COMBINED THERMAL STORAGE POND AND DRY COOLING TOWER
WASTE HEAT REJECTION SYSTEM FOR SOLAR-THERMAL
STEAM-ELECTRIC POWER PLANTS

Final Report

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ABSTRACT

The thermal performance and economics of the combined thermal storage pond and dry cooling tower waste heat rejection system concept for solar-thermal steam-electric plants have been evaluated. Based on the computer simulation of the operation of southwest-sited solar-thermal plants, it has been determined that the combined pond-tower concept has significant cost and performance advantages over conventional dry cooling systems.

Use of a thermal storage pond as a component of the dry cooling system allows a significant reduction in the required dry cooling heat exchange capacity and the associated parasitic power consumption. Importantly, it has been concluded that the combined pond-tower dry cooling system concept can be employed to economically maintain steam condensing temperatures at levels normally achieved with conventional evaporative cooling systems. An evaluation of alternative thermal storage pond design concepts has revealed that a stratified vertical-flow "cut-and-fill" reservoir with conventional membrane lining and covering would yield the best overall system performance at the least cost.

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SUMMARY AND CONCLUSIONS

This report presents the results of an engineering and economic evaluation of the combined thermal storage pond and dry cooling tower heat rejection system concept as applied to a solar-thermal steam-electric power plant. The combined pond-tower concept has been evaluated as a possible alternative to conventional dry cooling at such plants because it offers the potential for significant cost reduction without the consumptive use of water. The combined pond-tower dry cooling system concept involves the incorporation of a sealed water impoundment into the water flow circuit of a conventional dry cooling heat exchange loop using a surface steam condenser and finned-tube type of water-to-air heat exchanger. Use of the thermal storage pond in the dry cooling circuit allows the heat rejection load on the dry cooling tower to be spread over the entire 24 hour day and allows the utilization of the typically much cooler night air for cooling. The scope of the engineering and economic evaluation has included the preliminary evaluation of thermal performance of alternative system design and operation concepts, the assessment of the design requirements for the thermal storage pond, and the evaluation of the economics of the combined pond-tower concept in comparison to conventional alternatives.

The initial assessment of the thermal behavior of the pond-tower system revealed that a thermal storage pond coupled to a low capacity dry cooling tower will significantly increase the cooling performance of the heat rejection system. Also, it has been shown that a thermal storage pond coupled to a high performance dry cooling system either significantly reduces the amount of required dry cooling heat exchange capacity or will result in levels of performance unachievable using simple dry cooling. Additionally, the initial assessment has shown that two approaches to the operation of the combined pond-tower system are possible. One approach is to utilize the pond to maximize the electrical capacity of the plant during periods of peak electrical demand while the other approach is to utilize the pond to maximize the total daily electrical power production of the plant. Also, it has been found that an important

benefit of the thermal storage pond concept is the feasibility of turning the dry tower fans off during peak electrical demand periods with no increase in condensing temperature.

The assessment of alternative thermal storage pond design concepts has considered both thermal performance and economic factors. The general result is that a stratified, vertical-flow storage reservoir appears to offer the best potential for lowest cost and maximum thermal performance. The difficulties associated with inhibiting density-induced flow in a horizontal-flow type of pond limits the thermal performance potential of this alternative. Also, the construction cost per unit of stored volume for the horizontal-flow pond is estimated to be higher than that of a stratified pond. Correct design of the flow inlet and outlet devices at the top and bottom of the vertical-flow pond should result in a good approximation to ideal stratified flow thermal behavior. It has also been determined that conventional techniques for the construction of "cut-and-fill" water storage reservoirs can be utilized for the construction of a thermal storage pond. Currently available membrane lining and covering materials may be applied to minimize seepage and evaporative water losses.

The final task in the concept assessment has been the evaluation of the system cost. The combined dry cooling tower and thermal storage pond concept for waste heat rejection from solar-thermal steam-electric plants has been found to have a significant cost advantage in comparison to conventional dry cooling. This result has been obtained by examining the detailed thermal performance and cost of cooling systems for a nominal 100 MWe second-generation type of solar-thermal plant with 6 hours of thermal energy storage capacity sited in arid Southwest locations. In each case the cost of heat rejection is computed as a present-worth sum of capital investment, system operating and loss of capacity and energy production penalty costs and is presented as an incremental cost of power production due to heat rejection requirements. All costs related to thermal performance of the cooling system are developed through the simulation of the system thermal behavior for a one year operating period which is

represented by a correlated set of meteorological and steam condensing load data.

