MASTER

THE AEROSPACE CORPORATION

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ANALYSIS OF PHOTOVOLTAIC/THERMAL ELECTRIC
POWER PLANT SYSTEMS

by

Donald F. Gluck and William A. Kelley

Energy and Resources Division
THE AEROSPACE CORPORATION
El Segundo, California

March 1979

Prepared for

The United States Department of Energy
Division of Distributed Solar Technology
Under Contract No. EY76-C-03-1101
Project Agreement 8
Modification No. 5

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

ABSTRACT

A conceptual definition and performance evaluation of a 100 megawatt (MW) hybrid photovoltaic/thermal electric power plant has been carried out. The concept utilizes the ability of gallium arsenide photovoltaic cells to achieve high conversion efficiency at high incident fluxes and elevated temperatures. Solar energy is focused by a field of steerable mirrors (heliostats) onto a tower mounted receiver whose outer surface is covered with gallium arsenide (AlGaAs/GaAs) solar cells and whose inner surface is a water boiler. The solar cells convert a fraction of the incident radiation into electrical energy, and the remaining energy is extracted at approximately 200°C and used to power a Rankine cycle turbine generator (bottoming cycle). Water is used as the solar cell array coolant, as the thermodynamic working fluid, and as the thermal energy storage medium.

Parametric studies were conducted to select conceptual design parameters and operational characteristics which imply the lowest levelized busbar electric energy costs. Parameters varied were collector area, condenser surface area, fan power, ambient temperature, and electric and thermal energy storage capacities. The report describes the concept, outlines the design analysis method, summarizes the parametric study results, and defines the selected plant configuration. The lowest levelized busbar electric energy generation cost, 70 mills/kilowatt-hr., was achieved with a relatively small collector area, 0.8×10^6 square meters, and no stored energy. A rough comparison of this combined power plant with a similar photovoltaic plant, operated at lower solar cell temperature and with no bottoming cycle, showed the busbar cost of electricity (BBEC) from the combined system to be approximately 9% lower.

Foreword

This report documents the results of one of several analytical investigations conducted under Contract No. EY 76-C-03-1101, Project Agreement 8, Modification 5 "Photovoltaic Mission Analysis", for the U.S. Department of Energy, Division of Distributed Solar Technology. The analyses described here were carried out as part of Task 1: Photovoltaic Total Energy Systems.

This document was prepared by Dr. Donald Gluck of the Thermal Analyses Section and Mr. W. Kelley of the Photovoltaics Projects Office. It is based primarily on the work of Dr. Gluck, with computer coding by Bruce Kubert of the Data Concepts and Analysis Department.

The authors express their appreciation for the contributions to this investigation made by other Aerospace Corporation personnel. These include, in particular, the early work of P. L. Merryman, and J. Biddle. The help of Ms. F. K. Morris in typing the text and of F. C. Eggers in overseeing the production of the report is also gratefully acknowledged.

TABLE OF CONTENTS

1.0	INTRODUCTION	1
1.1	Background	1
1.2	Concept	3
1.3	Procedure	3
2.0	CONCLUSIONS	4
3.0	BASELINE POWER PLANT	6
3.1	Concept Selection	6
3.2	Thermoelectric Subsystems Selection	11
3.2.1	Working Fluid	13
3.2.2	Heat Projection Subsystem	14
3.2.3	Thermal Energy Storage Subsystem	17
3.3	Thermoelectric Subsystems Analysis	17
3.3.1	Preheater/Boiler	19
3.3.2	Array Preheater/Boiler	23
3.3.3	Turbine-Generation Subsystem	26
3.3.4	Thermal Energy Storage	31
3.3.5	Heat Rejection	34
4.0	COMPUTER MODELING	36
4.1	Photovoltaic/Thermal-Electric Plant Simulation Approach ...	36
4.2	Model Implementation	38
4.3	Cost Analysis	44
4.3.1	Collector and Receiver	44
4.3.2	Turbine-Generator	44
4.3.3	Heat Rejection Subsystem	48
4.3.4	Photovoltaic Electric Energy Storage and Power Conditioning	51
4.3.5	Thermal Energy Storage	51
4.3.6	Miscellaneous Components	51
4.3.7	Levelized Busbar Cost of Energy Generation	54
5.0	RESULTS	55
5.1	Simulation Without Storage	55
5.1.1	Parametric Studies	55
5.1.2	Detailed Results	61

TABLE OF CONTENTS (Cont'd)

5.2 Simulation with Thermal Storage	66
5.3 Simulation with Electric Storage	73
5.4 Simulation with Thermal and Electric Storage	79
5.5 Photovoltaic-Only Versus PVTE Power Plant	79
NOMENCLATURE	83
REFERENCES	88

LIST OF FIGURES

3-1.	Generic Photovoltaic/Thermal-Electric System	8
3-2.	Comparison of Single and Dual Loop Systems	9
3-3.	Photovoltaic/Thermal-Electric Plant Conceptual Design	11
3-4.	Thermodynamic Cycle Schematic	13
3-5.	Indirect (Heller) Condensation	17
3-6.	Thermal Storage Using Direct Contact Spray Condenser	19
3-7.	Tower Configuration	21
3-8.	Part-Load Efficiency Correction for Throttle Flow Ratio	31
3-11.	Expansion Process	35
3-12.	Wyodak Air Cooled Condenser Arrangement	37
4-1.	Photovoltaic/Thermal-Electric Plant Simulation Model	39
4-2.	Thermodynamic Cycle Modes of Operation	41
4-3.	Relationship Between Incident Solar Flux on Receiver and Thermodynamic Cycle Modes of Operation	43
4-4.	Cost of Turbine - Generator Building	49
4-5.	Cost of Turbine - Generator	51
4-6.	Cost of Direct Dry Cooling Heat Rejection Systems	52
4-7.	Cost of Storage Tanks and Insulation	54
4-8.	Electric Plant Equipment Costs	55
5-1.	Investigation of Air Mass Flow Rate	59
5-2.	Investigation of Heat Exchanger Size	60
5-3.	Variation in Design Ambient Temperature	61
5-4.	Investigation of Collector Area	62

LIST OF FIGURES (Cont'd)

5-5.	Cost of Electric Energy as Function of Collector Area for Various Thermal Storage Capacities	71
5-6.	Levelized Busbar Energy Cost vs Thermal Energy Storage Capacity for Various Collector Areas	72
5-7.	Cost of Electric Energy as a Function of Collector Area for Various Electric Storage Capacities	78
5-8.	Investigation of Collector Area and Storage Amount Both Thermal and Electric Storage	80

LIST OF TABLES

3-1.	Thermodynamic Cycle - Baseline System Definition	10
4-1.	Mode of Operation of Thermodynamic Cycle	42
4-2.	Capital Cost Relationships	47
5-1.	Design Parameters, Physical Characteristics and Rated (Nominal/ Mode 1) Performance for Optimized Power Plant	64
5-2.	Monthly and Yearly Performance of Optimized Power Plant with no no Storage, kW-hr	67
5-3.	Thermal Electric Performance of Optimized Power Plant with no Storage for Various Modes of Operation, kW-hr	69
5-4.	Capital Costs, Energy Delivered & Levelized Cost of Energy for Optimized Power Plant with no Storage	70
5-5.	Monthly and Yearly Performance of Typical Power Plant with Storage, kW-hr	75
5-6.	Thermal Electric Performance of Typical Power Plant with Storage for Various Modes of Operation, kWh	76
5-7.	Capital Costs, Energy Delivered & Levelized Cost of Energy for Typical Power Plant with Storage	78

1.0 INTRODUCTION

The objective of this study was to examine the performance of an idealized solar power plant which generates electrical energy from both solar cells (photovoltaic energy) and a working fluid thermodynamic bottoming cycle (thermal electric turbo-generated energy), considering as major variables the sensitivity to variations in collector area size and both thermal and electric storage capacity. The ultimate cost of electrical energy from these plants has been evaluated as a function of the above variables. This has permitted a preliminary evaluation of the merits of thermal electric enhancement of photovoltaic central receiver power generation systems.

1.1 BACKGROUND

An obstacle to the wide-spread use of photovoltaic solar energy conversion as a general source of electricity is the high cost of the photovoltaic modules. One promising approach to the overcoming of this obstacle is to employ optical concentration systems to collect large amounts of incident sunlight and direct it to relatively small photovoltaic cell assemblies. This procedure, in effect, replaces much of the expensive photovoltaic module area with an equivalent area of concentrating collector that is expected to be much less expensive (relative to current module prices, at least). A further potential advantage arises from the fact that, since the cost of the cell assembly in a high-concentration system becomes relatively insignificant in comparison to collector costs, one can afford to pay a significant premium for higher efficiency in the cell assembly and use, for example, gallium arsenide cells¹. Higher efficiency, in turn, implies smaller collector area for a given output and therefore lower overall cost.

Several previous studies have indicated that the economic attractiveness of high concentration photovoltaic power plants might be significantly enhanced if the thermal energy extracted in cooling the

photovoltaic arrays could be used to generate additional electricity in a bottoming cycle turbine. These studies have shown that large amounts of thermal energy must continually be removed from the photovoltaic cell assemblies in high-concentration plants in order to keep cell temperatures low enough for efficient conversion. If an economic use could be found for this surplus thermal energy, the competitive position of these plants would be substantially improved.

This study was specifically motivated by an earlier, abbreviated comparison of three options for the design and operation of a photovoltaic power plant using the central receiver concentrating configuration.² These options were: (a) provide photovoltaic electric energy only (b) provide both photovoltaic electric energy and thermal energy for heating and (c) provide photovoltaic energy and thermal electric energy, the latter by using the array cooling fluid as the working fluid in a thermodynamic bottoming cycle. This study was exploratory in nature and was subject to some major limitations; while the effect of varying insolation and ambient temperature were considered in a qualitative manner, there was no full year simulation of power plant operation, turbine off-design performance was not accounted for, nor was thermal storage included. Additionally array cooling was by sensible heating of water requiring massive flow rates. Hence these analyses were dominated by pump power requirements.

The present study was undertaken to overcome these and other limitations, and to provide more detailed and realistic information on operation and performance of a hybrid systems. A better understanding of the operation of such a system and of the effects of variation in critical system parameters required detailed simulation computations. In order to carry out such computations a computer program that simulates the operation of a photovoltaic/thermal-electric plant was developed.

1.2 CONCEPT

A baseline power plant system was conceptually defined consisting of a photovoltaic central receiver with a steam thermal bottoming cycle turbine. The total plant rating is nominally 100 MW, of which generally about 60 MW is provided by the photovoltaic array and about 40 MW is provided by the turbine.

This power plant employs a cylindrical central receiver, mounted on a tower, whose outer surface is covered with gallium arsenide solar cells formed into an array. This receiver also serves as the boiler for the thermal electric system. The solar concentration system consists of a field of steerable mirrors (heliostats) that direct multiple images of the sun onto the array surface with an effective concentration ratio as high as 1000. The cells are cooled by water which is converted to steam and used to drive the turbine. The steam leaving the turbine is condensed and pumped back to the boiler. The level of technology assumed is that projected to the 1980's and early 1990's. Such advanced technology is not utilized throughout, but is drawn upon for some key elements of the plant, as is discussed below.

1.3 PROCEDURE

This concept and the various subsystems comprising the plant are physically defined and their performance (including off-design) mathematically characterized. Cost algorithms are developed for the various subsystems. A rationale for power plant operation which considers both thermal and electrical energy storage and accounts for varying insolation and ambient temperature is developed. Parametric studies are then conducted of power plant performance using a computer model of the plant and whole year simulation of plant operation; an economic evaluation is conducted to identify preferred combinations of subsystem design parameters and preferred modes of power plant operation; and conclusions are then drawn concerning the economic competitiveness of a combined photovoltaic/thermal electric central power plant. The major parameters varied to determine their sensitivity included the collector field area size and the amounts of thermal and electric storage. The results of these analyses are presented herein.

2.0 CONCLUSIONS

This investigation, which in effect constitutes a conceptual definition of a combined photovoltaic/thermal electric power plant, has demonstrated the general feasibility of and identified some important technical and economic issues associated with this combined power plant. A single fluid loop with water functioning as both coolant and thermodynamic cycle working fluid appears workable. However, the extremely high heat transfer coefficients and low fouling factors required in the receiver preheater/boiler heat exchanger preclude relying on analytic predictions alone. A test program and pilot plant demonstration would ultimately be required to verify that the required thermal performance could be obtained. A high back pressure turbine compatible with moist steam is not an off-the-shelf item. Although the development effort is not considered major, some work by the manufacturer is required to assure that a reliable, efficient turbine of this type, in the 30 MW to 50 MW range, could be available. The dry-air-cooled direct condensation heat rejection subsystem is in an advanced state of development. It appears that this subsystem could be ordered today from GEA Gesellschaft fuer Luftkondensation. The direct contact spray condenser thermal energy storage subsystem does not appear to present any serious technical problems, save for the large water treatment requirement so incurred. Other concepts for more efficient utilization of the thermal energy from a solar photovoltaic central receiver system may exist or be developed. Therefore, the results of this study necessarily apply only the configuration employed herein. Other cycles, types of implementation, working fluids, cooling techniques, and modes of operation may be preferred for reasons not analyzed or may be forced by availability, environmental or other constraints. The choice of concept and configuration was, in general, made to minimize uncertainty in both performance and cost thus providing a reasonable assurance of a rational result.

The results have shown that thermal energy storage, of the type used here, failed to improve power plant economics. Such storage can improve plant flexibility and increase plant utilization and capacity factor -- at the expense of higher levelized busbar energy cost (BBEC). The power

plant configuration giving the best economic performance had an apparent BBEC of 70.5 mills/kW-hr. at a plant capacity (PCF) of 0.30 where 3 mills/kW-hr. has been added to account for the expense of operations and maintenance.

The economic results obtained are predicated on both optimistic and conservative assumptions. The key optimistic assumption is that collector (heliostat field) cost is \$60/m². For concentrator systems with two axis tracking DeMeo and Bos⁴ estimate collector costs as \$80/m² to \$90/m². A degree of conservatism in costing has been introduced by using a dry cooling heat rejection subsystem. This approach was chosen for environmental, siting, and water conservation reasons. This subsystem is more costly than either wet cooling or once-through subsystems. More importantly, both photovoltaic and thermal electric performance are degraded by the relatively high heat rejection temperature of dry air. The magnitude of this loss has not been quantified, but it is especially severe for low temperature systems of this type.

