneous plant operating loads, and general plant standby electrical use.

Turbine Standby - These are electrical parasitic power requirements associated with the Rankine cycle heat engine standby requirement. They include turbine turning power, circulation pumps for standby steam, and gland seal steam.

Availability - Availability losses represent the loss of output caused by scheduled and unscheduled plant outages.

The various loss mechanisms associated with each plant design are summarized on a series of efficiency water fall charts. The water fall charts show the impact of the significant losses on annual performance. For example, Figure 8.3 is an efficiency water fall chart for a 100 MWe CR-Salt system without

FIGURE 8.3. Annual Energy Losses for a CR-Salt System with a No-Storage Capacity Factor in a 100 MWe Plant (481,000 m² field)

storage. The bar on the left shows the total annual energy available in the insolation that could strike the concentrators if they were always directly oriented toward the sun through the entire year. The bars to the right show the impact of various loss mechanisms on the useful energy. The definition of the loss mechanisms has been given above, but the actual presentation of losses associated with the storage needs further discussion. Clearly, storage losses only apply to energy that passes through the storage subsystem; consequently, the bar entitled "storage" does not actually indicate the efficiency of the storage system, but rather the ratio of energy losses from storage. The annual

losses from storage depend upon both the efficiency of storage and the amount of energy that is dispatched to storage. Thus, no losses are reported from storage for the no-storage cases.

One water fall diagram is presented for each of the concepts, showing the efficiency of the 100 MWe no-storage plant. Additional water fall diagrams for capacity factors of 0.4 and for 2 MWe plants are found in Volume II, Appendix E.

Again, the plant design-point selection was based on an overall economic optimization rather than an attempt to optimize performance. A result of this approach is that plants do not necessarily represent the best performance that could be achieved. Rather, they represent a best performance/cost trade off. This approach becomes obvious when examining the results, in which conversion mismatch losses are high. In these cases, plants with higher efficiencies could obviously be designed, but the overall economics of the plant would be inferior to the overall economics of the designs selected for inclusion in this study.

Annual performance breakdowns of the central receiver concepts for 100 MWe no-storage designs are shown in Figures 8.3 to 8.6. The major loss mechanisms are the concentrator losses, mismatch losses, and energy conversion losses. The high mismatch losses result from the concentrator field being excessively large from a performance standpoint, which means that much of the collected energy can neither be used directly in the heat engine or be used to charge storage. The performance of the concepts shows more similarities than it does differences. When compared to the CR-Salt concept, the CR-Na and CR-Na/Salt concepts have slightly lower concentrator and receiver efficiencies, which is caused by the fact that the molten salt receiver uses a cavity design with a north field. A north field tends to be more efficient than a surround field because of the lower cosine losses. The CR-W/S system shows slightly reduced concentrator, receiver, and energy conversion efficiencies compared to the CR-Salt system due to using a surround field design and a lower-temperature energy conversion system.

A similar annual performance breakdown is shown for the dish system in Figure 8.7. The order the losses are presented is different for the dish

FIGURE 8.4. Annual Energy Losses for a CR-Na System with a No-Storage Capacity Factor in a 100 MWe Plant

system; since energy transport losses are electrical (rather than thermal) they appear following the conversion loss. Compared with the central receiver systems, the dish concept has a much higher concentrator efficiency and somewhat lower losses resulting from energy conversion mismatch. The annual efficiency of the Stirling engine is lower than that for the steam turbines used in the central receiver systems.

The annual performance breakdown for the trough system for the 100 MWe no-storage design is shown in Figure 8.8. The dominant loss mechanisms for the trough system are the concentrator, receiver, and energy conversion. The most significant areas where the trough system performance is lower than the central

FIGURE 8.5. Annual Energy Losses for a CR-Na/Salt System With a No-Storage Capacity Factor in a 100 MWe Plant (520,000 m² field)

receiver concepts are receiver losses (due to the trough's low flux receiver), and energy conversion losses (due to the lower temperature heat engine used by the trough).

8.3 SYSTEM CAPITAL AND OPERATION AND MAINTENANCE COST RESULTS

System capital and O&M results are presented and discussed in this section for each of the six systems evaluated. Total capital cost is broken into its six components and the trends in total capital, capital components, and O&M are shown as functions of plant power rating and capacity factor.