Importantly, the economic analysis has shown that the pond-tower cooling concept is cost effective in maintaining the plant conversion efficiency (i.e., condensing temperature) at levels similar to that achieved with conventional evaporative cooling systems. With the pond-tower system it is not necessary to compromise the plant efficiency to reduce, to acceptable levels, the investment in dry cooling heat exchange capacity. This result holds even for a site at which the maximum ambient temperature is 112°F. The total cost of heat rejection with the combined pond-tower system has been determined to be between about 5 to 7 mills/KWhr depending on the site and amount of thermal energy storage in the solar plant. The cost of simple dry cooling systems of equivalent cooling performance is in the range of 10 to 16 mills/KWhr, again depending on the site and plant. Thus, the potential cost reduction is large, especially if one considers that the cost of conventional evaporative cooling with low-cost water would be about 3 mills/KWhr.

For the nominal 100 MWe solar plant considered in the economic analyses the optimum size thermal storage pond was found to be in the range of 4 to 8 acres with an average depth of 25 feet. The optimum size of the dry cooling towers coupled with these ponds were determined to be only 20 to 50% of the size required for conventional dry cooling.

Section 1

THE COMBINED DRY COOLING TOWER AND THERMAL STORAGE POND WASTE HEAT REJECTION CONCEPT FOR SOLAR-THERMAL ELECTRIC POWER STATIONS

1.1 Requirements for Dry Cooling of Solar-Thermal Steam-Electric Plants

Currently there is a national effort to develop economical and reliable means of producing electric power using the sun's energy. One concept under active consideration involves the optical collection of sunlight for absorption and conversion to thermal energy at a central receiver. Electric power is then generated by using the thermal energy to generate steam for powering a steam turbine-generator. After expanding through the turbine, the steam is condensed and returned to the steam generator. The heat extracted from the steam in the condensing process is normally rejected to the environment as waste heat. This solar steam Rankine cycle is expected to operate at steam temperatures and pressures similar to those found in conventional fossil-fuel steam-electric plants.

The projected solar-thermal steam-electric industry will encounter special problems with respect to waste heat rejection. The region of the United States currently considered to be most favorable for the introduction of a large scale solar-thermal industry is the Southwest because of the low incidence of cloud cover. Unfortunately, this is precisely the region of the country facing the greatest shortage of water (1,4). Because of the copious amounts of water utilized in conventional steam-electric plant heat rejection systems, this arid region is the least favorable location for siting steam-electric power plants. The extent of the availability of water for possible use in power generation in this area of the country has recently been evaluated by several authors (2,3,4). In general, these studies point to the future need for "dry" cooling of conventional nuclear and fossil steam-electric plants in the Southwest.

The general shortage of water for power production in the Southwest is exacerbated for solar-thermal power plants because of the special siting constraints for such plants. Large land areas will be required for solar-thermal collector fields, thereby making it difficult to site the plant near the limited water supplies which do exist in the Southwest. In view of these circumstances it is reasonable to assume that, in the Southwest, heat rejection from solar-thermal steam-electric plants will, in most cases, have to be achieved without the consumptive use of water. This requirement for the use of "dry" cooling at solar-thermal steam-electric plants may increase the steam condensing temperature of the plant and thus have an adverse impact on both the design of the steam turbine and on the cycle efficiency.

Traditionally, steam-electric plant condensing temperatures of about 95°F have been achieved by using natural bodies of water for a source of condenser cooling water. The cooling water is heated in the condenser and simply returned to its source where natural cooling processes act to ultimately transfer the waste heat to the atmosphere. More recently, closed-cycle water cooling loops have been employed. In such systems, heat rejection to the environment is achieved predominately through the evaporation of the heated circulating water. Condensing temperatures experienced with these systems range up to about 130°F. The temperature of the water supplied to the condenser by such evaporative cooling systems varies with the ambient wet bulb temperature. The plant generation capacity reductions which may be experienced with closed-cycle evaporative systems can be as much as 5% of the nominal plant capacity.