A rough comparison of the combined photovoltaic/thermal electric power plant with an photovoltaic-only power plant with cooler solar cells showed the BBEC of the former to be 9 to 10% lower. This important comparison merits a more detailed evaluation.

3.0 BASELINE POWER PLANT

3.1 CONCEPT SELECTION

This section provides the basis and rationale for choosing the particular power plant configuration with which to conduct the performance and economic analyses. This baseline plant was not expected to be the "optimum" choice, but is believed to be a reasonable one -- and is a choice that can be characterized as to performance and cost. Many of the decisions made in selecting the subsystems comprising this plant are admittedly subjective.

Figure 3-1 shows a block diagram of the major elements of the power plant, and shows schematically how the two types of electric energy are generated and combined. The major elements of the thermodynamic cycle are the four subsystems shown: preheater/boiler, turbine, heat rejection and thermal storage.

Two concepts were considered for this thermodynamic cycle -- single loop and dual loop (Figure 3-2). The former uses the same fluid both as coolant and as working fluid; the latter uses different fluids for each function. Additionally, there were a number of alternatives considered for each subsystem. The alternatives evaluated and the basis for the decisions are summarized in Table 3-1.

A single loop power plant configuration was selected (Figure 3-3). This power plant employs the ability of AlGaAs/GaAs cells to achieve high conversion efficiency at high concentration ratios and elevated temperatures.¹ However, conversion efficiency falls as temperature increases and 200°C was considered the upper temperature limit if excessive cell degradation is to be avoided.

The cells are directly cooled by water flowing in an annular passage just inside the surface of the cylinder on which the array is mounted. This annular passage also serves as a boiler in which the cooling water is heated and, at some point in the flow, brought to boiling temperature. The resulting mixture of pressurized hot water and steam is further heated and emerges from the upper end of the boiler as wet steam.

Figure 3-1
 Generic Photovoltaic/Thermal-Electric System

Figure 3-2
Comparison of Single and Dual Loop Systems

Table 3-1

**Thermodynamic Cycle -
Baseline System Definition**

ITEM	ALTERNATIVE ACCEPTED	ALTERNATIVE(S) REJECTED	BASIS FOR DECISION	PROBLEMS OR PENALTIES
• FLUID LOOP CONCEPT	SINGLE	DUAL	• HIGHER COST & EFFICIENCY LOSS DUE TO EXTRA HEAT EX- CHANGER IN DUAL CONCEPT	• SINGLE LOOP DOESN'T ALLOW USE OF APPRO- PRIATE FLUID FOR EACH FUNCTION
• WORKING FLUID	WATER	ORGANIC, AMMONIA	• WATER SUPERIOR FOR COOLING ARRAY • TECHNOLOGY FOR LARGE STEAM TURBINES MORE ADVANCED	• LARGE TURBINE • MOISTURE FORMATION IN TURBINE
• HEAT REJEC- TION SUBSYSTEM	DRY	ONCE-THROUGH, EVAPORATIVE	• LESS ENVIRONMENTAL IMPACT • COOLING WATER INCREASINGLY MORE DIFFICULT TO OBTAIN • EASIER SITING	• LOWER CYCLE EFFICIENCY • MORE COSTLY
• HEAT REJEC- TION CON- DENSATION METHOD	DIRECT (GEA)	INDIRECT (Heller)	• "SMALL" PLANT SIZE • WYODAK PLANT EXPERIENCE	• POSSIBLE SCALE- UP PROBLEM • LARGE DUCTING UNDER VACUUM
• HEAT REJEC- TION AIR MOVEMENT METHOD	MECHANICAL	NATURAL	• FLEXIBILITY TO MEET AMBIENT TEMPERATURE VARIATION • LOWER CAPITAL COST	• NOISE • ENERGY REQUIRED TO DRIVE FAN
• THERMAL ENERGY STORAGE SUBSYSTEM	SPRAY OR JET CONDENSER	SURFACE CONDENSER	• CYCLE EFFICIENCY • LOWER COST	• WATER PURITY AND CLEANLINESS STANDARDS MUST BE MET BY ALL STORED WATER

Figure 3-3
Photovoltaic/Thermal-Electric Plant Conceptual Design

This wet steam then flows to a steam drum where the water droplets settle out. The hot water from the steam drum is returned to the boiler at about the point where boiling begins (top of the "preheater") and the steam is either delivered directly to a low pressure turbine or is recondensed and stored as hot water in the storage tank. When thermal energy from the storage tank is needed, hot water from the top of the tank is flashed to steam and this steam is used to drive the turbine. After passing through the turbine, the steam is directed to the condenser/cooling tower, condensed to hot water, and pumped back to the boiler. The D.C. electricity generated by the AlGaAs/GaAs cell assembly is converted to AC by power conditioning equipment, combined with AC power from the turbine, and delivered to the load.

The flow details of the selected thermal system are shown schematically in Figure 3-4. The unbroken line shows the fluid path when the steam leaving the central receiver drives the turbine directly, with neither energy storage nor energy extraction. This mode of operation constitutes a closed cycle with no change in internal energy. The dashed line shows the fluid path for energy storage. Here water is removed from the condensate storage tank, boiled in the central receiver, and recondensed in a direct contact spray condenser. Condensation is achieved by mixing the steam directly with colder water. The resultant hot condensate is introduced at the top of the temperature stratified water storage tank. The thermal energy so stored can be extracted at some later time as shown by the dot-dash line. The hot water is expanded through a Joule-Thompson (throttling) valve to a pressure equal to the vapor pressure of the colder water. The resultant steam-water mixture, again in the ratio of 1 part steam to about 40 parts water, is separated, with the steam sent to the turbine inlet and the water pumped into the bottom of the storage tank. The turbine exhaust steam is condensed and returned to the condensate storage tank.

3.2 THERMOELECTRIC SUBSYSTEM SELECTION

This section presents the major subsystem considerations utilized in structuring the system configuration.

Figure 3-4
Thermodynamic Cycle Schematic

3.2.1 Working Fluid

For the high concentration ratios being considered the heat flux on the array surface is extremely high and one needs to be concerned about reaching critical heat flux levels⁵. A high performance coolant is required. Water, because of its high thermal conductivity and heat of vaporization, and its moderate viscosity, is an excellent coolant. Organic liquids such as Refrigerant 11 (trichloromonofluoromethane) and isobutane, because of low thermal conductivity and heat of vaporization, necessitate much higher pressure drop and pumping power than water for the same heat transfer coefficient. Ammonia, while having better cooling properties than the organics, requires system pressures of the order of 1000 psia. This would require an appreciably stronger and heavier structure to cool the array.

While water is superior as an array coolant, it has serious deficiencies when used as a turbine working fluid. Cycle efficiency studies show that for the allowable array temperature ($\leq 200^{\circ}\text{C}$), if water is sufficiently superheated to be essentially dry during the expansion process, ideal thermodynamic cycle efficiency is only about 10%. Expansion from the saturated vapor condition into the two phase region can yield theoretical efficiencies of about 20%. However, the turbine must be designed to operate in an erosive, wet environment.

On the other hand the shape of the saturated vapor line for some of the organics is such (nearly parallel to the isentropic lines) that expansion can be achieved with very little or no superheat. Thus ideal cycle efficiencies of the order of 16% can be achieved with such fluids. McDonald and Vepa⁷ argue that ammonia is preferable to the organics as a working fluid, since comparable, ideal, dry expansion efficiencies can be achieved with a marked reduction in turbine size. While these advantages are considerable, the steam turbine is in a very advanced state of development, and experience with large ammonia or organic turbines is limited. Hence, for the preliminary analyses of this study a single loop system with water as the cooling and working fluid was selected.

3.2.2 Heat Rejection Subsystem

The major alternatives for heat rejection are once-through cooling, wet (evaporative) cooling towers, and dry cooling towers. Once-through cooling requires location near and utilization of a natural, large body of water. This approach limits site location. A wet cooling tower, retains some of the advantages of the once-through system by virtue of the lowered sink temperature achieved by evaporation of water. Its principal disadvantages are humid environment, maintenance problems, and the cost and availability of water. The last problem is felt to be the most significant. Water is a precious resource in many areas of the world. Low rainfall areas such as the Southwestern United States are subject to periodic droughts. Moreover, even where water is not scarce, political and environmental factors may effectively prevent the use of existing water for power production.

Dry cooling towers have two principal disadvantages. They are costly because of large heat exchange surface area requirements and they result in lower overall cycle efficiency. These problems are caused by low air-side heat transfer coefficients and the high temperature of dry air compared to humidified air or water.

Weighing all of the above it was decided to choose the dry cooling tower system. For the thermodynamic cycle under consideration, a low source temperature bottoming cycle, this choice entails a particularly large efficiency penalty. Thermodynamic efficiency and combined power plant efficiency would certainly be much improved if once-through or wet tower cooling were used.

Within the context of a dry cooling system, additional decisions need to be made. It is necessary to choose between direct and indirect condensation, and between natural and mechanical draft. The direct GEA (Gesellschaft fuer Luftkondensation) system⁸ utilizes an extended surface air-cooled condenser in which the turbine exhaust steam is condensed directly (illustrated in Figure 3-2). Large ducts are required to convey the exhaust steam to the exchanger coils. The indirect or Heller system⁹ uses a jet condenser (Figure 3-5). Cool water from the tower is sprayed

Figure 3-5
Indirect (Heller) Condensation

into the jet condenser where it mixes with and condenses the exhaust steam from the turbine. Large circulating water pumps recycle most of the heated condensate to the dry cooling tower, with an amount equal to the exhaust steam being pumped back to the boiler circuit. These two systems have been compared by many investigators¹⁰. It has been found that as power plant size increases, the size and cost of the transport duct between the turbine exhaust and the condenser eventually causes the direct condenser system to be more expensive than the indirect one. As plant size decreases the complexity of the indirect system eventually causes it to become more expensive than the direct one. For conventional power plants, crossover occurs at power levels of 100 to 300 MW, corresponding to steam flow rates of about 2/3 to 2 million lbs/hr. At the maximum concentration ratio now being considered, 1000 suns, the required steam flow rate is about 1 million lbs/hr. As there is no clear advantage for either system, it was decided to use the one most favored in the United States¹¹, the direct GEA design. This design was used in the 330 MW Wyodak coal fired power plant started up in 1977 in Wyoming.

Two techniques are generally used to move the air through the condenser heat exchanger and through the tower. Natural draft utilizes the tower height to produce a chimney effect to move the air. Mechanical draft uses a fan at the top of a shorter tower to pull the air through the condenser and discharge it from the tower. The pros and cons of the two techniques have also been studied by numerous investigators, 9, 12, 13. Christopher and Forster⁹ claim that the scale effect is most important. They argue that as the natural draft tower requires the full height regardless of power output, it is preferred for large power plants. They give the crossover power level for conventional power plants as about 100 MW, corresponding to steam flow rates of 2/3 million lbs/hr. Again, this is less than the 1 million lbs/hr anticipated for the photovoltaic bottoming cycle. However, Janzon and Underwood¹² prefer mechanical draft systems up to 225 MW for wet towers and Leung et. al.¹³ find natural and mechanical draft to be competitive for dry towers at a power level of 800 MW. As no consensus emerges from the prior art, the choice is made based on

the special requirements of the photovoltaic bottoming cycle application. Wigham¹⁴ points out that mechanical draft provides the flexibility, associated with auto-variable-pitch fans, and multiple speed motors. By varying the air mass flow rate through the condenser a means is provided to compensate for daily and seasonal variation in ambient temperature. Indeed this technique is used in practice by GEA¹⁵. Thus the mechanical draft system was chosen as baseline in this work, and although a variable air mass flow rate technique was not used in the work that follows, such a technique is clearly practical and should be implemented in future studies.

3.2.3 Thermal Energy Storage Subsystem

An hot water storage system that uses a direct contact spray or jet condenser (Figure 3-6) was chosen as the baseline. Steam leaving the boiler portion of the tower is condensed by direct mixing with a large flow of relatively cold water from the bottom of a stratified storage tank. Spray condensers, while not as common as surface condensers, are in fairly wide use. However, a detailed comparison between spray and surface condensers was not attempted. The spray condenser is considered to be potentially less expensive and more efficient; less material is required and heat transfer is by direct mixing. However, a storage system based on the use of a spray condenser requires that an enormous quantity of water, the total capacity of the storage tank, be suitable for use in the boiler and turbine. If a surface condenser is used, the water system is physically separated from the working fluid system and only the small amount of water in the latter system need be treated to assure adequate purity and cleanliness. It may well be that future studies will show the above concerns to be of overriding importance. Because the situation is indeterminate at present the spray condenser was chosen for purposes of this evaluation.

3.3 THERMOELECTRIC SUBSYSTEMS ANALYSIS

In this section the various thermal electric subsystems -- pre-heater/boiler, turbine-generator, heat rejection, and thermal energy storage -- are physically defined and their operation is described. Each subsystem is mathematically characterized with the equations used in the program.

Figure 3-6
Thermal Storage Using Direct Contact Spray Condenser

3.3.1 Preheater/Boiler

This subsystem is critically important to plant operation. The intense concentrated thermal flux must be removed to prevent array overheating. However, the temperature of the fluid must be maintained as high as possible to achieve worthwhile efficiency from the thermal electric portion of the power plant. Thus, subsystem must operate with the constraint that array temperature is close to but is nowhere greater than 200°C.