FIGURE 8.6. Annual Energy Losses for a CR-W/S System with a No-Storage Capacity Factor in a 100 MWe Plant (581,000 m² field)

Plant capital cost is plotted in Figure 8.9 against plant power rating for each of the six technologies at the no-storage condition. Dish systems are always the least expensive to build and enjoy a factor of four advantage in unit capital cost for a 0.5 MWe plant. All of the systems benefit from economies of scale, but to different degrees. In general, centralized components such as central receivers and steam Rankine engines rely on economies of scale to achieve low unit costs at larger sizes. Distributed components, such as concentrators and Stirling engines, tend to have constant unit costs regardless of the size of the installation. As a result, central receiver systems, with mostly centralized components, show large economies-of-scale, while

FIGURE 8.7. Annual Energy Losses for a Dish System with a No-Storage Capacity Factor in a 100 MWe Plant (395,000 m² field)

dish systems, with mostly distributed type components, benefit relatively little from economies-of-scale. Trough systems, with a mixture of centralized and distributed components, benefit more than dish systems but less than central receiver systems from economies of scale. At 100 MWe, the capital cost advantage of the dish has been reduced to \$400 to \$600/kWe over the central receiver systems. The trough system is always three to four times more expensive to build than the dish system.

Figure 8.10 shows how plant capital costs vary with plant power rating for optimal LEC plants, which may utilize storage. In general, as addressed in the next section, adding collection and storage capacity to a system with a fixed

FIGURE 8.8. Annual Energy Losses for a North-South Trough System With a No-Storage Capacity Factor in a 100 MWe Plant (930,000 m² field)

turbine capacity benefits the system economics. This benefit was found to apply to all of the central Rankine systems evaluated except for the water/steam system at 100 MWe. At this particular design point, the reduction in conversion efficiency when operating from storage increases the LEC more than the benefit of a higher capacity factor decreases the LEC. Dish systems do not reduce their LEC from adding storage because energy conversion capacity must be increased in addition to increasing energy collection and storage. Also battery storage is much more expensive (and less efficient) than thermal storage. The result of the factors discussed above is that plant capital costs for optimal trough and central receiver systems are (with one exception) much

FIGURE 8.9. Initial Capital Cost of No-Storage Systems

higher than for the no-storage cases, but the optimal dish system is its no-storage case. An important implication here is that the minimum capital cost plant is not usually the minimum LEC plant.

The variation of plant capital cost with capacity factor is shown in Figure 8.11 for each of the six technologies at 100 MWe. As expected, plant capital costs rise with capacity factor for all of the technologies as collection and storage capacity are added. Plant capital costs rise the fastest for dish systems, which must add generating capacity to increase capacity factor and must pay for expensive, and inefficient, battery storage.

Operation and maintenance costs, expressed as a fraction of initial capital, are shown in Figure 8.12 as a function of power level for no-storage cases of each of the six technologies. Operation and maintenance costs of operating labor cost, which is nearly a constant, and maintenance labor and

FIGURE 8.10. Initial Capital Cost of Optimum Systems

materials costs, which were estimated to be proportional to component capital costs. The O&M fractions for trough and central receiver systems drop considerably from the fractions for a 0.5 MWe power plant as operating labor costs are spread over larger and larger plant sizes. The dish system has a small operating labor requirement and high maintenance charges for the receiver and Stirling engine. The combination of these two significant differences causes O&M cost (as a fraction of initial capital) for the dish systems to be higher than for the trough or central receiver systems except at the 0.5 MWe power level.

Figure 8.13 shows how O&M cost (expressed as a fraction of initial capital) varies with capacity factor and technology for 100 MWe plants. Capacity factor is shown to have little effect on the O&M fraction for trough or central receiver systems. In general, constancy with increasing capacity factor would be expected because maintenance costs are estimated as fractions

FIGURE 8.11. Initial Capital Costs for 100 MWe Systems

of capital investment and operating labor has become a small part of total O&M cost at 100 MWe. Dish systems show a sharp drop in the O&M fraction as capacity factor is increased. Adding battery storage to dish systems increases capital costs much faster than it increases maintenance costs. In other words, maintenance costs for batteries represent a relatively small fraction of their purchase cost. Batteries do require periodic replacement, however. If battery interim capital replacement was included with O&M, the overall dish O&M cost fraction would rise above 0.20 for a 0.45 capacity factor.

8.4 SYSTEM LEVELIZED ENERGY COST RESULTS

System LEC results are presented and discussed in this section for each of the six systems evaluated. The LEC approach to economic evaluation of solar thermal power plants is discussed in Volume II, Appendix B. The trends in LEC are shown as functions of plant power rating and capacity factor.

FIGURE 8.12. Effect of Power Level on Operation and Maintenance Cost Fractions in No-Storage Plants

The trend in LEC as a function of power level is shown in Figure 8.14 for each of the technologies at no-storage conditions. Figure 8.15 shows the same trend at the optimal LEC design point for each power level. Levelized energy cost, like plant capital cost, shows strong economies of scale, especially for the central receiver systems. Economies of scale are much less a factor for the dish system, while the benefit of economies-of-scale is intermediate for trough systems. A comparison of Figures 8.18 and 8.15 reveals that the optimal dish system has no storage, although all trough and central receiver systems (except the 100 MW_e water/steam system) reduce their LEC by adding collection and storage capacity. Overall, dish systems have the lowest LEC at small power levels and at intermediate power levels with no storage. Central receiver systems have the lowest LEC at 100 MWe and are competitive with dishes at intermediate power levels when they incorporate storage. The trough concept is not

FIGURE 8.13. Effect of Capacity Factor on Operation and Maintenance Cost Fractions in 100 MWe Plants

economically competitive with the other concepts at any combination of power level and storage. Among the optimal LEC design points for central receiver systems, the water/steam system consistently has a slightly higher LEC than the molten salt, sodium, or sodium/salt systems. The LEC for the CR-Salt, CR-Na, and CR-Na/Salt systems show no distinct differences for either optimal LEC designs or no-storage designs.