The use of dry cooling for heat rejection at steam-electric plants can result in a highly variable steam condensing temperature and in steam condensing temperatures in excess of 160°F. As a result, power production capability reductions of more than 10% may occur. For a mechanical draft type of dry cooling tower, the difference between the steam condensing

temperature and ambient dry bulb temperature is approximately constant for a constant steam condensing load.

1.2 Current Status of Steam-Electric Plant Dry Cooling

All closed-cycle steam-electric plant heat rejection systems function to transfer the latent heat of vaporization of the condensing steam to the atmosphere. Dry cooling heat exchange concepts involve only the sensible heating of the ambient air.

To date, two basic types of dry cooling systems have been employed at conventional steam-electric plants. These are referred to as "direct" condensing systems and "indirect" condensing systems (5). In the direct system, the turbine exhaust steam is ducted to a finned-tube heat exchanger. The steam condenses isothermally in the tubes while air is passed across the outside finned surfaces of the tubes. In the "indirect" system, the turbine exhaust steam is condensed in either a surface or direct contact condenser. The cooling water which circulates through the condenser experiences a temperature rise while absorbing the latent heat of the condensing steam. The circulating water is then passed through an extended surface air cooled heat exchanger where it is recooled. The indirect system is considered to be more attractive for large plant sizes. The basic disadvantage of the indirect system is that the two stage heat rejection process (i.e., steam→circulating water→air) results in an increased temperature difference between the condensing steam and the ambient air.

The evaluated costs of both these types of systems are several times that of evaporative cooling and thus neither has been employed except in special circumstances. Currently there is a major effort directed toward the research, development, and demonstration of advanced and economic dry cooling technologies for steam-electric plants (6). The efforts take the form of evolutionary improvements in the basic concepts outlined above and the development of an ammonia intermediate heat exchange system. The

evolutionary improvements being considered involve examination of new lower-cost heat exchange surfaces and the combination of dry cooling with limited amounts of evaporative cooling. The ammonia system concept involves the use of a steam condenser/ammonia boiler at the turbine exhaust and the condensing of the ammonia vapor in an extended surface air-cooled heat exchanger.

The steam turbines used at nuclear and fossil-fueled steam-electric plants with closed-cycle evaporative cooling are essentially identical to those used with "once-through" cooling. Such turbines are usually limited to operation at condensing pressures of 5 in. Hg or less. For dry cooling systems, alternative steam turbine designs which allow operation at higher condensing pressures have been employed in view of the much higher condensing pressures which may be experienced (7).

1.3 Dry Cooling at Solar-Thermal Steam-Electric Plants

The direct application of existing dry cooling technology at solar-thermal steam-electric plants is undesirable from both economic and plant performance standpoints. First, a solar-thermal plant with several hours of high-temperature thermal storage will not operate during most of the night. Thus, the highly capital intensive dry cooling facility would not be efficiently utilized. Second, the steam condensing requirement of the solar-thermal plant will occur during the day while ambient temperatures are near the daily maximum. The condensing temperature resulting from the use of conventional dry cooling on summer afternoons would undoubtedly be undesirably high and would result in significant reductions in generation capability. The specific maximum condensing temperatures which would occur would depend on the size and performance characteristics of the dry cooling tower. In brief, since dry cooling is not currently economical for conventional steam-electric plants and since the low capacity factor of solar-thermal steam-electric plants will further weigh against the use of current dry cooling concepts, it becomes essential to consider novel dry cooling concepts which have the potential for significant cost reductions and performance improvements.

1.4 The Combined Dry Cooling Tower and Thermal Storage Pond Waste Heat Rejection Concept

1.4.1 The Concept and Its Potential Benefits

The combined dry cooling tower and thermal storage pond concept for waste heat rejection at solar-thermal plants has the potential for maintaining the cost of heat rejection and the power production capability of the solar-thermal steam-electric plant at acceptable levels while avoiding the consumptive use of water. A schematic drawing of the combined system concept is shown in Figure 1.1. Essentially, this concept involves the coupling of a conventional indirect dry cooling system with a volume of stored water. The purpose of integrating the stored water with the condenser cooling flow circuit is to add thermal capacitance or thermal storage to the dry cooling system. A recent study of this combined system concept as it applies to conventional steam-electric plants has shown the concept to be an attractive alternative to conventional dry cooling concepts (8). The benefits of an appropriately sized storage "pond" for a solar-thermal steam-electric plant would be two-fold. These are:

1. Heat rejection can proceed continuously over the 24-hour day and not only when there is an actual need for steam condensing.
2. Advantage can be taken of the normally relatively cool nighttime air temperature.