The receiver is divided into preheater and boiler sections with the array cooled by an annular flow as shown in Figure 3-7. Preceding inward from the solar cells the thermal resistance encountered are those of the solar cell, the interface, the outer annular wall, fluid fowling, and fluid heat transfer. Typical conductances are: the cell, $C_c = 28,000$ watts/m² °C; the outer wall, $C_w = 55,000$ watts/m² °C; fluid fowling, $C_f = 23,000$ watts/m² °C; and fluid heating, $h = 23,000$ watts/m² °C, with an overall heat transfer coefficient, U , of about 7100 watts/m² °C. Here U is defined as

$$U = \frac{1}{\frac{1}{C_c} + \frac{1}{C_w} + \frac{1}{C_f} + \frac{1}{h}} \quad (3.3-1)$$

The fluid heat transfer coefficient, h , is different for the preheater and boiler and, moreover, varies within each section due to changes in fluid properties with temperature and quality. For the preheater h is given by the Colburn form of the Dittus - Boelter equation

$$h = 0.023 C_p G_l (Re_l)^{-0.2} (Pr_l)^{-0.6} \quad (3.3-2)$$

All symbols are defined in the nomenclature.

Figure 3-7
Tower Configuration

For the boiler the Chen Equation⁶ is used to predict the heat transfer coefficient, viz.

$$h = h_{\text{micro}} + h_{\text{macro}}$$

$$h = 0.000122 \frac{k_l^{0.79} C_p^{0.45} \rho_l^{0.49} g_c^{0.25}}{\sigma^{0.5} \mu_l^{0.29} \lambda^{0.24} \rho_v^{0.24}} (T_w - T_s)^{0.24} (\Delta P_T)^{0.75} S + h_l F \quad (3.3-3)$$

The boiling heat transfer is due to the additive effects of microconvective heat transfer due to nucleation, growth and separation of vapor bubbles, and microconvective heat transfer due to bulk fluid movement. The suppression factor, S , is a function of F which in turn is a function of the Martinelli¹⁶ parameter, X_{tt} . This last parameter is important in determining pressure drop in two phase flow. For vapor fractions, x , above about 0.7 wall dryout tends to occur and the critical heat flux may be exceeded. Therefore, exit quality from the receiver is maintained at 0.7 by the use of a steam drum and a liquid return line.

For the preheater the pressure drop is the sum of the frictional and hydrostatic pressure drops as given by:

$$\Delta P_f = \frac{G_l^2}{2\rho_l} \frac{2fL_p}{t} \quad (3.3-4)$$

and

$$\Delta P_H = \rho_l L_p g / g_c \quad (3.3-5)$$

The Fanning friction factor, f , is a function of Reynolds number as given in Perry's Handbook¹⁷.

For the boiler the pressure drop is again the sum of the frictional and hydrostatic losses now given by

$$\Delta P_f = \Delta P_l \phi_l^2, \text{ based on liquid}$$

or

$$\Delta P_f = \Delta P_v \phi_v^2, \text{ based on vapor} \quad (3.3-6)$$

$$\Delta P_h = R_v \rho_v + (1-R_v) \rho_l L_B \quad (3.3-7)$$

The Lockhart-Martinelli correlation¹⁶ is used with these equations to determine boiler pressure drop.

To achieve the desired heat transfer coefficients for the preheater and boiler, annulus gap thicknesses of about 0.15 cm. and 1.25 cm., respectively, were required. The small dimension of the preheater annulus will likely provide fabrication difficulties. Should this occur an alternative to this very narrow annulus gap is proposed. A wider gap could be used in conjunction with a turbulence promoter, to increase heat transfer rates. This device could be simply a ribbed wall. Pressure drop for this design will probably be greater than for the unobstructed narrower annulus. However, preheater pressured drop is not a problem as, for the flow rates under consideration, pump outlet pressure can be made quite large without requiring a significant amount of electric drive power. Pressure loss in the boiler while having only a minor effect on pump power requirements does degrade the thermodynamic availability of the working fluid. Fortunately the heat transfer coefficient in the boiler can be maintained sufficiently large by the microconvective contribution (Equation (3.3-3)) and hence a narrow gap to increase microconvective heat transfer is not required. Pressure drop in the boiler is typically about 70,000 Newtons/m² (10 psi).

3.3.2 Array Preheater/Boiler

In the preheater portion of the receiver, water is heated from pump outlet conditions until boiling occurs. The thermal energy absorbed by a differential element of water is

$$dQ^P = \left[F_s C (\alpha - \eta_{sc} \eta_{pf}) - h_a (T_{sc} - T_i) \right] dA^P \quad (3.3-8)$$

where cell efficiency is given by

$$\eta_{sc} = a + b T_{sc} \quad (3.3-9)$$

The heat transfer coefficient, h_a , is an effective one, accounting for both convective and radiative losses from the receiver. These losses are small compared to the heat absorbed, because of the low cell temperature (less than 200°C) and high concentration ratios (about 1000) involved. Hence only the first term of Eq. (3.3-8) is significant. Values used for the above parameters are: $C=1000$, $\alpha = 0.77$, $\eta_{pf} = 0.9$, $h_a = 0$, $a = 0.257$, and $b = -5.28 \times 10^{-4} \text{ } ^\circ\text{C}^{-1}$. An energy balance on a fluid element gives

$$dQ^P = U (T_{sc} - T) dA^P = W C_p dT \quad (3.3-10)$$

and solving for cell temperature in terms of fluid temperatures gives

$$T_{sc} = T + \frac{W C_p}{U} \frac{dT}{dA^P} \quad (3.3-11)$$

By substituting Eqs. (3.3-9) and (3.3-11) into Eq. (3.3-8), and rearranging and integrating, the cell temperature can be eliminated and the temperature anywhere in the preheater, T , can be solved for in terms of the temperature entering the preheater T_1 . Thus,

$$T = \frac{D - (D - BT_1) e^{-BEA^P}}{B} \quad (3.3-12)$$

where

$$B = F_s C \eta_{pf}^b + h_a \quad (3.3-13)$$

$$D = F_s C (\alpha - \eta_{pf} a) + h_a T_1 \quad (3.3-14)$$

$$E = \left[WC_p + F_s C \eta_{pf}^b WC_p/U + WC_p/U \right]^{-1} \quad (3.3-15)$$

Here, h_a is an effective heat transfer coefficient accounting for the small effects of convection and radiation. It is required that the above parameters be constant over the preheater area. Solar array power is given by:

$$P^P = \int_0^{A^P} F_s C \eta_{pf} \eta_{sc} dA^P \quad (3.3-16)$$

Substituting (3.3-8), (3.3-9), and (3.3-11) into (3.3-16) and integrating gives

$$P^P = F_s C \eta_{pf} \left[\left(a + b \frac{D}{B} \right) A^P + b (1/B - WC_p E/U) ((D - BT_1)/(BE)) (e^{-BEA^P} - 1) \right] \quad (3.3-17)$$

Thus fluid and cell temperatures and array power can be determined locally. As certain of the above parameters vary with temperature, the preheater was divided into twenty-four nodes of equal surface area and (3.3-12) and (3.3-17) were applied to each node. For each node V and C_p are evaluated at the mean temperature for the increment. The required physical and thermodynamic properties have been coded in analytic or table look-up form. Pressure drop is calculated as well for each node using (3.3-4) through (3.3-7). The computer program has a routine to solve for the position where the onset of boiling occurs.

The thermal energy absorbed by the fluid in a differential element of the boiler, dA^B , is given by

$$dQ^B = \left\{ F_s C \left[\alpha - \eta_{pf} (a + b T_{sc}) \right] - h_a (T_{sc} - T_I) \right\} dA^B \quad (3.3-18)$$

An energy balance on a fluid element gives

$$dQ^B = U (T_{sc} - T) dA^B = W \lambda dx \quad (3.3-19)$$

Substituting (3.3-19) into (3.3-18) and integrating, the area required for a given change in quality can be determined.

$$A^B = \frac{W \lambda \Delta x}{U} \frac{(U + F_s C \eta_{pf} b + h_a)}{F_s C \left[\alpha - \eta_{pf} (a + b T) \right] + h_a (T - T_I)} \quad (3.3-20)$$

Solar array power generated in boiler area A^B is given by

$$P^B = \int_0^{A^B} F_s C \eta_{pf} (a + b T_{sc}) dA^B \quad (3.3-21)$$

Substituting (3.3-19) into (3.3-21) rearranging and integrating we obtain:

$$P^B = F_s C \eta_{pf} \left[(a + bT) A^B + b \frac{W \lambda \Delta x}{U} \right] \quad (3.3-22)$$

The boiler was divided into twelve nodes to account for the effect of quality changes on the above parameters.

Because of the reduced heat transfer coefficient for single phase gas flow, array temperature becomes excessive if superheating is attempted. Using the upper portion of the receiver as a superheater section without a solar array outer surface is not an economical use of the heliostat field. Furthermore as the Chen correlation (Eq. 3.3-3) is limited to qualities less than 0.7, where wall dry-out is not a problem, the fluid exiting the boiler is composed of 70% vapor, 30% liquid. The two phase exit fluid is sent to a settling chamber (steam drum) where the water and steam are separated, with the water recirculated to the bottom of the boiler and the steam (saturated at this condition) sent to turbine or storage. For a given tower size and configuration and exit steam quality the mass flow rate through the tower must be matched to the solar flux reaching the tower, F_s . This means that the mass flow rate varies as the insolation level changes during the day. As pressure drop and heat transfer rates are functions of the mass flow rate, a Newtonian integration procedure for several simultaneous equations was programmed to calculate the required mass flow rate for a given insolation level. Physically, flow rate variation can be achieved by using a three way valve with a bypass line from the pump's outlet to inlet.

3.3.3 Turbine-Generation Subsystem

The cycle operating conditions require a low pressure steam turbine that must operate under somewhat stringent conditions: Namely, the turbine must accept saturated steam at the inlet and discharge to a relatively high exhaust pressure. The former condition necessitates that the design be capable of handling wet steam, while the latter condition requires that the design contend with higher exhaust end loadings.

Inasmuch as the turbine is not an off-the-shelf design and presents development problems, turbine manufacturers were contacted to obtain cost and any availability information. The most pertinent response was received from General Electric's Industrial Sales Division¹⁸. The letter stated that "the low pressure steam conditions do not pose a significant problem from a design standpoint, however, the special construction is costly". The low pressure saturated steam inlet condition requires "stainless steel inlet gear, special moisture removal provisions in the low pressure stages, and very large piping to the unit. These and similar considerations amount to about a 30-40% premium for this design versus a typical 1200 psig/900°F inlet that would normally be associated with steam turbine generation in the 40-50 MW range. This design also calls for a higher back pressure than would be considered normal, adding another 10-20% (in cost)". The cost data supplied by General Electric was twice that estimated by Colorado State University/Westinghouse Company¹⁹ for a similar turbine generator.

Performance information for such turbines, especially in off-design operation, is not readily obtained. At the minimum, performance prediction equations should account for variation in mass flow rate, exhaust conditions, and moisture formation. After some consideration of the problem, a General Electric Report²⁰ by R. C. Spencer et. al. was adapted to accomplish this end. This report could not be used directly for, among other things, it treated a three section (high, intermediate, and low pressure sections) turbine -- whereas the turbine of interest is a low pressure one. Spencer's method starts with a base efficiency, comparable to isentropic efficiency, from which they subtract efficiency losses. The base efficiency varies from 84 to 93%, and appears to represent the best that can be obtained with well designed turbines operating under ideal conditions. A base efficiency value of 87.5% was used in this work.

Spencer found that a change in the throttle flow ratio affected only the high pressure turbine section. Their results were applied directly to the single section low pressure turbine here considered, by using curve 6 (for a pressure ratio of 6) of Figure 3-8. Such a technique is clearly an approximation, but is believed to be conservative in predicting performance. This is so because the pressure ratio for the low pressure turbine is nearer 18 than 6. Moreover, if throttle flow ratio drops below 0.2, flow is directed to storage rather than to the turbine. The intent in doing this is both to keep power plant efficiency high and to reduce dependence on Figure 3-8 data.

It is known from experience that turbine efficiency is reduced by about 0.75% for each 1.0% moisture²¹.

Finally, Spencer found that exhaust loss is a strong function of annulus velocity as shown in Figure 3-9. As the enthalpy change available to do work is small for this bottoming cycle, the losses of Figure 3-9 can seriously penalize cycle efficiency. The biggest factor causing exhaust velocity variation is ambient temperature variation. Such changes cause condensation temperature, steam specific volume, and, hence, annulus velocity to vary from the design point. This can be compensated for by varying the air flow rate through the condenser^{8, 15}, or even by varying the steam flow split between storage and turbine. Such strategies could be implemented in a final design but are not considered in this study.

The curves of Figure 3-9 show exhaust losses for various low back pressure turbines. The exhaust loss for the high back pressure turbine of interest is difficult to predict and probably is not given accurately by any of these curves. As the turbine would be of an advanced type for a power plant which would go on stream in the years between 1990 and 2000, it is assumed that exhaust loss will, by then, be relatively small and approximated by curve 1, of Figure 3-9.

Figure 3-8
Part-Load Efficiency Correction for Throttle Flow Ratio

Figure 3-9
Exhaust Loss vs. Annulus Velocity for Various Turbines

Spencer gives mechanical losses as a function of generator rating. In the range of interest for this study, generator ratings of 25 to 60 MVA, the mechanical loss, ℓ_m , is about 0.4 MW. Generator losses, ℓ_g , are determined from the expression

$$\ell_g = P_t \frac{K_1}{100} K_2 \quad (3.3-23)$$

where the factor K_1 and K_2 are given in graphical form by Spencer. Generator losses vary from 1 to 3% of rated generator power.

All of the above losses are included in the computer program's turbine-generator performance equations. Typical turbine-generator efficiencies are 75% at the design point and substantially lower in off-design operation. Thermodynamic cycle efficiency is about 12.4% at the design point and 11.8% for a full years operation.