Figure 8.16 shows the effect of increasing capacity factor on LEC for 100 MWe systems. Again, capacity factor is increased by adding collection and storage capacity to trough and central receiver systems. Dish systems must add generating capacity as well. Increasing the capacity factor reduces the LEC of most trough and central receiver systems by capturing economies of scale in some of the collection components and by spreading fixed energy conversion costs and fixed plant parasitics over a larger annual power output. Much of

FIGURE 8.14. Effect of Power Level on Levelized Energy Costs in No-Storage Systems

the potential for economies of scale have already been captured at the 100 MWe plant size, so the reduction in LEC is not as great as it is for smaller plants. Increasing the capacity factor for dish systems only increases the LEC. This result does not necessarily infer that adding energy storage would never be considered in an application for the dish system. Adding storage to dish systems does not make sense from the standpoint of minimizing LEC, but it might maximize the value of the energy produced (such as shifting the plant output to peak load periods) depending on the requirements of the grid (load).

8.5 COMPARISON TO DOE LONG-TERM GOALS

As an aid to planning research activities needed to make solar thermal energy a technically and economically attractive option, the U.S. Department of

FIGURE 8.15. Effect of Power Level on Levelized Energy Cost in Optimum Systems for Each Technology

Energy's National Solar Thermal Technology Program has established long-term performance and cost goals for solar thermal technology (DOE 1986). The goals are intended to provide achievable targets that, when met, would result in the widespread usage of solar thermal technologies in the marketplace. This section compares the results of this study to the long-term DOE goals for central receiver, dish, and trough systems.

There are several basic differences between the results of this study and the long-term DOE goals that are present for all of the systems:

- insolation - The DOE long-term goals are based on an annual insolation assumption of 2690 kWh/m². The results of this study are based on 1976 Barstow, California, insolation, which recorded direct normal insolation of 2848 kWh/m².

FIGURE 8.16. Effect of Capacity Factor on Levelized Energy Costs in 100 MWe Systems

- availability - The DOE long-term goals assume 100 percent plant availability in the calculation of LEC. The results of this study assume varying degrees of plant availability for each concept in calculating the LEC.
- indirect and contingency costs - The DOE long-term goals implicitly assume a 20 percent rate for indirect and contingency costs in calculating system installed costs. The estimate of indirect and contingency costs developed for this study was 35 percent.

8.5.1 Comparison to Central Receiver Goals

In Table 8.1, the estimated concentrator efficiency is shown to be slightly lower than the goal value. Estimated heliostat costs are significantly higher, however. The discrepancy in cost can largely be attributed to

TABLE 8.1. Comparison of Simulation Results to Long Term DOE Goals for Central Receiver Systems

Component	Goal		Salt ^(a)		Sodium ^(a)		Sodium/Salt ^(a)		Water/Steam ^(b)	
	Effl- ciency	Cost	Effl- ciency	Cost	Effl- ciency	Cost	Effl- ciency	Cost	Effl- ciency	Cost
Concentrator	0.64	\$40/m ²	0.61	\$100/m ²	0.61	\$100/m ²	0.60	\$100/m ²	0.60	\$100/m ²
Receiver	0.90	\$30/m ²	0.87	\$30/m ²	0.83	\$19/m ²	0.83	\$19/m ²	0.79	\$35/m ²
Transport	0.99	\$25/m ²	1.0	\$14/m ²	1.0	\$14/m ²	1.0	\$14/m ²	1.0	\$9/m ²
Storage	0.98	\$20/kWh	0.99	\$12/kWh	0.99	\$19/kWh	0.99	\$12/kWh	NA	NA
Conversion	0.39	\$350/kWe	0.41	\$390/kWe	0.41	\$360/kWe	0.41	\$380/kWe	0.38	\$300/kWe
Balance of Plant	NA	\$30/m ²	NA	\$32/m ²	NA	\$31/m ²	NA	\$31/m ²	NA	\$37/m ²
System	0.22	\$1000/kWe	0.19	\$3500/kWe	0.18	\$3700/kWe	0.19	\$3400/kWe	0.14	\$1800/kWe
O&M	NA	\$9/m ² -year	NA	\$5/m ² -year	NA	\$5/m ² -year	NA	\$4/m ² -year	NA	\$5/m ² -year
Energy Cost	NA	\$0.04/kWe	NA	\$0.05/kWe	NA	\$0.06/kWe	NA	\$0.05/kWe	NA	\$0.07/kWe

(a) 100 MWe, 0.6 capacity factor.