Figure 1.1
Schematic of Combined Dry Cooling Tower
And Thermal Storage Pond Concept

The specific method of operating the system which is depicted in Figure 1.1 would depend on the design of the system, the conditions under which the system operates, and the requirements for heat rejection. In all cases, however, the basic modes of operation would be as follows:

Daytime	1. <u>Discharge Mode</u> - Relatively cool water is withdrawn from the pond, heated in the condenser, partially cooled in the dry tower, and returned to the pond at an increased temperature.
Nighttime	2. <u>Regeneration Mode</u> - Relatively warm water is withdrawn from the pond, passed through the dry tower, and returned to the pond at a decreased temperature.

The discharge mode would function essentially during the daylight hours while there is an actual steam condensing load. The regeneration mode would function essentially at night.

The extent to which the combined benefits would be accrued would depend upon the design and operation of the system. An estimate of the potential magnitude of these benefits can be made by examining the information presented in Table 1.1 and Figure 1.2. Table 1.1 presents data on the increase in the duration of the period of heat rejection if the heat rejection process is extended to the full 24-hour day through the use of a waste heat thermal storage pond. Assuming that the total quantity of heat to be rejected for a given 24-hour day is

$$Q = (UA\Delta T) \cdot t \quad (1)$$

where UA = performance or size factor for tower,
 ΔT = working temperature difference in dry tower
 (condensing temperature - ambient temperature), and
 t = period of heat rejection,

Table 1.1
REPRESENTATIVE AVERAGE DURATION OF DAILY
OPERATION OF SOLAR-THERMAL PLANT

<u>Month</u>	<u>Average Duration of Daily Period of Plant Operation</u> *	<u>Average Duration of Night-time Shutdown = Additional Available Period of Heat Rejection</u>	<u>Percentage Increase In Period of Heat Rejection with Combined System</u>
Jan.	10.1 hrs.	13.9 hrs.	137%
Feb.	11.4	12.6	110%
Mar.	12.1	11.9	98%
Apr.	13.3	10.7	80%
May	14.2	9.8	69%
June	14.6	9.4	64%
July	14.5	9.5	65%
Aug.	13.5	10.5	77%
Sep.	12.7	11.3	88%
Oct.	11.3	12.7	112%
Nov.	10.6	13.4	126%
Dec.	9.6	14.4	150%

*Assumptions - Site = Albuquerque, N.M.

Operation From Storage = 2 hr.

Operation From Receiver = Period of Daylight - 2 hrs.

Figure 1.2

Local Diurnal Ambient Temperature Extremes

then an increase in "t" allows a corresponding decrease in either the size of the tower or a decrease in the operating temperature difference. Assuming a 64% increase in the duration of the period of heat rejection during June, as given in Table 1.1, there is the potential for approximately a 40% reduction in the tower size for a given cooling duty.

The benefits of rejecting heat during the night can be estimated for various regions by examining the average daily temperature variation information presented in Figure 1.2. Note that diurnal temperature variations of 30°F or more are common in the Southwest, especially in the summer months. Again, in reference to Equation (1), a decrease in the ambient air temperature allows a lower steam condensing temperature or a smaller dry cooling tower size.

1.4.2 Scope of Combined System Concept Evaluation Project

Dynatech has undertaken an engineering and economic assessment of the combined dry cooling tower and thermal-storage pond concept as it applies to solar-thermal steam-electric plants. The objective of this work has been to determine the general system design and operation schemes which will result in the maximum benefit of waste-heat thermal storage pond concept and which will result in the minimum cost of heat rejection.