3.3.4 Thermal Energy Storage

A schematic showing the storage and extraction of energy for the spray condenser thermal energy storage subsystem is shown in Figure 3-6. The hot water leaving the condenser is at a lower temperature than that of the steam leaving the boiler. This difference in temperature corresponds to the pressure drop required to effect condensation. Having chosen this pressure drop, usually about 2 psi, an energy balance gives us the required water/steam ratio, R , for a given cold water temperature:

$$Wh_b + WRh_c = W(1 + R)h_h$$

$$\text{or} \quad h_b + Rh_c = (1 + R)h_h \quad (3.3-24)$$

$$\text{and} \quad R = \frac{h_b - h_h}{h_h - h_c} \quad (3.3-25)$$

The stored energy is extracted by isenthalpically expanding the hot water through a throttling valve (Figure 3-10) from the storage pressure to a pressure equal to the vapor pressure of the cold water. The throttling process path is typically from about 58 down to about 35 PSIA corresponding to 290°F hot water and 260°F cold water. The quality at the end of the expansion is about 3 to 4%. After expansion the vapor and liquid are separated, with the vapor expanded through the turbine and the cold liquid pumped into the bottom of the storage tank. For the case of energy extraction, an energy balance around the throttling valve gives:

$$W_T (R_S + 1) h_h = W_T h_T + W_T R_S h_h$$

or

$$(R_S + 1) h_h = h_T + R_S h_h \quad (3.3-26)$$

and

$$R_S = \frac{h_T - h_h}{h_h - h_c} \quad (3.3-27)$$

For a given enthalpy of the steam leaving the boiler, storage system size and efficiency are determined by the spread between hot and cold storage temperatures. If a large difference is maintained the water/steam ratio and the required storage tank capacity are small. However, energy degradation and loss in efficiency associated with storage are great for this case, as the temperature, T_c , of the flashed steam going to the turbine from storage is now low. A small temperature difference improves storage efficiency, but requires a large storage tank capacity. This would be a trade study of interest for further system optimization.

For a given water/steam ratio, R , and temperature difference T_h minus T_c , the question of storage tank capacity is still open. It is likely more preferable to dump some thermal energy than to provide adequate

Figure 3-10
Expansion Process

storage for every condition. Dumping could be accomplished by routing the steam, which would have been stored, directly to the condenser through a throttled turbine bypass line. This is another trade study of importance to system optimization.

3.3.5 Heat Rejection

A transverse high fin heat exchanger arranged in banks, with air in crossflow is utilized for heat rejection. Such banks are used in the process industry and are similar to condensers used for dry cooling in the Rugeley power plant⁹ and in various plants by GEA^{8, 15}. This type of condenser was analyzed in detail with a computer subroutine which considered tube longitudinal and transverse pitch, tube diameter, and fin height and spacing for various numbers of tube rows. Air side heat transfer coefficient was determined from the correlation of Briggs and Young²² with air side pressure drop calculated using the friction factor correlation of Robinson and Briggs²³. Tube side film condensation heat transfer coefficients were obtained from Chato²⁴. Fluid fouling conductances of 2860 and 5555 watt/m²C were used for the air and condenser side, respectively.

A typical heat exchanger configuration used 2100 banks. Each bank was 1.83 m high and contained 1.90 cm. O. D. tubes, 1.98 m long, 3 rows deep, with a transverse pitch of 6.0 cm. Eleven fins per inch, 0.043 cm thick, and 1.9 cm high were used on the air side. Figure 3-11 shows the arrangement used by GEA-Gesellschaft fuer Luftkondensation for the Wyodak 330 MW coal fired power plant¹⁵ and assumed for the baseline plant in this study. The tube banks are mounted at the top of each tower just above the array of fans. The low pressure exhaust steam leaving the turbine is conveyed via large diameter ducts to headers at the top of each condenser. These large ducts are necessary in order to limit steam pressure and temperature drop from the turbine exhaust to the condenser. A pressure loss of 0.17N/cm² (0.25 psi), corresponding to a temperature drop of 1°C, was used in this investigation. That this was reasonable was shown by using the Lapple technique²⁵ for compressible flow with frictional effects. Duct diameter for a 110 meter length of duct was calculated as 2.8 meters, which is smaller than the 4.1 meters required for the Wyadok plant. However, the flow rate of steam for the plant under consideration is about 950,000 lb/hr compared to 1,900,000 lb/hr for the Wyodak plant.

Figure 3-11
Wyodak Air Cooled Condenser Arrangement

4.0 COMPUTER MODELING

4.1 Photovoltaic/Thermal-Electric Plant Simulation Approach

Simulation models were previously developed at Aerospace to simulate the performance of heliostat field optics of central receiver systems^{2, 3}. These basic building blocks along with insolation and weather tapes were used to structure the desired simulation of the power plant system described in Section 3. The main features of the optical simulation are two axis tracking of the sun; utilization of direct, but not diffuse, insolation; assumed uniform distribution of concentrated solar flux on the central receiver; and provisions to account for geometry, aiming, shading and reflectivity losses. A special computer subroutine accounts for geometry (i.e. cosine effects) by investigating the heliostat field and calculating a mean loss. Optical losses vary with time of day and time of year. A representative value for the sum of all optical losses up to but not including the receiver is 35%, i.e., optical train efficiency is 65% up to the receiver. Weather and insolation data were obtained from an available climatology data base.

The flow of the computations in the photovoltaic/thermal-electric plant model is illustrated schematically in Fig. 4-1. For each system simulation, the input parameters listed in the upper left portion of the chart are specified. The computation begins with a "system sizing" run (mode 1) that determines the combination of turbine-generator rating and photovoltaic power conditioner rating which meets the specified plant rating. For the input parameters listed, the system sizing run determines the insolation required at the receiver that just provides the rated plant output. This insolation is called the "nominal solar flux" and defines the various thermal-electric system operating modes (Section 4-1) and the power conditioner and turbine rating. With turbine-generator and power conditioner rating specified, the simulation then computes a set of tables in which the AC output of the generator, the DC output of the photovoltaic array, and the system electrical losses are given as functions of the ambient temperature and of the amount of insolation incident on the photovoltaic cell assembly. Separate tables are computed for each mode of operation in appropriate combinations.

Photovoltaic/Thermal - Electric Plant Simulation Model

Figure 4-1

Photovoltaic/Thermal-Electric Plant Simulation Model

The program then proceeds to carry out an hour-by hour computation of system performance on the basis of a magnetic tape record of hourly values for the direct normal insolation and ambient temperature at the site. An algorithm that describes the performance of the system optics (the heliostat field and central receiver) computes the hourly amount of insolation that reaches the cell assembly. Hourly values for the generator output (AC) and array output (DC) are then obtained by interpolation in the tables discussed in the preceding paragraph (after identification of the mode of operation and therefore of the appropriate table to be used). The DC array output is then used as an input to the previously developed photovoltaic system simulation model that computes line losses, power conditioning losses, and losses in the electric storage subsystem (if any). The AC turbine output is similarly corrected for losses and added to the AC output of the power conditioner to yield the total plant output. These outputs are also compared to the rated capacities of the AC generator and DC power conditioner and electrical or thermal energy is stored or dispatched according to algorithms previously established in the program, if storage has been provided.

These hourly results are then summed to provide data on the annual performance of the thermal-electric and photovoltaic portions of the system and of the system as a whole. This plant performance information, when combined with estimates of the capital cost of the plant and of the annual cost of operation and maintenance, together with assumed values for key economic parameters (system lifetime, discount rate, inflation rate, etc.) permits calculation of the levelized busbar cost of electric energy from the plant.

Weather and insolation data were obtained from a data tape for Phoenix, Arizona for 1973, in an hour by hour format.

4.2 Model Implementation

Figure 4-2 shows a schematic of the thermodynamic bottoming cycle with modes of operation identified and related to the previous operational description. The rationale for using these various modes of operation and the criteria used in switching from mode to mode is given in Table 4-1. Figure 4-3 gives a clearer picture of how the solar flux incident on the

Figure 4-2
Thermodynamic Cycle Modes of Operation

Table 4-1

Mode of Operation of Thermodynamic Cycle

NUMBER	NAME	CONTROL CRITERIA	RATIONALE
1	NOMINAL FLOW TO TURBINE, DIRECT SOLAR, NO STORAGE	NOMINAL SOLAR FLUX	THIS FLOW RATE USED FOR TURBINE AND SYSTEM DESIGN
2	ENERGY STORAGE, NO FLOW TO TURBINE	SOLAR FLUX LESS THAN CRITICAL LEVEL AND NO STORED THERMAL ENERGY AVAILABLE	TURBINE EFFICIENCY LESS THAN STORAGE EFFICIENCY FOR LOW FLOW RATES CORRESPONDING TO LOW SOLAR FLUX LEVELS
3	OFF-DESIGN FLOW TO TURBINE, DIRECT SOLAR, NO STORAGE	SOLAR FLUX LESS THAN NOMINAL BUT ABOVE CRITICAL	TURBINE EFFICIENCY GREATER THAN STORAGE EFFICIENCY FOR INTER- MEDIATE SOLAR FLUX LEVELS
4	DIRECT SOLAR PLUS THERMAL ENERGY STORAGE, FLOW TO BOTH TURBINE AND STORAGE	SOLAR FLUX GREATER THAN NOMINAL AND SUFFICIENT THERMAL ENERGY STORAGE CAPACITY AVAILABLE	INCREASES PLANT CAPACITY FACTOR, AND FOR A GIVEN COLLECTOR AREA MAY REDUCE COST OF ELECTRIC ENERGY GENERATION
12	DIRECT SOLAR PLUS THERMAL ENERGY EXTRACTION, FLOW THROUGH RECEIVER AND FLOW FROM STORAGE ARE COMBINED AND SENT TO TURBINE	SOLAR FLUX LESS THAN CRITICAL AND STORED ENERGY AVAILABLE	OPERATIONALLY IT IS SIMPLER TO COMBINE FLOWS THAN SEND RECEIVER STEAM TO STORAGE AND EXTRACT LATER. IN THE LIMIT AS SOLAR FLUX GOES TO ZERO (nighttime) THIS MODE BECOMES ENERGY EXTRACTION ONLY
14	DIRECT SOLAR PLUS DUMP- ING, PORTION OF FLOW ORDINARILY STORED IS EXPANDED ISENTHALPICALLY	SOLAR FLUX GREATER THAN NOMINAL AND INSUFFICIENT THERMAL ENERGY STORAGE CAPACITY AVAILABLE	IT IS NOT ECONOMICAL TO MAKE STORAGE SUFFICIENTLY LARGE TO ALWAYS AVOID DUMPING

Figure 4-3
Relationship Between Incident Solar Flux on Receiver
and Thermodynamic Cycle Modes of Operation

receiver and these thermodynamic cycle modes of operation are related. The "nominal" solar flux in Figure 4-3 is that value of solar flux incident on the receiver which just yields a net plant output of 100 MW under nominal or rated plant operating conditions. The "critical" solar flux has been set at 0.2 of the nominal flux. At incident solar fluxes on the receiver between the critical and nominal values, the thermodynamic cycle is operated in mode 3. This mode is shown by the solid line in Figure 4-2.

As the mass flow rate through the receiver is matched to the level of incident flux, the flow rate decreases as the flux decreases from the nominal to the critical value. Figure 3-8 indicated that turbine efficiency decreases as mass flow rate decreases, and suggests that at flow rates below 0.2 of the nominal flow it is more efficient to store the thermal energy of the steam as hot water than to flow directly to the turbine.

If any storage capacity has been specified, then at fluxes below the critical value the thermodynamic cycle is operated in either mode 12 or mode 2 depending upon whether or not there is thermal energy in storage. If there is no thermal energy in storage, operation is in storage mode 2, as shown by the dashed line in Figure 4-2. Here steam is condensed by water pumped from the bottom of the storage tank and the thermal energy is stored as hot water at the top of the tank. Energy is extracted from storage during operation of the thermodynamic cycle in mode 12 to augment the subcritical amount of direct boiler energy. This mode is indicated by a combination of the solid line and the dot-dash line in Figure 4-2. Due to the mixing necessary between the two steam sources some decrease in turbine output results (~25%). In the limit condition of no incident solar flux on the receiver (nighttime), flow is only from storage and mode 12 is described by the dot-dash line only. During energy extraction, hot water from storage is expanded isenthalpically to a lower temperature and pressure. The liquid and vapor are separated, with the liquid, which represents about 96% of the mass flow from storage, pumped back into the bottom of the storage tank. This direct hot water return conserves the thermal energy in storage. The separated vapor flows into the turbine mixed with the throttled vapor from the central receiver. The intent here is to maintain the flow to the

turbine close to the design point where energy conversion efficiency is highest. This somewhat compensates for the degradation in available energy which occurs when these two streams, at different temperatures and pressures, are mixed. The turbine exhaust flows to the condenser and the liquid returns to the receiver or is held in the condensate storage tank as required to maintain the proper receiver liquid level.

At fluxes above the nominal value the thermodynamic cycle is operated in either mode 4 or mode 14 depending upon whether or not there is sufficient thermal energy storage capacity. If storage capacity is available the flow from the receiver is split between flow to the turbine and flow to storage, constituting mode 4 operation. In this and in mode 14 the flow rate through the turbine is maintained at the nominal rate of mode 1. The dual intent of mode 4 operation is (a) to increase plant capacity factor (PCF), where PCF is the actual energy output of the plant divided by the product of plant rated output (100 MW) and the numbers of hours in the year (8670 hours) and (b) for a given collector area to minimize the cost of generation of electric energy. The premise underlying intent (b) is that the cost of the turbine-generator and heat rejection subsystems are fixed regardless of how many hours a year they are operated. Thus for a given collector area the cost of the thermal energy storage subsystem must be weighed against the value of the electrical energy generated by providing this subsystem.

Mode 14 is the alternative to mode 4 when the specified thermal energy storage capacity is full, i.e. there is no "cold water" left in the storage tank with which to condense any more steam. In mode 14 the flow that would have been routed to storage is expanded isenthalpically and mixed with the turbine exhaust flow before entering the condenser. This energy dumping mode of operation penalizes system performance in two ways: (a) the thermal energy dumped is not utilized to produce electrical energy, (b) the heat load on the condenser is increased, raising turbine exhaust pressure and decreasing the power obtained from the nominal flow to the turbine. The premise underlying inclusion of this penalty is that the

turbine-generator uses a simple governor technique which limits turbine flow to the nominal value. As the generator is undoubtedly producing less than rated power under these exhaust conditions, it may be possible in subsequent studies to use a governor technique which permits higher turbine flow rates than the nominal value and hence increased power plant output in mode 14 operation.