(b) 100 MWe, 0.24 capacity factor.

the heliostat design basis. The figures for this study were based on glass-metal technology, but the goal figure is based on expectations for stressed-membrane heliostats. The previous DOE cost goal for heliostats based on glass-metal technology was \$100/m² (DOE 19840), which exactly matches the estimates produced in this study.

Receiver efficiencies were also estimated to be lower than the goal value. As might be expected, the thermal efficiencies of the sodium and water/steam external receivers were lower than the efficiency of the molten salt cavity receiver. From a cost perspective, the higher flux capabilities of the sodium receivers reduce material requirements and costs per unit of concentrator field size, yielding a unit cost much lower than that for molten salt or water/steam receivers, and lower than the long-term goal.

Transport efficiencies (thermal efficiency) were estimated to be near 100%, to at least two significant digits. This estimate compares well with expectations established in the goals. Transport costs were estimated to be roughly half of the cost goal.

Storage system efficiencies, like transport, were estimated to be slightly higher than, but essentially the same as, the goal values. Sodium storage costs were estimated to be near the goal value, but molten salt storage costs were estimated to be less expensive by a factor of about 1/3.

Energy conversion costs and performance were both estimated to be slightly higher than the goal value for molten salt and sodium systems, although it should be noted that the DOE performance goal is calculated on a different basis than the energy conversion performance reported here. The DOE performance goal is an overall net efficiency, which includes all annual turbine startup energy, any turbine thermal mismatch, and all turbine parasitic power requirements. The energy conversion efficiency reported in this study is a gross efficiency, with the other factors being included separately. Including turbine startup energy, thermal mismatch, and parasitic power in the energy conversion efficiency in the efficiencies from this study would result in a comparable efficiency of roughly 0.36 for the CR-Salt system and similar reductions for the other systems. The water/steam conversion cost is significantly lower than the goal because no steam generator is required for the (optimal)

no-storage system shown in Table 8.1. Water/steam conversion efficiency is lower because of a lower turbine inlet temperature.

Estimated balance-of-plant costs for central receivers compare favorably with the established goal.

System efficiency was estimated to be from three to four percentage points (18 or 19 versus 22) lower than the goal value for molten salt and sodium systems and significantly lower for the water/steam system. The factors that cause the system performance to be significantly lower than the goal are overall plant parasitics, turbine startup energy, turbine thermal mismatch, and plant availability. The first three factors are not accounted for explicitly in the DOE goals, and availability is not included in the DOE goals.

System capital costs in \$/kWe are much higher than the goal, but this difference may be misleading because the unit capital cost for a solar thermal plant will vary over a factor of two or more depending on the capacity factor. Economically efficient plants tend to incorporate larger collection and storage systems than the minimum required for a given size turbine. For example, the molten salt and sodium systems shown in Table 8.1 are all designed to achieve capacity factors of 0.6 and have unit capital costs of about \$3500/kWe. The water/steam system was designed for a capacity factor of 0.24 and has a unit capital cost of \$1800/kWe.

Overall, the molten salt and sodium systems were estimated to produce energy at a cost of \$0.05 to \$0.06/kWh, or 25 to 50% higher than the goal. The two biggest differences are caused by the system efficiency and cost of the heliostat. Dropping the heliostat cost from \$100/m² to the goal value of \$40/m² (without any changes in concentrator efficiency) would reduce the LEC predicted for these concepts to roughly \$0.05/kWh.

8.5.2 Comparison to Dish Goals

The dish concentrator efficiency was estimated to be higher and the receiver efficiency lower than values specified by the goals, but the combination of concentrator and receiver efficiency estimated was nearly identical to the goal (see Table 8.2). Estimated concentrator costs were within 10% of the goal value.

TABLE 8.2. Comparison of Simulation Results to Long Term DOE Goals Dish Systems

Component	Goal		Dish ^(a)	
	Efficiency	Cost	Efficiency	Cost
Concentrator	0.78	\$130/m ²	0.86	\$120/m ²
Receiver	0.90	\$70/m ²	0.82	\$21/m ²
Transport	0.99	\$7m ²	0.96	\$17m ²
Storage	NA	NA	NA	NA
Conversion	0.41	\$300/kWe	0.35	\$190/kWe
Balance of Plant	NA	\$20/m ²	NA	\$31/m ²
System	0.28	\$1200/kWe	0.21	\$1300/kWe
O&M	NA	\$10/m ² /yr	NA	\$20/m ² /yr
Energy Cost	NA	\$0.05/kWhe	NA	\$0.07/kWhe

(a) 100 MWe, 0.27 capacity factor.