4.3 COST ANALYSIS

4.3.1 Collector and Receiver

The philosophy used in this cost analysis has been to use currently achievable costs for all elements of the power plant except the heliostat field (collector) and the gallium arsenide solar cell covered receiver. For these two elements cost goals have been used. Table 4-2 presents in summary form the capital cost relationships used for the various subsystems. All costs are in 1978 dollars. A cost of $\$60/\text{m}^2$ was used for the collector. This is lower than the 92 to $\$104/\text{m}^2$ range recommended by DeMeo and Bos⁴ for two axis tracking heliostat subsystems, and assumes significant improvements will be achieved by the 1990's. A cost of $\$10,000/\text{m}^2$ was used for the gallium arsenide solar cell covered central receiver. This cost value is in fairly wide use, but can only be achieved on a mass production basis⁵.

4.3.2 Turbine-Generator

The cost of the turbine-generator building was obtained by interpolating between costs for a 10 megawatt (MW)²⁶ and a 1000 MW²⁷ plant, as shown in Figure 4-4. The cost of the turbine-generator was obtained from General Electric's Industrial Sales Division¹⁸. General Electric states that "the low pressure steam conditions do not pose a significant problem from a design standpoint, however, the special construction is costly." The low pressure saturated steam inlet condition requires "stainless steel inlet gear, special moisture removal provisions in the low pressure states, and very large piping to the unit. These and similar considerations amount to about a 30-40% premium for this design versus a typical 1200 psig/900°F inlet that would normally be associated with steam turbine generation in the 40-50 MW range. This design also calls for a higher back pressure than would be considered normal, adding another

Table 4-2: Capital Cost Relationships

<u>Subsystem</u>	<u>Cost Element</u>	<u>Parameter</u>	<u>Capital Cost Relationship</u> <u>1978 \$ Cost =</u>
Collector	Entire Subsystem (Mirrors, structure, motors, tracking sensor and mechanism, etc.)	Area, A_c sq. meters	$60A_c$
Receiver	Entire Subsystem (Gallium Arsenide solar cells, structure, etc.)	Area, A_r , sq. meters	$10,000 A_r$
Turbine-Generator	Turbine-Generator Bldg.	Rated gen. output, P_t , Kilowatt	$3685 P_t^{0.56}$
	Turbine-Generator	Rated gen. output, P_t , KW Turbine annulus area, E_A , sq. meters	$1730 P_t^{0.75} + 211,000 E_A$
Heat Rejection	Entire Subsystem	Bare tube surface area, A_T , sq. m	$740 A_T^{0.89}$
Photovoltaic Electric Energy Storage & Conditioning	Storage Batteries	Storage Capacity, S_E , Kilowatt-hr	$40 S_E$
	Power Conditioner (DC to AC Inverter)	Rated photovoltaic output, P_p , Kilowatt	$100 P_p$

Table 4-2: Capital Cost Relationships (Cont'd)

<u>Subsystem</u>	<u>Cost Element</u>	<u>Parameter</u>	<u>Capital Cost Relationship</u> <u>1978 \$ Cost =</u>
Thermal Energy Storage	Hot water storage tanks	Capacity, V, gal.	(a) $(50.70) V^{0.65}$, $V < \frac{10^6}{2}$ (a) $(1.287) V^{0.93}$, $V > \frac{10^6}{2}$
	Tank insulation	Capacity, V, gal.	$18.1 V^{0.6}$
	Piping, valves, fitting, insulation		40% of cost of storage tanks
	Spray condenser		5% of cost of storage tanks
	Condensate storage tanks	Capacity, V, gal.	$52.9 V^{0.63}$
Miscellaneous	Main pump and electric motor drive	Flow rate, C, lb/hr Developed head, H, lbf/in ²	$0.0496 [CH]^{0.708} + \frac{[CH]^{1.025}}{9193}$
	Storage pump and steam turbine drive	Flow rate, C, lb/hr. Developed head, H, lbf/in ²	$0.0496 [CH]^{0.708} + 2.45 [CH]^{0.406}$
	Electric plant equipment	Rated gen. output, P _t , KW	$6225 P_t^{0.62}$
	Instrumentation		\$275,000

Figure 4-4
Cost of Turbine - Generator Building

10-20% piece-wise." The cost data supplied by General Electric¹⁸ reflects this 40-60% additional cost. Indeed, the installed costs for the 40 MW turbines from Ref. 18 - are about double the costs estimated by Colorado State University/Westinghouse Inc.¹⁹ for a similar turbine. While cost estimates developed by Colorado State University/Westinghouse Inc.¹⁹ were not used, the same form of cost relationship was adopted

$$C = K_1 P_t^{0.75} + K_2 E_A \quad (4.3-1)$$

The coefficients K_1 and K_2 were calculated using cost data from General Electric, P_t was set equal to 40,000 kW. E_A was calculated using the steam flow rates and back pressures given in Ref. 27 with an annulus velocity of 400 ft/sec. The cost relationship finally obtained is

$$C = 1730 P_t^{0.75} + 211,000 E_A \quad (4.3-2)$$

This result is shown graphically in Figure 4-5.

4.3.3 Heat Rejection Subsystem

Cost data were obtained for the entire heat rejection subsystem from the General Atomic Company²⁸ acting as agents for GEA of Germany. This subsystem includes heat exchangers, fans and motors, tower and structure, and exhaust steam ducting and condensate piping (Figure 3-12). From the data package sent by General Atomic cost estimates were made of the dry air, direct condensation heat rejection subsystems for three power plants: (a) Wyodak, 330 MW coal fired power plant built near Gillette, Wyoming by Pacific Power & Light Co. and Black Hills Power and Light Co., (b) Braintree, 85 MW combined-cycle gas turbine/steam turbine power plant to be built in the Boston metropolitan area by the Braintree Electric Light Department, (c) Barstow, 10 MW central receiver solar thermal power plant proposed to the Department of Energy to be built in Barstow, California. Results are shown in Figure 4-6 where cost in 1978 dollars is given as a function of heat exchanger bare tube surface area by a straight line data fit.

Figure 4-5
Cost of Turbine - Generator

Figure 4-6
Cost of Direct Dry Cooling Heat Rejection Systems

4.3.4 Photovoltaic Electric Energy Storage & Power Conditioning

Costs used for these subsystems are those used in previous photovoltaic energy studies at Aerospace Corp. For the storage batteries the cost used is \$40/kilowatt-hour of stored energy based on 100% depth of discharge. The power conditioning equipment which inverts the power from DC to AC and handles necessary control and distribution functions is priced at \$100/kilowatt of rated output.

4.3.5 Thermal Energy Storage

The thermal energy storage subsystem consists of hot water storage tanks; insulation; pipes, valves, and fittings; spray condenser; and condensate storage tanks. The hot water storage tanks, which are the most expensive element of this group, were priced using the estimation techniques of Guthrie²⁹. Cost of these pressure vessels is a function of pressure rating as shown in Figure 4-7. As storage pressure will be below 60 psi a value of 0.86 was used for the coefficient, a . Because of the difficulty of transporting large tanks, a 500,000 gallon capacity (about 20 ft. diam. by 150 ft high) was taken as the upper limit. Beyond this capacity, multiple tanks are required. The cost of insulation based on a two inch thickness is a small percentage of tankage cost, as shown in the figure. The cost of pipes, valves, and fitting is taken as 40% of tankage costs, based on Martin Marietta data²⁶. The spray condenser was estimated to cost 5% of the tankage cost. Cost of the low pressure condensate storage tanks was estimated from data provided by Guthrie²⁹.

4.3.6 Miscellaneous Components

The cost of the main pump and electric motor drive, and the storage tank pump with steam turbine drive were estimated using additional data provided by Guthrie²⁹, as shown in Table 4-2. Central electric power plants require a variety of electric plant equipment including switchgear, switchboards, protective equipment, electric structure and wiring containers, power and control wiring, and master controls. The cost for the above was estimated using data from Martin Marietta²⁶ and United Engineers and Constructors²⁷, as shown in Figure 4-8.

Figure 4-7
Cost of Storage Tanks and Insulation

Figure 4-8
Electric Plant Equipment Costs

Finally a flat charge for instrumentation of \$275,000 is assessed.

4.3.7 Levelized Bus Bar Cost of Energy Generation

The above direct capital costs are converted to a levelized bus bar energy cost (BBEC) in mills/kW-hr using the equation given by DeMeo and Bos⁴,

$$\text{BBEC} = \frac{(C) (IF) (FCR) (1000)}{E} + O \& M \quad (4.3-3)$$

where C is the direct capital cost for the entire power plant, \$

IF is a factor to account for indirect capital costs

FCR is the levelized fixed charge rate

E is the total electrical energy generated during the year

O & M is the levelized operations and maintenance cost

The indirect factor accounts for architect and engineering fees, contingencies, spare parts, and interest during construction. A value of 1.50 has been assumed. The levelized fixed charge rate accounts for amortization of the loan for direct and indirect capital over the life of the power plant, (usually 30 years) at the prevailing interest rate as well as accounting for income taxes and miscellaneous costs over the life of the plant. As per the recommendation of DeMeo and Bos⁴ FCR is set equal to 0.15. The total electric energy, E, (photovoltaic plus thermal electric) generated during the year is calculated by the computer program during each system simulation run. O & M is taken as 3 mills/kW-hr.

Results of an investigation to determine the preferred design and operating conditions for the subject power plant are presented in this section. In order to limit the range of this investigation to a tractable domain, certain parameters were maintained constant. Perhaps the most important of these, concentration ratio is held constant at 1000. This is about the maximum practical value realizable in practice. There is clearly a theoretical optimum concentration ratio, for while high concentration ratios mitigate the very high costs of the gallium arsenide solar cells, the temperature difference between the solar cells and the working fluid increases with concentration ratio. Thus for a fixed maximum allowable solar cell temperature, thermodynamic cycle efficiency falls as concentration ratio increases. Some exploratory computer runs made during the course of this study showed that a concentration ratio of 1000 was definitely superior to one of 500 for this type of power plant. However, this study can provide no other guidance on the question of optimum concentration ratio.

5.1 Simulation Without Storage

Parameters that were studied during the course of this investigation are:

- o Collector areas
- o Heat exchanger size
- o Mass flux of cooling air
- o Design ambient temperature
- o Thermal energy storage capacity
- o Storage tank temperature differential

The number of parameters under study required that a degree of engineering judgement be used in selecting conditions for computer simulation. To simplify the problem, power plants with no energy storage were investigated first. Results from the computer runs with no storage were then applied in studying power plants with energy storage.

5.1.1 Parametric Studies

Collector area, heat exchanger size, air mass flux, and design ambient temperature were studied in an iterative manner, with the optimum

performance domain being slowly discovered. Results of this investigation are shown in Figures 5-1 through 5-4. As air mass flux through the condenser heat exchanger increases both air side heat transfer coefficient and fan power requirements increase. The former effect reduces the steam condensation temperature increasing the energy delivered by the turbine. The latter effect constitutes a system energy debit. As Figure 5-1 shows, the optimum mass flux is 4.75 kg/sec-m^2 (3500 lbs/hr-ft^2), with levelized bus bar energy cost (BBEC) increasing fairly rapidly at mass fluxes below the optimum.

As condenser heat exchanger area increases, steam condensation temperature decreases with a concomitant increase in turbine energy output. Heat exchanger cost increases with the 0.89 power of tube surface area, with the optimum found (Figure 5-2) at $2.24 \times 10^4 \text{ m}^2$ ($2.41 \times 10^5 \text{ ft}^2$). In a manner analogous to that shown in Figure 5-1 BBEC rises fairly rapidly if the heat exchanger is undersized.

An investigation was conducted of power plant performance for turbines designed for various ambient temperatures. Results are shown in Figure 5-3. It is seen that a power plant with turbine designed for high ambient temperatures (summertime) performs better than one designed for a low ambient temperature (wintertime). The "wintertime" turbine design is more costly than the "summertime" design because the rated power and annulus area are greater, i.e. P_t and E_A in Equation (4.3-1) are greater. However, the year long performance (kW-hr of energy generated) of the turbine is poorer for the "wintertime" design than the "summertime". This is due mainly to the ability of the high ambient temperature design to achieve near rated power all year long, while the low ambient temperature design generates considerably less than rated power on hot days.

As the collector is such a large contributor to the total cost of the power plant (about 60% of the total cost for a near optimum plant with no storage) it is expected that BBEC will depend strongly on collector area. This is borne out by Figure 5-4. To the right of the optimum collector area, $0.8 \times 10^6 \text{ m}^2$, more solar energy falls on the collector than can be effectively used and it is necessary to dump considerable

Figure 5-1
Investigation of Air Mass Flow Rate

Figure 5-2
 Investigation of Heat Exchanger Size

Figure 5-3
Variation in Design Ambient Temperature

Figure 5-4
Investigation of Collector Area

energy, both thermal and electrical. Designs with collector areas less than $0.8 \times 10^6 \text{ m}^2$ do not often achieve rated power output during the year. Nevertheless, it is necessary to pay for a turbine, heat exchanger, and power conditioner which can achieve rated power, at least during the highest insolation levels of the year.

5.1.2 Detailed Results

The following results were obtained using optimized values of the various parameters investigated. In Table 5-1 results are given for the nominal, that is, rated (mode 1) performance of the power plant (100 MW output). The rated generator power is 38,000 kW, accounting for 36.5% of the nominal plant output. Fan and pump power are 509 and 277 kW, in total less than 1% of the rated plant output.

Photovoltaic array output is 66,000 kW. Power conditioner losses are taken as 5% of the array output. The "nominal" solar flux on the receiver after concentration (1000) and after optics losses is 596 kW/m^2 ($189,000 \text{ Btu/hr ft}^2$). For comparison purposes the peak flux on the receiver during the entire year is 734 kW/m^2 ($232,800 \text{ Btu/hr ft}^2$). A large portion of the time the plant is generating less than rated power and is operating in mode 3. Although operation above the nominal flux is not overly frequent, a significant fraction of the electrical energy generated is provided by this mode (mode 14). This is shown in table 5-3 on page 67.

The nominal steam flow rate is 122.7 kg/sec., almost one million pounds an hour. The steam undergoes a temperature change from 147.3 to 72.2°C (297.2 to 162.8°F) in expanding through the turbine. The large concentration ratio with resultant large receiver flux causes a significant temperature difference between the solar cells ($\sim 195^\circ\text{C}$) and the steam (147.3°C). As the ambient temperature for the nominal design case is 43.3°C , it is seen that the condenser heat exchanger initial temperature difference (condenser temperature minus initial air temperature) of 28.2°C (50.7°F) requires that the cooling fluid enter the receiver at 71.5°C . Thus the temperature range available to perform useful work is constrained at the upper and lower temperatures by heat exchanger temperature differences.

Table 5-1: Design Parameters, Physical Characteristics and Rated
(Nominal/Mode 1) Performance for Optimized Power Plant

DESIGN PARAMETERS

Concentration Ratio	1000	
Design Ambient Temperature	43.3°C	110°F
Condenser Heat Exchanger Bare Tube Area	$2.24 \times 10^4 \text{ m}^2$	$2.41 \times 10^5 \text{ ft}^2$
Air Mass Flux	4.75 kg/sec-m^2	3500 lb/hr ft^2
Collector Area	$0.8 \times 10^6 \text{ m}^2$	
No Energy Storage		

PHYSICAL CHARACTERISTICS & NOMINAL PERFORMANCE

Receiver Height	11.3m	
Receiver Diameter	22.6m	
Receiver Area	800 m^2	
Rated Generator Power	38,000 kW	
Rated Photovoltaic Power	66,090 kW	
Fan Power	509 kW	
Pump Power	277 kW	
Power Condition Losses	3305 kW	
Rated Net Power Plant Output	100,000 kW	
Steam Flow Rate	122.7 kg/sec	$973,650 \text{ lb/hr}$
Nominal Solar Flux on Receiver	596 kW/m^2	$189,000 \text{ Btu/hr ft}^2$
Temperature of Saturated Steam Leaving Boiler/Entering Turbine	147.3°C	297.2°F
Temperature of Turbine Exhaust Fluid	72.7°C	162.8°F
Condenser Temperature	71.5°C	160.7°F

Table 5-1: Design Parameters, Physical Characteristics and Rated
(Nominal/Mode 1) Performance for Optimized Power
Plant (Cont'd)

Pressure at Pump Outlet		111 psi
Pressure Leaving Boiler		64.2 psi
Turbine Exhaust Pressure		5.5 psi
Overall Heat Transfer Coefficient in Pre- heater Section of Receiver		Btu/hr ft ² °F
Fluid Heat Transfer Coefficient in Boiler Section of Receiver		4500 Btu/hr ft ² °F
Turbine Annulus Area	4.16 m ²	44.8 ft ²
Air Mass Flow Rate	2.06 x 10 ⁴ kg/sec	1.635 x 10 ⁸ lb/hr
Overall Heat Transfer Coefficient in Condenser Based on Bare Tube Area		105 Btu/hr ft ² °F
Condenser Heat Exchanger Face Area	7.61 x 10 ³ m ²	8.19 x 10 ⁴ ft ²

Continuing in Table 5-1 it is seen that the pressure drop in the preheater/boiler is 47 psi, which for the cooling fluid flow rates and specific volumes encountered results in only minor pump power requirements. The pressure leaving the pump is important in controlling the position in the receiver where boiling begins. This pressure ultimately controls the maximum solar cell temperature that is reached. Active control of pump outlet pressure is required to compensate for receiver solar flux variation.

Heat transfer coefficients in the preheater and boiler must be very high for photovoltaic power plants with high concentration ratios. This, of course, has been a design "driver" from the inception of this study as discussed in Section 3.1.

Finally, condenser heat exchanger face (frontal) area is 7610m^2 ($81,900\text{ft}^2$). This is on the order of that used for the Wyodak power plant. As the heat exchanger used was only three rows deep, it should be possible to reduce face area if required.

Table 5-2 gives top-level monthly and yearly power plant performance data in kilowatt-hours. The demand is merely the plant rating (100,000 kW) times the hours in each month and in the year. Monthly variation in incident energy on the receiver, and in photovoltaic (AC energy from conditioner) and thermal electric energy output (net AC energy from turbine) are shown. It is seen that 63.3% of the net energy is generated in the photovoltaic portion of the power plant. Energy that is dumped is just a small percentage of the total generated. If Table 5-2 is used to calculate solar cell efficiencies it should be remembered that cell packing factor is 0.9. If the table is used to calculate thermodynamic cycle efficiency it should be remembered that ~77% of the incident solar energy is absorbed at the receiver and that DC energy conversion must be accounted for as well.

Table 5-3 gives a break down of the performance of the thermodynamic bottoming cycle for the various modes of operation. It is seen that substantial energy is generated in mode 14, (above nominal) almost as much as in mode 3, although the power plant does not operate in this mode too often. This is because the flow rate to the turbine is at nominal value all during mode 14 operation, while flow rates well below nominal are used in

Table 5-2: Monthly and Yearly Performance of Optimized Power Plant with no Storage, kW-hr

	DEMAND	LOSSES	ENERGY INCIDENT ON RECEIVER	AC ENERGY FROM CONDI- TIONER	NET AC ENERGY FROM TURBINE
JAN	.7440E+08	0.	.3006E+08	.1062E+08	.5942E+07
FEB	.6720E+08	0.	.6810E+08	.8900E+07	.4984E+07
MAR	.7440E+08	0.	.8095E+08	.1059E+08	.5891E+07
APR	.7200E+08	0.	.1291E+09	.1626E+08	.9388E+07
MAY	.7440E+08	0.	.1534E+09	.1919E+08	.1119E+08
JUN	.7200E+08	0.	.1326E+09	.1653E+08	.9744E+07
JUL	.7440E+08	0.	.1288E+09	.1633E+08	.9649E+07
AUG	.7440E+08	0.	.1318E+09	.1674E+08	.9982E+07
SEP	.7200E+08	0.	.1267E+09	.1593E+08	.9407E+07
OCT	.7440E+08	0.	.1183E+09	.1503E+08	.8743E+07
NOV	.7200E+08	0.	.9072E+08	.1052E+08	.6022E+07
DEC	.7440E+08	0.	.8814E+08	.1174E+08	.6751E+07
YRLY TOTL	.8760E+09	0.	.1319E+10	.1685E+09	.9769E+08

TOTAL ENERGY FROM POWER PLANT	DEMAND MINUS POWR PLANT ENERGY	BACKUP ENERGY REQUIRED	ELECTRIC ENERGY DUMPED
.1656E+08	.5783E+08	.5783E+08	.3754E+05
.1388E+08	.5331E+08	.5331E+08	.1407E+06
.1648E+08	.5791E+08	.5791E+08	.1264E+06
.2565E+08	.4634E+08	.4634E+08	.3420E+06
.3038E+08	.4401E+08	.4401E+08	.3699E+06
.2632E+08	.4567E+08	.4567E+08	.1915E+06
.2598E+08	.4841E+08	.4841E+08	.1405E+06
.2672E+08	.4767E+08	.4767E+08	.1332E+06
.2538E+08	.4661E+08	.4661E+08	.1937E+06
.2378E+08	.5061E+08	.5061E+08	.2207E+06
.1654E+08	.5545E+08	.5545E+08	.1061E+06
.1849E+08	.5590E+08	.5590E+08	.2870E+04
.2662E+09	.6097E+09	.6097E+09	.2005E+07

mode 3. Thermal energy dumped is a very small percentage of that sent directly to the turbine. That fan and main pump energy are only a small percentage of the net turbo generator energy is obvious.

Table 5-4 gives cost, performance, and economic results for the optimized power plant without storage. Equipment costs are given in dollars, energy delivered in kilowatt-hours, and levelized bus bar costs in mills/kilowatt-hour. The major share of the cost is for the heliostat field (collector), \$48 million compared to a total for the plant of \$79.8 million. For this concentration ratio (1000), receiver cost is 16.7% of collector cost. The thermal electric portion of the power plant clearly is effective in reducing the cost of energy generation. Levelized bus bar cost of energy for the photovoltaic system above is 83.6 mills/kW-hr compared to a cost of 67.5 mills/kW-hr for the combined photovoltaic/thermal electric power. However, this result can be misleading. The important comparison would be between this combined power plant and a photovoltaic power plant with solar cells cooled below the 200°C temperature assumed in the baseline design concept. While a detailed comparison was not attempted in this study a rough, first order comparison is to be given subsequently.

5.2 Simulation with Thermal Energy Storage

Computer simulations were also conducted for various amounts of thermal energy storage over a range of collector areas. For all of these runs heat exchanger size, air mass flux, and design ambient temperatures were the same as for the non-storage runs, namely $2.24 \times 10^4 \text{ m}^2$, 4.75 kg/sec-m², and 43.3°C, respectively. That this is not strictly proper is obvious. Because of the efficiency loss associated with using thermal energy storage the temperature and pressure of the steam entering the turbine, when extracting energy from storage, have been lowered. This will likely change the preferred heat exchanger size and air mass flux. However, except where operation from storage is extensive this is probably a second order effect. It is hoped to explore this further in subsequent work.

Table 5-3: Thermal Electric Performance of Optimized Power Plant with no Storage
for Various Modes of Operation, kW-hr

	THERMAL ENERGY STORED		THERMAL ENERGY DIRECTLY TO TURBINE			
	MODE 2	MODE 4	MODE 3	MODE 4	MODE 14	
JAN	0.	0.	.36371662E+08	0.	.12291520E+08	
FEB	0.	0.	.28206525E+08	0.	.11680377E+08	
MAR	0.	0.	.30930340E+08	0.	.16457547E+08	
APR	0.	0.	.39634801E+08	0.	.32467566E+08	
MAY	0.	0.	.42166030E+08	0.	.43530956E+08	
JUN	0.	0.	.34283557E+08	0.	.41740008E+08	
JUL	0.	0.	.42317743E+08	0.	.33217322E+08	
AUG	0.	0.	.47219534E+08	0.	.30166059E+08	
SEP	0.	0.	.35960337E+08	0.	.36199241E+08	
OCT	0.	0.	.33893507E+08	0.	.32476941E+08	
NOV	0.	0.	.32321448E+08	0.	.14535410E+08	
DEC	0.	0.	.41725759E+08	0.	.11799524E+08	
YEAR	0.	0.	.44503184E+09	0.	.31656287E+09	

	THERMAL ENERGY DUMPED (MODE 14)	FAN ENERGY (MODES 3,4,12, AND 14)	MAIN PUMP ENERGY (MODES 2,3,4, 12 AND 14)	STORAGE PUMP ENERGY (MODE 12)	TOTAL TURBO GENERATOR ENERGY (MODES 3,4,12,14)
JAN	.85916965E+06	.16090034E+06	.56449547E+05	0.	.61592987E+07
FEB	.22193683E+07	.14889559E+06	.90324427E+05	0.	.52234623E+07
MAR	.25796957E+07	.18548461E+06	.12041173E+06	0.	.61973453E+07
APR	.75015757E+07	.21841341E+06	.31736846E+06	0.	.99233306E+07
MAY	.90715449E+07	.25911000E+06	.32409388E+06	0.	.11772515E+08
JUN	.63025755E+07	.23149005E+06	.20090222E+06	0.	.10176876E+08
JUL	.43763457E+07	.24258911E+06	.13408816E+06	0.	.10026108E+08
AUG	.43283780E+07	.23638797E+06	.14235703E+06	0.	.10361060E+08
SEP	.62648939E+07	.22058155E+06	.23024310E+06	0.	.98577033E+07
OCT	.68614712E+07	.20111880E+06	.28909521E+06	0.	.92333420E+07
NOV	.29814158E+07	.15069059E+06	.13942440E+06	0.	.63218540E+07
DEC	.10344761E+07	.16836396E+06	.72855506E+05	0.	.69925486E+07
YEAR	.54380911E+08	.24340259E+07	.21176221E+07	0.	.10224545E+09

	NET TURBO GENERATOR ENERGY FROM MODE 3	MODE 4	MODE 14
JAN	.44150153E+07	0.	.15269336E+07
FEB	.34890044E+07	0.	.14952379E+07
MAR	.37847699E+07	0.	.21066791E+07
APR	.51294375E+07	0.	.42581113E+07
MAY	.55168986E+07	0.	.56724124E+07
JUN	.44157976E+07	0.	.53286886E+07
JUL	.54467350E+07	0.	.42026977E+07
AUG	.61312248E+07	0.	.38510802E+07
SEP	.47052889E+07	0.	.47015897E+07
OCT	.44646865E+07	0.	.42784415E+07
NOV	.41196852E+07	0.	.19020538E+07
DEC	.52368990E+07	0.	.15144297E+07
YEAR	.56855443E+08	0.	.40838355E+08

Table 5-4: Capital Costs, Energy Delivered & Levelized Cost of Energy for Optimized Power Plant with no Storage

CAPITAL COSTS FOR THERMO ELECTRIC	1352435.16	
TURBINE-GENERATOR BLDG	5586500.94	6938936.10
SUB-TOTAL, TURBINE-GENERATOR		
COOLING SYSTEM	5510512.02	5510512.02
SUB-TOTAL, COOLING SYSTEM		
CONDENSATE STORAGE	0.00	
THERMAL STORAGE	0.00	
TANK INSULATION	0.00	
PIPING, VALVES, FITTING	0.00	
SPRAY CONDENSER	0.00	0.00
SUB-TOTAL, STORAGE SYSTEM		
CONDENSATE PUMP + ELECTRIC MOTOR DRIVE	53310.19	
STORAGE TANK PUMP + STRAIN TURBINE DRIVE	98307.24	151617.43
SUB-TOTAL, PUMPS AND DRIVES		
ELECTRIC PLANT EQUIPMENT	4301355.60	4301355.60
SUB-TOTAL, ELECTRIC PLANT EQUIPMENT		17177421.14
TOTAL CAPITAL COST, THERMO-ELECTRIC		

Table 5-4: Capital Costs, Energy Delivered & Levelized Cost of Energy for Optimized Power Plant with no Storage (Cont'd)

CAPITAL COST FOR PHOTOVOLTAIC COLLECTOR	48000000.00	48000000.00	
SUB-TOTAL, COLLECTOR			
ARRAY	7399689.97	7999689.97	
SUB-TOTAL, ARRAY			
POWER CONDITIONER	6608859.15	6608859.15	
SUB-TOTAL, POWER CONDITIONER			
ELECTRIC STORAGE	0.00	0.00	
SUB-TOTAL, ELECTRIC STORAGE			
MISCELLANEOUS	0.00	0.00	
SUB-TOTAL, MISCELLANEOUS			
TOTAL CAPITAL COST, PHOTOVOLTAIC		62608549.12	
ELECTRICAL ENERGY DELIVERED, PHOTOVOLTAIC			.16846954E+09
TOTAL CAPITAL COST, THERMO-ELECTRIC		17177421.14	
ELECTRICAL ENERGY DELIVERED, THERMO-ELECTRIC			.97693798E+08
TOTAL CAPITAL COST, PLANT		79785970.26	
TOTAL ELECTRICAL ENERGY DELIVERED, PLANT			.26616334E+09
LEVELIZED BUS BAR COST FOR PHOTOVOLTAIC EXCLUSIVE OF O+M		83.62	
LEVELIZED BUS BAR COST FOR THERMO-ELECTRIC EXCLUSIVE OF O+M		39.56	
TOTAL LEVELIZED BUS BAR COST FOR TOTAL POWER PLANT EXCLUSIVE OF O+M		67.45	
TOTAL LEVELIZED BUS BAR O+M COST FOR TOTAL POWER PLANT		3.00	
TOTAL LEVELIZED BUS BAR COST FOR TOTAL POWER PLANT		70.45	

An investigation of the effect of storage tank temperature difference on full year performance was conducted for a power plant with collector area of $1.2 \times 10^6 \text{ m}^2$ and storage tank capacity of 2.25 hours of turbine-generator operation. Storage tank capacity is here characterized as that required to provide flow to the turbine at the regeneration (mode 12) rate for a specified length of time. Three storage tank temperature differences were evaluated: 16.7°C (30°F), 27.8°C (50°F), and 41.7°C (75°F). It was found that levelized bus bar energy cost (BBEC) was substantially the same at temperature differences of 16.7°C and 27.8°C , and lower than that at the temperature difference of 41.7°C . It was therefore decided to use a storage tank temperature difference of 16.7°C in what follows.