The most significant differences between the long-term goals and the results of this study for the dish system are found when comparing estimates of capital and O&M costs for the dish power conversion unit (receiver and energy conversion) with the long-term goals. The receiver was estimated to cost only 1/3 of the goal value, and the Stirling engine cost two-thirds of the goal value. The cost advantage of the power conversion unit (relative to the goals) was mitigated by much higher annual maintenance costs (estimated at double the long-term goal) and lower efficiency for both receiver and energy conversion.

Transport costs and efficiencies were estimated to be worse than expectations specified by the goals. Field transformer and line losses were estimated to be four times higher than the goal value, and costs were estimated to be more than twice as high.

The balance-of-plant goal for dish systems is lower than for central receiver systems, but costs per square meter of concentrator aperture area were

estimated to be about the same for both systems. Dish system balance-of-plant requirements and costs are actually less, but their concentrator fields are also smaller, which keeps the unit cost approximately the same.

The dish system capital cost was estimated to be less than 10% higher than the goal. This did not translate into a reduction in the LEC projection relative to the goal, however, because of a much lower efficiency and higher annual O&M cost estimated for the dish system relative to the goal values. The estimated LEC for the dish system was \$0.02/kWh higher than the \$0.05/kWh goal.

8.5.3 Comparison to Trough Goals

Although long-term goals have not been specifically developed for trough-electric systems, comparisons can be made to most of the trough-thermal component goals. Estimates of cost and performance from this study are compared to the long-term goals for trough systems in Table 8.3.

TABLE 8.3. Comparison of Simulation Results to Long Term DOE Goals for Trough Systems

<u>Component</u>	<u>Goal</u>		<u>Trough^(a)</u>	
	<u>Efficiency</u>	<u>Cost</u>	<u>Efficiency</u>	<u>Cost</u>
Concentrator	0.65	\$110/m ²	0.64	\$150/m ²
Receiver	0.90	\$30/m ²	0.60	\$14/m ²
Transport	0.98	\$30/m ²	0.93	\$48/m ²
Storage	0.98	\$20/kWh _t	0.99	\$7/kWh _t
Conversion	NA	NA	0.26	\$630/kWe
Balance of Plant	NA	\$20/m ²	NA	\$17/m ²
System	NA	NA	0.08	\$8200
O&M	NA	\$6/m ² /yr	NA	\$6/m ² /yr
Energy Cost	NA	NA	NA	\$0.15/kWe

(a) 30 MW, 0.52 capacity factor.

Concentrator efficiency was estimated to be about the same as the goals figure, but receiver efficiency was considerably lower. Concentrator costs were estimated to be higher than the goal value; the difference between estimated costs and the trough concentrator goal is partly due to the cost of the support structure of the trough concentrator design evaluated, which may be more rigorously designed than necessary (see Section 6.1).

The estimated receiver cost does not include the expensive connecting flex hose, which may partly explain why estimated receiver costs are less than the goal value and transport costs were estimated to be higher than the goal. Transport unit costs, like so many components, are strongly dependent on the size of the transport system. In addition to costing more than the goal figure, the transport component was estimated to be considerably less efficient than the goal.

Trough storage costs were estimated to be about one-third of the goal value. As discussed in Section 7.4, storage costs (per kWh) are inversely proportional to the media cycle temperature range. A specific example of differences in media cycle temperature causing a factor of three difference in unit cost was cited in Section 7.4.

No conversion component goals have been established for trough systems. As might be expected, trough conversion efficiency is lower and unit costs are higher than for either dish or trough systems.

Balance-of-plant costs were estimated to be 15% lower than the long-term goal and are about half the unit cost of central receiver and trough systems. Trough systems benefit from having a higher ground cover ratio, which reduces land requirements and costs.

Operating and maintenance costs were estimated to be slightly higher than central receiver systems, but equal to the trough long-term goal for process heat. Levelized energy costs were estimated to be much higher than for either the central receiver or dish system.

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9.0 CONCLUDING DISCUSSION

This section of the report discusses overall conclusions of the study and areas of significant uncertainty in the analysis. System performance conclusions are discussed in Section 9.1, cost conclusions in Section 9.2, overall conclusions in Section 9.3, and areas of uncertainty in Section 9.4.

9.1 SYSTEMS PERFORMANCE CONCLUSIONS

The major conclusions concerning system performance are grouped into four categories. First, an overall comparison of concepts is presented, followed by discussions of the impact of plant size on performance and the impact of capacity factor on performance. Finally, the impact of parasitics, startup energy requirements, and system mismatch are discussed.

9.1.1 Overall Evaluation

The six concepts were compared based on average annual system efficiency, and the following conclusions were drawn:

- The dish system demonstrates the highest overall solar-to-electric conversion efficiency for designs without energy storage. For higher capacity factors the efficiency of the dish system is penalized by the poor round-trip storage efficiency of battery storage. Even for capacity factors of up to 0.4, though, the dish system efficiency remains equal to the efficiency of the central receiver systems for larger plant sizes. Efficiency is better for the dish system than for the central receiver systems at plant sizes of 10 MWe and less. The reasons for the high efficiency of the dish system are the very high concentrator efficiency and the low energy losses for heat engine startup and thermal mismatch.
- The trough system consistently has the lowest average annual overall conversion efficiency. This low efficiency is caused by high receiver losses and low heat engine efficiency associated with low-temperature collectors (and hence low-temperature input to the heat engine).