Power plant performance at thermal storage capacities of 0.0, 2.25 hours ($58.5 \text{ kW}_e\text{-hr}$), 4.5 hours ($117 \text{ kW}_e\text{-hr}$), and 9.0 hours ($234 \text{ kW}_e\text{-hr}$), respectively hours were investigated. Results are shown in Figure 5-5. It is seen that the lowest BBEC is still obtained with no storage for a collector area of $0.8 \times 10^6 \text{ m}^2$. However, as collector area increases it is seen that power plants with storage yield lower values of BBEC than those without storage, although all such plants have a higher BBEC than does the optimum plant with no storage. However, plant capacity factor (PCF) is only 0.30 for this optimum plant. Increases in plant capacity factor can be achieved beyond 0.30, subject to the penalty of higher values of BBEC. Figure 5-6, a cross plot of Figure 5-5, shows what is only suggested in Figure 5-5 -- that for each collector size there is an optimum storage capacity. For a collector size of $1.8 \times 10^6 \text{ m}^2$ this optimum capacity is greater than nine hours. This certainly is not surprising, because for power plants with large collector areas, considerable thermal energy would have to be dumped if no storage were available.

The thermal energy storage used with this power plant costs roughly \$75/kW-hr of electricity. This value is established by the following calculation where the first factor is typical mode 12 performance,

$$\frac{947,000 \text{ lb steam}}{29,000 \text{ kW-hr}} \times \frac{30 \text{ lb water}}{1 \text{ lb steam}} \times 0.123 \frac{\text{gal.}}{\text{lb water}} \times \frac{\$0.62}{\text{gal.}} = \$74.71 \text{ kW-hr}$$

Cost of Electric Energy as Function of Collector Area for Various Thermal Storage Capacities

Figure 5-6

Levelized Busbar Energy Cost vs Thermal Energy Storage
Capacity for Various Collector Areas

i.e. about 32.7 pounds of steam flow to the turbine in mode 12 are required to generate one kW-hr of electrical energy. The second factor is the storage tank circulation ratio. The third factor is the specific volume of water and the fourth factor is a rough cost of thermal energy storage from Figure 4-7. Electric energy storage cost is almost half as expensive as thermal energy storage. For comparison purposes, tables analogous to Tables 5-2, 5-3, and 5-4 for the optimum power plant with no storage are given here as Tables 5-5, 5-6, and 5-7 for a power plant with storage. The design parameters for this power plant are: collector area = $1.2 \times 10^6 \text{ m}^2$, hours of turbine generator operation from storage = 4.5, design ambient temperature = 43.3°C , heat exchanger area = $2.24 \times 10^4 \text{ m}^2$, and air mass flux = $4.75 \text{ kg/sec} - \text{m}^2$.

5.3 Simulation with Electric Storage

Computer simulations were also conducted for various amounts of battery type electric storage over a range of collector areas. For all of these runs, too, heat exchanger size, air mass flux, and design ambient temperatures with the same as for the non storage runs. When excess DC electrical energy was available, over and above the power conditioner rating this excess energy was diverted to a storage battery system. The assumed cost of the storage battery system for this study was \$40/kW. 100% discharge of the batteries was assumed and the effective loss due to electric storage was set at 30%. When the solar flux from the cells dropped below the power level of the power conditioner, energy was removed from the battery storage and supplemented the cells to maintain full output conditions as long as the storage lasted.

Power plant performance at electric storage capacities of 2.25, (144 kW_e-hr) 4.5 (288 kW_e-hr) and 6.75 (432 kW_e-hr), respectively hours were studied. The results are shown in Figure 5-7. In this figure it is again obvious that the lowest BBEC is still obtained with no storage and for a collector area of $0.8 \times 10^6 \text{ m}^2$. Again, the pattern indicated with thermal storage is evident wherein minimum BBEC for a given storage increases directly as a function of collector area. It should be borne in mind that one hour of electric storage is equivalent to 2.6 hours of thermal storage, as explained below.

Table 5-5: Monthly and Yearly Performance of Typical Power Plant
with Storage, kw-hr

	DEMAND	ENERGY INCIPIENT ON RECEIVER	AC ENERGY FROM CONDI- TIONER	NET AC ENERGY FROM TURBINE	TOTAL ENERGY FROM POWER PLANT	DEMAND MINUS POWR PLANT ENERGY	BACKUP ENERGY REQUIREO	ELECTRIC ENERGY DUMPED
JAN	.7440E+08	.1201E+09	.1313E+08	.8562E+07	.2169E+08	.5269E+08	.5269E+08	.3037E+07
FEB	.6720E+08	.1022E+09	.1112E+08	.7305E+07	.1843E+08	.4876E+08	.4876E+08	.2598E+07
MAR	.7440E+08	.1214E+09	.1309E+08	.8626E+07	.2171E+08	.5267E+08	.5267E+08	.3183E+07
APR	.7200E+08	.1193E+09	.1281E+08	.8117E+07	.2118E+08	.5080E+08	.5080E+08	.2771E+07
MAY	.7440E+08	.1230E+09	.1242E+08	.1495E+08	.3737E+08	.3701E+08	.3701E+08	.7363E+07
JUN	.7200E+08	.1198E+09	.1207E+08	.1262E+08	.3169E+08	.4029E+08	.4029E+08	.6525E+07
JUL	.7440E+08	.1233E+09	.1244E+08	.1279E+08	.3223E+08	.4215E+08	.4215E+08	.5664E+07
AUG	.7440E+08	.1197E+09	.1244E+08	.1335E+08	.3329E+08	.4109E+08	.4109E+08	.5790E+07
SEP	.7200E+08	.1190E+09	.1274E+08	.1290E+08	.3164E+08	.4034E+08	.4034E+08	.5949E+07
OCT	.7440E+08	.1178E+09	.1272E+08	.1256E+08	.3029E+08	.4409E+08	.4409E+08	.5555E+07
NOV	.7200E+08	.1211E+09	.1294E+08	.8819E+07	.2175E+08	.5023E+08	.5023E+08	.3209E+07
DEC	.7440E+08	.1322E+09	.1470E+08	.9818E+07	.2432E+08	.5006E+08	.5006E+08	.3091E+07
YRLY TOTL	.8760E+09	.1979E+10	.2013E+09	.1353E+09	.3366E+09	.5392E+09	.5392E+09	.5823E+08

Table 5-6: Thermal Electric Performance of Typical Power Plant with Storage for Various Modes of Operation, kWh

	THERMAL ENERGY STORED		THERMAL ENERGY DIRECTLY TO TURBINE		
	MODE 2	MODE 4	MODE 3	MODE 4	MODE 14
JAN	.14012591E+07	.17696207E+08	.13570568E+08	.40400234E+08	0.
FEB	.11169584E+07	.15453993E+08	.13927977E+08	.29371506E+08	.17731443E+07
MAR	.18586205E+07	.18484803E+08	.14891999E+08	.34250888E+08	.27927291E+07
APR	.67766059E+06	.32762715E+08	.14276269E+08	.48187676E+08	.13649423E+08
MAY	.75429526E+06	.35017779E+08	.14854573E+08	.50249092E+08	.24871370E+08
JUN	.85368630E+06	.30889937E+08	.93731924E+07	.47346637E+08	.20889781E+08
JUL	.54492787E+06	.29180143E+08	.15311529E+08	.49466378E+08	.15342251E+08
AUG	.64156878E+06	.31326273E+08	.14320716E+08	.54246571E+08	.12558287E+08
SEP	.48593723E+06	.32311629E+08	.13752378E+08	.47687583E+08	.14129176E+08
OCT	.15460791E+06	.33010362E+08	.14254906E+08	.47896038E+08	.88295413E+07
NOV	.78975988E+06	.20633985E+08	.15163528E+08	.35003338E+08	.15161086E+07
DEC	.54081251E+06	.19829121E+08	.15956184E+08	.43708573E+08	C.
YEAR	.10217679E+08	.31659673E+09	.16965292E+09	.52781457E+09	.11635181E+09

	THERMAL ENERGY DUMPED (MODE 14)	FAN ENERGY (MODES 3, 4, 12, AND 14)	MAIN PUMP ENERGY (MODES 2, 3, 4, 12 AND 14)	TOTAL TURBO GENERATOR ENERGY (MODES 3, 4, 12, 14)	NET TURBO GENERATOR ENERGY FROM STORAGE (MODE 12)
JAN	C.	.15914180E+06	.21236973E+06	.89782118E+07	.19915486E+07
FEB	.63320169E+06	.14347817E+06	.22465042E+06	.77111211E+07	.17033824E+07
MAR	.15629828E+07	.16502063E+06	.28398219E+06	.91225707E+07	.21571889E+07
APR	.83729211E+07	.25269125E+06	.55761596E+06	.14056516E+08	.36074649E+07
MAY	.14350572E+08	.30592584E+06	.54419346E+06	.15884659E+08	.38775432E+07
JUN	.11672110E+08	.26300484E+06	.36351017E+06	.13321239E+08	.32782567E+07
JUL	.78480283E+07	.27677452E+06	.32553158E+06	.13461651E+08	.30009536E+07
AUG	.66957228E+07	.28508990E+06	.38072878E+06	.14088714E+08	.33201815E+07
SEP	.70520540E+07	.26771443E+06	.46017724E+06	.13703759E+08	.34801033E+07
OCT	.42858442E+07	.25117735E+06	.55093906E+06	.13444245E+08	.35537103E+07
NOV	.49732900E+06	.17255353E+06	.35081280E+06	.93818979E+07	.22719999E+07
DEC	0.	.18210439E+06	.32465824E+06	.10172817E+08	.21787212E+07
YEAR	.62970766E+08	.27246765E+07	.45791916E+07	.14332740E+09	.34421055E+08

	NET TURBO GENERATOR ENERGY FROM STORAGE		MODE 12 THERMAL ENERGY TO TURBINE	
	MODE 3	MODE 4	AFTER MIXING	FROM STORAGE
JAN	.16551802E+07	.49156875E+07	0.	.19354029E+08
FEB	.17546135E+07	.36165783E+07	.23080108E+06	.16359832E+08
MAR	.18720562E+07	.42383585E+07	.35888501E+06	.20509971E+08
APR	.18545850E+07	.60570558E+07	.16490077E+07	.33605793E+08
MAY	.19412946E+07	.63942108E+07	.27371081E+07	.35893386E+08
JUN	.12235757E+07	.59715963E+07	.121460965E+07	.31700297E+08
JUL	.19848257E+07	.62473561E+07	.15555350E+07	.29053990E+08
AUG	.18499464E+07	.68748391E+07	.13014582E+07	.32173661E+08
SEP	.17931472E+07	.69691421E+07	.15556790E+07	.32414344E+08
OCT	.18571322E+07	.61012634E+07	.10520762E+07	.33289295E+08
NOV	.18327279E+07	.44050954E+07	.19857334E+06	.21490108E+08
DEC	.19969084E+07	.54425179E+07	0.	.20826685E+08
YEAR	.21716093E+08	.66333701E+08	.12795620E+08	.32777139E+09

Table 5-7: Capital Costs, Energy Delivered & Levelized Cost
of Energy for Typical Power Plant with Storage

CAPITAL COSTS FOR THERMO-ELECTRIC		
TURBINE-GENERATOR BLDG	1309947.60	
TURBINE-GENERATOR	5398518.47	
SUB-TOTAL, TURBINE-GENERATOR		6697966.07
COOLING SYSTEM	5510512.02	
SUB-TOTAL, COOLING SYSTEM		5510512.02
CONDENSATE STORAGE	219281.28	
THERMAL STORAGE	4803385.90	
TANK INSULATION	350321.28	
PIPING, VALVES, FITTING	2045066.87	
SPRAY CONDENSER	255633.36	
SUB-TOTAL, STORAGE SYSTEM		7763688.69
CONDENSATE PUMP + ELECTRIC MOTOR DRIVE	74373.34	
STORAGE TANK PUMP + STRAIN TURBINE DRIVE	129524.28	
SUB-TOTAL, PUMPS AND DRIVES		203897.62
ELECTRIC PLANT EQUIPMENT	4152001.92	
SUB-TOTAL, ELECTRIC PLANT EQUIPMENT		4152001.92
TOTAL CAPITAL COST, THERMO-ELECTRIC		24603066.32