- The average annual overall conversion efficiency of the central receiver systems falls between the dish and the trough systems. The four central receiver systems all have similar performance at low capacity factors. However, as storage is added, the CR-W/S system overall conversion efficiency generally drops because of decreased heat engine efficiency when operating with lower-temperature steam provided from the oil/rock storage. In the no-storage case, the performance variations that exist among the central receiver systems are primarily caused by differences in collector efficiency and heat engine efficiency. The field efficiency of the molten salt system is substantially higher than any other central receiver system because of the high efficiency of a north field combined with a cavity receiver. The heat engine efficiency of the CR-Salt and CR-Na systems are identical, but the heat engine efficiency of the CR-W/S system is slightly lower for the no-storage case and decreases further as storage is added.

9.1.2 Impact of Plant Size On Performance

The impact of increasing plant size on average annual overall conversion efficiency depends on the concept.

- The dish system does not show a significant variation in overall conversion efficiency with plant size. Because of the modular nature of the concept, the conversion efficiency of one dish unit is not significantly different from the conversion efficiency of an array of units.
- The trough system shows an increase in average annual conversion efficiency with increasing plant size up to a plant size of 30 MWe. The poor performance of small installations is primarily caused by poor conversion efficiency for small-size heat engines. As plant size increases, energy conversion efficiency increases, but the efficiency of the transport subsystem decreases.

Therefore, when a 100 MWe system is considered, the rise in heat engine efficiency caused by the larger heat engine is more than offset by the drop in transport-related losses.

- The central receiver systems all show strong performance economies of scale. Small central receiver systems have low annual overall energy conversion efficiency caused by high spillage losses and low heat engine efficiency. As plant size increases from 0.5 MWe to 10 MWe, overall system efficiency improves dramatically. There is little change in system efficiency between 30 MWe and 100 MWe; this constancy appears to be caused by the fact that spillage losses are essentially eliminated for plant sizes of 30 MWe and above.

9.1.3 Impact of Capacity Factor On Performance

As with plant size, the impact of the increasing capacity factor on average annual conversion efficiency depends on the concept.

- The dish average annual system conversion efficiency decreases as the capacity factor increases because of the poor round-trip storage efficiency associated with battery storage. As the capacity factor increases, more electric energy passes through the battery storage subsystem and associated losses increase.
- CR-Salt and CR-Na systems demonstrate an improvement in average annual system efficiency as the capacity factor is increased. This improvement is caused by improved collector field efficiency and is associated with larger fields and reduced energy conversion mismatch and startup losses. The trough system also shows an improvement in average annual system efficiency as the capacity factor is increased because of reduced energy conversion mismatch and startup losses.
- The CR-W/S system has an average annual system efficiency approximately equal to the other central receiver concepts at low capacity factors. However, as storage is added to allow the concept to operate at a higher capacity factor, the system efficiency generally declines. This decline is caused by a progressively larger fraction of the receiver output energy passing through storage. With oil/rock

storage, the heat engine efficiency is degraded whenever the plant is operating from storage; this is due to the lower steam pressure and temperature resulting from the oil/rock storage temperature of 580°F. The only exception to this case is for the 0.5 MWe case, where increasing capacity factor does increase the system efficiency. For the 0.5 MWe CR-W/S plants, the increases in concentrator and receiver efficiency and the decreases in conversion mismatch and startup losses more than compensate for the reduced efficiency when operating the heat engine from storage.

9.1.4 Parasitic, Startup, And Mismatch Losses

The impact of plant parasitics (both thermal and electric), startup energy requirements, and thermal mismatch between collector and turbine were found to be significant for all systems analyzed. For example, these factors for the no-storage 100 MWe central receiver salt case accounted for a combined efficiency multiplier of 0.75, constituting a greater loss than the receiver efficiency.

9.2 COST CONCLUSIONS

Concentrator and O&M are the two dominate cost components for all systems across the range of power levels examined. Operation and maintenance dominates for all technologies at small power levels. Operation and maintenance continues to be the largest contributor to annualized cost for dish systems, even at the higher power levels, but the concentrator edges out O&M as the dominate cost component for trough and central receiver systems at higher power levels.

Several components of the dish system stand out as being much higher or lower in cost than corresponding components of other systems. Dish systems are projected to run automatically; hence, their operating labor requirement (essentially nil) is much lower than either the trough or central receiver systems. On the other hand, maintenance labor and materials for the dish power conversion unit are so high that the total O&M cost for the dish is higher than for trough or central receiver systems. Energy conversion cost per kWe for the dish is roughly half that of the 100 MWe Rankine systems, but electric storage is much more expensive than any of the thermal storage options. Two component

cost differences stand out for the trough system. Distributed thermal transport cost is up to four times higher than that for central receiver or dish transport subsystems, but land-related costs are lower for the trough system because of a higher ground cover ratio or packing factor.