Table 5-7: Capital Costs, Energy Delivered & Levelized Cost
of Energy for Typical Power Plant with Storage (Cont'd)

CAPITAL COST FOR PHOTOVOLTAIC COLLECTOR	72000000.00	72000000.00	
SUB-TOTAL, COLLECTOR			
ARRAY	11999534.95	11999534.95	
SUB-TOTAL, ARRAY			
POWER CONDITIONER	6826836.86	6826836.86	
SUB-TOTAL, POWER CONDITIONER			
ELECTRIC STORAGE	0.00	0.00	
SUB-TOTAL, ELECTRIC STORAGE			
MISCELLANEOUS	0.00	0.00	
SUB-TOTAL, MISCELLANEOUS			
TOTAL CAPITAL COST, PHOTOVOLTAIC		90826371.82	
ELECTRICAL ENERGY DELIVERED, PHOTOVOLTAIC			.20131560E+09
TOTAL CAPITAL COST, THERMO-ELECTRIC		24603066.32	
ELECTRICAL ENERGY DELIVERED, THERMO-ELECTRIC			.13525489E+09
TOTAL CAPITAL COST, PLANT		115429438.13	
TOTAL ELECTRICAL ENERGY DELIVERED, PLANT			.33657049E+09
LEVELIZED BUS BAR COST FOR PHOTOVOLTAIC EXCLUSIVE OF O+M		101.51	
LEVELIZED BUS BAR COST FOR THERMO-ELECTRIC EXCLUSIVE OF O+M		40.93	
TOTAL LEVELIZED BUS BAR COST FOR TOTAL POWER PLANT EXCLUSIVE OF O+M		77.17	
TOTAL LEVELIZED BUS BAR O+M COST FOR TOTAL POWER PLANT		3.90	
TOTAL LEVELIZED BUS BAR COST FOR TOTAL POWER PLANT		80.17

Figure 5-7
Cost of Electric Energy as a Function of Collector Area for Various Electric Storage Capacities

5.4

Simulation with Thermal and Electric Storage

In an effort to establish the relative impact of both thermal and electric storage, the curves on Figure 5-8 were produced. It should be noted that the electric and thermal storage times and hours represent unequal amounts of storage in terms of kilowatt hours. An hour of thermal storage is equal to 26,000 kWh which is the amount produced by the turbine-generator when it is operated for 1 hour at the reduced output necessitated by the lower temperature level of the steam taken from the thermal storage tank (nominally 36 MW). On the other hand, due to the 64 megawatt output of a photovoltaic power conditioner, 1 hour of electric storage is equal to 64,000 kWh. This makes for somewhat uneven comparison between thermal and electric storage. Of particular interest is the sharply defined optimum point which shifts radically towards larger collector areas as storages increased.

5.5

Photovoltaic-Only Versus PVTE Power Plant

A comparison was made of combined photovoltaic/thermal electric power plant with photovoltaic only power plant. Only a rough comparison of these two power plants is attempted here. In this comparison the turbine-generator and the turbine generator building are removed, and that fraction of electric plant equipment attributable to the thermal electric portion of the plant is removed. Since the thermal-electric portion of the combined plant provides 38 MW out of a gross plant output of 104 MW, it is assumed that the electric plant equipment that can be removed is $38/104$ of the total cost. It is also assumed that the gallium arsenide cells are cooled as before, but the thermal system is operated below atmospheric pressure. This permits the coolant and the cells to operate at lower temperatures thereby providing higher cell efficiency than in the combined power plant. Since only about 10% of the thermal energy is converted to electricity by the turbine-generator, the amount of thermal energy to be removed is approximately the same for both the combined and photovoltaic

Figure 5-8
Investigation of Collector Area and Storage Amount
Both Thermal and Electric Storage

only systems. The condenser heat exchanger (cooling system) is therefore retained, without changes. Thus the costs eliminated, referring to Table 5-4, are:

$$\$1,352,435 + \$5,586,500 + \$4,301,355 \left(\frac{38}{104} \right) = \$8,510,000$$

The photovoltaic energy delivered must be corrected to reflect the lower temperature achievable if the thermodynamic bottoming cycle is eliminated. The average efficiency of the solar cells for the present plant is 15.9% corresponding to an average cell temperature of 185°C. When the cells are at this temperature the cooling fluid is at about 145°C, i.e. there is a 40°C temperature differential in the boiler. The initial temperature differential in the condenser heat exchanger is 28°C. We assume that the average daytime ambient temperature of Phoenix, AZ is 21°C. Thus the average solar cell temperature is 21°C + 28 + 40°C = 89°C. The solar cell efficiency corresponding to this temperature, from Equation (3.3-9), is 21%. Photovoltaic energy delivered is therefore:

$$.1685 \times 10^9 \times \frac{0.210}{0.159} = 0.2225 \times 10^9 \text{ kW-hr}$$

Assuming the rating of the power conditioner increases proportionally, the additional cost of the power conditioner is $\$6,261,000 \times \left(\frac{0.210}{0.159} - 1.0 \right) = \$2,008,000$. Therefore, the levelized bus bar energy cost for the photovoltaic-only power plant exclusive of O & M is given by

$$\begin{aligned} \text{BBEC} &= \frac{(\$79,780,000 - \$8,510,000 + \$2,008,000) (1.5) (0.15) (1000)}{0.2225 \times 10^9 \text{ kW-hr}} \\ &= \frac{0.165 \times 10^{11}}{0.2225 \times 10^9} = 74.1 \frac{\text{mills}}{\text{kW-hr}} \end{aligned}$$

Adding 3 mills/kW-hr for O & M we have

$$\text{BBEC} = 77.1 \frac{\text{mills}}{\text{kW-hr}}$$

The equivalent cost for the combined photovoltaic/thermal electric power plant was 70.5 mills/kW-hr. The difference is 6.6 mills/kW-hr or 9.4%.

This rough comparison shows the combined power plant to be more economical than the photovoltaic-only, although the difference is not overwhelming. More detailed analyses of the photovoltaic-only power plant are required to clarify this comparison. If the cost differential remains this close, the final choice between these two power plants may be made on non-economic grounds.

NOMENCLATURE

- a = solar cell efficiency at 0°C
A = area
b = temperature coefficient of solar cell efficiency
C = concentration ratio (ratio of mirror area to receiver area)
C = cost in 1978 dollars
 C_c = conductance of solar cells
 C_f = conductance across fouled surface
 C_p = liquid specific heat
 C_w = conductance of wall
- E = total energy generated during year
 E_a = turbine annulus area
- f = Fanning Friction factor
 $F = (R_e / R_{e_l})^{0.8}$
 F_s = solar flux incident of receiver from one mirror
FCR = levelized fixed charge rate
- g = local gravitational field
 g_c = gravitational constant
G = mass velocity
- h = heat transfer coefficient
 h_b = enthalpy of saturated steam leaving boiler
 h_c = enthalpy of stored cold water
 h_h = enthalpy of stored hot water
 h_t = enthalpy of flashed vapor leaving storage
- IF = factor to account for indirect costs
- k = thermal conductivity
 K_1, K_2 = coefficients in turbine-generator cost relationship
- l = generator losses
 l_m^g = mechanical losses in turbine-generator system
 L_B = height of boiler section
 L_p = height of liquid preheater section

O & M = levelized operations and maintenance cost

P = pressure

PCF = plant capacity factor, i. e. energy generated during the year divided by the product of plant rated power and the hours in the year

P_p = photovoltaic power conditioner rating

P_t = generator power rating

ΔP = change in pressure, pressure drop

ΔP_t = change in vapor pressure corresponding to $T_w - T_s$

Pr = Prandtl Number

Q = rate of flow of thermal energy

r_h = hydraulic radius

R = ratio of cold water to steam mass flow rate to condense steam for storage

R_s = ratio of cold water to steam mass flow rate during energy extraction from storage

R_v = fraction of annulus volume filled by vapor

Re = Reynolds number

S = suppression factor

S_E = electric energy storage capacity

t = annulus thickness

T = temperature

U = overall heat transfer coefficient

W = mass flow rate of fluid through boiler

W_t = mass flow rate of fluid from storage

x = vapor fraction, quality

X = Martinelli parameter

α = absorptivity of solar cells

η = efficiency as a fraction

λ = heat of vaporization

μ = viscosity

ρ = density

σ = surface tension

ϕ^2 = Lockhart-Martinelli multiplier

Superscripts

B = boiler

P = preheater

Subscripts

c = cold, collector

f = fluid friction

h = hydrostatic, hot

I = ambient

l = liquid, liquid flowing alone in tube or annulus

macro = macro convective contribution to heat transfer

micro = micro convective contribution to heat transfer

PF = packing fraction

S = saturated

SC = solar cell

t = bare tube

tt = both liquid and vapor turbulent

v = vapor

w = wall

REFERENCES

1. L. W. James and R. L. Moon, "GaAs Concentrator Solar Cells", 11th IEEE Photovoltaic Specialists Conf. Rec. Scottsdale, Arizona. IEEE, No. 75CH0948-OED, May 1975.
2. S. L. Leonard, E. J. Rattin, and B. Siegel, Mission Analysis of Photovoltaic Solar Energy Conversion, Aerospace Report No. ATR-77(7574-07)-1, The Aerospace Corp., El Segundo, California, March 1977.
3. Energy and Transportation Division, Mission Analysis of Photovoltaic Solar Energy Systems: Final Report, Report No. ATR-75(7476-01)-2, Aerospace Corp., El Segundo, California, April 30, 1975.
4. E. A. De Meo, and P. B. Bos, "Perspectives on Utility Central Station Photovoltaic Application", Solar Energy, Vol. 21, No. 3, 1978.
5. Personal Communication; Varian Associates Inc., Palo Alto, Ca.
6. Rohsenow, Warren M., "Boiling" Section 13, Handbook of Heat Transfer, Ed. W. M. Rohsenow and J. P. Hartnett, McGraw Hill, 1973.
7. Colin F. McDonald and Kosla Vepa, Ammonia Turbomachinery Design Considerations for the Direct Cycle Nuclear Gas Turbine Waste Heat Power Plant, General Atomics Report GA-A14171, January 1977.
8. Hans H. V. Cleve, Wilford J. Westre, and Joe Y. Parce, "Economics and Operating Experience with Air-Cooled Condensers," presented at American Power Conference, Chicago, Ill., April 1971.
9. P. J. Christopher and V. T. Forster, "Rugeley Dry Cooling Tower System", Proc. Instn. Mech. Engrs., Vol. 184, Pt 1, No. 11, pp. 197-221, 1969-70.
10. Paul Leung and R. E. Moore, "Thermal Cycle Arrangements for Power Plants Employing Dry Cooling Towers, ASME Series A, Journal of Engineering for Power, Vol. 93, No. 2, pp. 257-264, April 1971.
11. Michael Otis Surface, "System Designs for Dry Cooling Towers", Power Engineering, Sept. 1977.
12. H. J. Janzon, and G. W. Underwood, "Design Application and Operating Experience with Cooling Towers on the American Electric Power System", ASME Preprint, No. 72-Pwr-3, March 1, 1972.
13. Paul Leung, G. R. Reti, and J. R. Schilling, "Dry Cooling Tower Plant Thermodynamics and Economic Optimization", ASME preprint No. 72-Pwr-5, May 10, 1972.
14. Ian Wigham, "Designing Optimum Cooling Systems", Chemical Engineering, Vol. 78, No. 18, pp. 95-102, August 9, 1971.

15. F. J. Schulenberg, "Direct Steam Condensation in the Utility Industry Using Air-Cooled Heat Exchangers", Workshop on Dry Cooling Systems, Franklin Institute Research Laboratories Report No. FIRL/F4202, Jan. 1976.
16. R. W. Lockhart and R. C. Martinelli, "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes", Chem. Eng. Prog., Vol. 45, No. 1, 1949.
17. T. B. Drew, H. H. Dunkle, and R. P. Genereaux, "Flow of Fluids", Section 5 of Chemical Engineers Handbook, ed. John H. Perry, McGraw Hill, Third Edition, 1950.
18. Letter from D. R. Wright, General Electric Inc., Industrial Turbine Sales, El Monte, California 91734 to O. J. Mead, Aerospace Corp., March 28, 1978.
19. Colorado State University and Westinghouse Electric Corporation Solar Thermal Electric Power Systems, Appendices of Vol. 3 of Final Report, Report No. NSF/RANN/SE/GI-37815/FR74/3, Nov. 1974.
20. R. C. Spencer, K. C. Cotton, and C. N. Cannon, A Method for Predicting the Performance of Steam Turbine-Generators ... 16,500 kW and Larger, General Electric Report-2007C, July 1974.
21. Zygmunt M. Skusarek, The Economic Feasibility of the Steam-Ammonia Power Cycle, Report PB 184-331, The Franklin Institute Research Laboratories, undated.
22. D. E. Briggs, and E. H. Young, "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Bank of Finned Tubes", Chem. Engr. Progr. Symp. Series, Vol. 59, No. 41, pp. 1-10, 1965.
23. K. K. Robinson and D. E. Briggs, "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes", Chem. Eng. Prog. Symp. Series, Vol. 62, No. 64, pp. 177-184, 1965.
24. J. C. Chato, J. Am. Soc. Heating Refrig. Aircond. Engrs., p. 52, Feb. 1962.
25. T. B. Drew, H. H. Dunkle, and R. P. Genereaux, "Flow of Fluids, Section f, pp 379-381, Chemical Engineer's Handbook, ed. John Perry, McGraw Hill, 1950.
26. Martin Marietta Corporation, Denver, Co., Cost Analysis, 10 MW_e Central Receiver Solar Thermal Pilot Plant, ERDA Contract E (04-3)-1110 March 8, 1976.

27. U. S. Atomic Energy Commission, Report No. WASH-1230, Pressurized Water Reactor Plant, Vol. 1, June 1972, (Study performed by United Engineers and Constructors, Phila. Penn.
28. Letter and data package from W. B. McKinney Sales Manager, General Atomic Inc., San Diego, Calif. 92138 to O. J. Mead, Aerospace Corp., March 22, 1978.
29. K. M. Guthrie, "Capital Cost Estimating", from Modern Cost Estimating Techniques, Ed. Herbert Popper, McGraw Hill Book Company, 1970.