Central receiver systems rely on economies of scale from all components but the concentrator to achieve relatively inexpensive systems at larger sizes. On the other hand, the distributed nature of dish system components (other than balance-of-plant components) means that costs tend to vary only slightly with size. The trough system is somewhere in between these two extremes.

9.3 OVERALL CONCLUSIONS

Within the ground rules of this study, all of the central receiver concepts analyzed and the dish system analyzed appear to show the potential to be economically competitive electric production technologies. In establishing energy cost goals for solar thermal electric plants, DOE characterized the expected range of LEC from intermediate capacity factor coal plants to range from \$0.05 to 0.08/kWh in the late 1990's (DOE 1985). All of the central receiver concepts can achieve energy costs within this range for the larger plant sizes, and the dish system can achieve such costs over a significant range of plant sizes.

The lowest LEC achieved by a trough in this study was \$0.15/kWh. Based on these results, the trough concept does not appear competitive with either the central receiver concepts or the dish concept as a source of electric power. Compared to the economic feasibility range cited above of \$0.05 to 0.08/kWh the trough concept also does not appear to be a good candidate for an economic electric power producer. This result is particularly interesting in light of the fact that the worlds largest solar thermal electric plants (located in Daggett, California, and built by LUZ Industries Ltd.) are trough plants. One factor that may partially explain the success of the LUZ plant is that it is operated as a hybrid; solar collected heat is used to produce saturated steam, which is then superheated by fossil fuel. In this application, the efficiency of the energy conversion system could be much higher than the efficiency of the trough system analyzed in this study because the turbine inlet steam conditions

need not be dictated by the quality of heat available from the trough concentrators. A reason for the currently greater commercial application of troughs than of central receiver or dish technologies may be that troughs are the most economical approach for near-term systems. The results of this study are aimed at providing projections of the long-term capabilities of solar thermal systems and do not address the current economic ranking of the concepts.

From an LEC standpoint, the differences in any of the central receiver concepts analyzed was trivial for plants without energy storage. For plants with moderate capacity factors (0.3 and higher), the CR-W/S concept is significantly more expensive than the other three central receivers because of the impact of the storage system on the energy conversion efficiency. Because of this, the difference in LEC between the CR-W/S system and the other central receivers becomes more pronounced at higher capacity factors.

Plant size was found to have a significant impact on the LEC for all concepts analyzed. The highly modular dish concept was least affected by plant size, but even this system had much higher LEC at the 0.5 and 2 MWe levels than at the larger plant sizes. For dish systems, the most important LEC driver at the smaller sizes is balance-of-plant costs, which have relatively large fixed cost components and therefore become a larger part of the total plant cost as plant size decreases.

A number of factors in addition to balance-of-cost plants increase the LEC of the central receiver and trough concepts for small plants. Decreasing overall efficiencies of the energy conversion subsystem for small plants are a major factor in reducing the efficiency (and hence raising the LEC). The unit costs in \$/kW for energy conversion increase by over a factor of four between 100 MWe and 0.5 MWe. This increase also substantially impacts costs for both central receivers and troughs. In addition to these factors, central receiver systems have several unique aspects that increase LEC for small plants. Both tower and receiver unit costs increase substantially at smaller power levels. For the systems included in this study (with large glass-metal heliostats), the annual collection efficiency tends to drop off dramatically for the smallest plant sizes. These factors explain why the central receiver concepts have the

greatest increase in LEC with decreasing plant size of any of the concepts considered. Although further optimization of the collector for small central receiver systems would undoubtedly raise the collection efficiency, it would have no impact on the other cost drivers. It is clear that any of the central receiver systems considered here will be significantly more expensive for small applications than they would be for larger plants.

Increases in capacity factor tend to decrease the LEC for concepts using thermal energy storage, provided that the energy is stored at roughly the receiver outlet temperature. The most important reason for this fact is that many plant costs (such as the energy conversion subsystem and balance-of-plant components) tend to stay fixed as the solar multiple of the plant increases. The overall plant cost does not increase to as great a percentage as does the annual energy output. For the CR-W/S system, energy storage takes place at a lower temperature, significantly reducing system efficiency when operating from storage; the result is that increasing capacity factor increases the LEC for the CR-W/S concept.

Increases in capacity factor beyond the no-storage case substantially increases LEC for the dish system. The reasons for this are the high storage cost for electric energy storage and the low storage efficiency of the system. In addition, the dish system tends to have few fixed costs, so overall costs tend to increase at nearly the same rate as system energy output when solar multiple is increased. However, these results do not necessarily indicate that energy storage would never be used for a dish system. Increases in the capacity factor may also increase the value of the energy in some applications. Thus, although the LEC increased, the overall economics of the system could potentially be improved (Dirks and Chiang 1986). In addition to this, for very small systems the LEC for the dish is so much lower than other concepts that the capacity factor for the dish can be increased to levels of intermediate-load plants (0.3 to 0.45 capacity factor) and still achieve a lower LEC than the other concepts.

All of the annual performance estimates for the concepts evaluated in this study were substantially lower than the performance goals established in the DOE solar thermal technology program (DOE 1986). Some of this variation is due

to specific component efficiencies being lower than goals (for example, the Stirling engine annual efficiency of 35% is substantially less than the DOE goal of 41%), but one of the major driving forces is the impact of plant parasitics and startup loads on plant annual efficiency. These factors often represented an overall efficiency of 75% and constituted a significant energy loss.

9.4 AREAS OF SUBSTANTIAL UNCERTAINTY

There are obviously a number of uncertainties in providing projections of solar thermal capabilities. Few commercial plants have been built, and performance/cost data for advanced systems that could be built in the next decades are extremely limited. Although the performance analysis was to a great extent based on empirical data, substantial areas of uncertainty exist.

- Component Performance - Many of the major components analyzed in this study have not been built or tested on a commercial scale. An example is a 100 MWe molten salt receiver. While the designs and performance estimates used in this study are based on the best available design-study and field-test data, the actual performance of components, such as large heliostats, large central receivers, and low-cost dish concentrators remains an area of uncertainty.
- System Optimization - This was not an optimization study, consequently some subsystems may not be optimum and performance may not represent the best achievable.
- Off-Design-Point Operation - Results from this study indicate that a considerable portion of the total plant energy losses occurs during plant operation away from design point conditions. These losses include reduced efficiencies during part-load operations, conversion mismatch, heat engine startup and standby, reduced operation resulting from equipment failure, and other factors. There is still considerable uncertainty in the magnitude of these effects. Relevant data on the impact of these factors for operating plants is limited; in addition, the SOLSTEP code is not extremely detailed in the modeling of all of these phenomena.

- Plant Availability - The question of plant availability was not extensively investigated and would directly affect annual power production.
- Parasitic Loads - Parasitic loads were scaled from previous designs. It is not clear how accurately previous design studies have evaluated parasitic power requirements.

Although all of the cost components have some uncertainty, O&M, central receiver absorbers, and the dish power conversion unit stand out as having a higher level of uncertainty than the other components. The uncertainty in O&M costs stems from ill-defined requirements for operating labor, maintenance labor, and materials. As opposed to the other components, O&M requirements are not easily specified by direct engineering calculations. Experience with operating facilities is the best source of information, but data are limited at this time. Central receiver absorber designs have become more standardized, yet costs estimated by various organizations for similar equipment differed by a factor of two and greater for individual subcomponents. Finally, a higher-than-average level of uncertainty is associated with the initial capital and annual maintenance costs for the dish power conversion unit (receiver and Stirling engine). The dish power conversion unit has not been subjected to as many detailed investigations of its manufactured cost as have other distributed components, namely the heliostat, yet this is the critical component for the dish system. Maintenance costs, although known to be several times higher than any other solar component, are not well defined for the same reasons mentioned above for O&M in general.

One of the major uncertainties in this study's results is the energy cost projections for very small plants, particularly below 10 MWe. Few detailed design studies have been done for plants in this size range, and there is considerable uncertainty in many areas, such as plant O&M requirements and the design of optimal central receiver systems.

Another significant uncertainty affecting the comparison of the concepts is the capital cost and maintenance requirements of the Stirling engine for the dish system. Achieving the Stirling engine cost and performance used in this study will require further engineering development and significant production

economies of scale, and therefore involves a large uncertainty. The combined capital cost and annualized engine maintenance cost for the dish system constitutes roughly 50 percent of the system LEC for that concept, so the economic viability of the dish-Stirling system would be substantially affected by increases in engine capital or maintenance cost.

A factor that would probably not affect the relative performance of the solar concepts in this study, but which would have a significant impact on the absolute performance and energy cost that could be achieved, is the climatic characteristics used in the analysis. The simulation analysis in this study used measured 1976 insolation values for Barstow, California, in order to be comparable to several previous system studies (Laity et al. 1980; Apley et al. 1980). Data that has been collected at Barstow since 1976 has had lower direct normal insolation than those used previously. Although (in some cases substantially lower) the simulation results in this study are felt to be reasonable for the climatic data used, it should be recognized that the actual climate in a given year will play a major role in the energy that can be produced by a solar thermal power plant. The insolation characteristics used in this study can be considered good relative to the last few years experienced at Barstow, but they are probably not optimistic for a number of good solar sites in the southwestern United States. Barstow has never been considered to be the absolute best site for a solar thermal plant.

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