The efforts on this project have centered on three main tasks. These tasks are (1) the performance evaluation of alternative system design and operation options, (2) the evaluation of the technical difficulties which may be encountered in the implementation of the concept, and (3) the preliminary evaluation of the combined system economics through the computer simulation of the operation of a solar-thermal plant for one year periods on a case-study basis. The performance assessment involves an examination of the relative importance and impact of the various system design and operation parameters. The evaluation of the engineering practicality of the waste heat thermal storage pond has been concerned with

developing storage pond design concepts which are economical and which will result in the desired thermal-hydraulic behavior. The results of the simulation studies provide a basis for assessing the costs and levels of performance which can be achieved by solar-thermal steam-electric plants using air cooling.

Section 2

GENERAL CONSIDERATIONS IN EVALUATING WASTE HEAT REJECTION SYSTEMS FOR SOLAR-THERMAL ELECTRIC PLANTS

2.1 Introduction

There are a number of factors to consider in the evaluation of alternative waste heat rejection system design and operation options. These considerations can be generally grouped in three (3) levels. They are:

1. the electrical utility system level,
2. the steam plant level, and
3. the waste heat rejection system level.

This section briefly reviews the important considerations at each of these levels which are relevant to the evaluation of the combined dry cooling tower and thermal storage pond waste heat rejection system for solar-thermal steam-electric plants.

2.2 Design and Operating Considerations at the Electrical Utility System Level

The performance of the waste heat rejection system of the solar-thermal plant will have an impact at the overall utility system operation level. The inferior thermal performance of a dry cooling heat rejection system significantly influences both the electrical generation capacity (kw) of the plant and the electrical energy production (kwhrs) of the plant. The impact of the waste heat system on the generation capacity of the plant is important since the utility system must provide sufficient generating capacity to meet the projected load with a high degree of reliability. In general, those waste heat rejection system designs which lead to greater

losses of generating capacity incur in proportionately greater economic penalties because of the additional "backup" capacity required in the utilities generation network.

Similarly, the loss of electrical energy production due to the less-than-ideal cooling capability of a dry cooling system will result in an economic penalty which should be considered in the economic design of the waste heat system. Since, the fuel cost of a solar-thermal plant is essentially zero, the solar plant would have a high priority in the economic dispatch of the available generating plants. The energy production losses resulting from less-than-ideal cooling of the solar-thermal unit and the costs associated with replacing this lost power using other generating units in the utility system should be considered in the waste heat rejection system economic design process.

In summary, the economic penalties at the utility system level resulting from the generally inferior cooling performance of a dry cooling systems must be considered in the economic design of the waste heat rejection facility. A general formulation of the heat rejection system economic design problem which incorporates the capacity and energy production loss penalties is:

$$C = C_c + C_o + \Lambda_c + \Lambda_o$$

where

C = total cost of the heat rejection system

C_c = capital costs of the dry cooling system

C_o = operating cost of the dry cooling system

Λ_c = loss of capacity penalty

Λ_o = loss of energy production penalty

The goal of the system design process is thus the determination of the system design which results in the minimum value of C, the total system cost.

2.3 Design and Operating Considerations at Plant Level

The operating condensing temperature of a steam-electric plant directly affects the electrical power output of the turbine-generator. Also, during the plant design process, the expected operating steam condensing temperature can affect the design and selection of the low-pressure stages of the turbine.

The inlet steam conditions for a solar-thermal plant will be similar to those of a fossil-fired steam-electric plant. In Figure 2.1 the heat rates of three fossil-steam turbines with different low-pressure stage designs are presented as a function of condensing pressure [7]. The important observations with respect to these curves are that (1) the turbines which are more efficient at the normal operating back pressures of 1.5" to 3" Hg are either limited to low back pressure operation (less than 5" Hg (133°F)) or are very inefficient at the extreme pressures which may be encountered with dry cooling and (2) the turbine design which is more efficient at the high condenser pressures is also much less efficient in the normal operating condenser pressure range.

Figure 2.1

Normalized Heat Rate Versus Turbine Back Pressure -
Fossil Steam Conditions (Ref. 7)

2.4 Design and Operating Considerations at the Waste Heat Rejection System Level

From the viewpoint of the waste heat rejection system design engineer, the two design parameters which are most important in the specification of an economic design system are the size of the heat rejection facility and the circulating water flow rate. Normally, the system design engineer would be involved in evaluating the application of commercially available heat exchange equipment and thus would not be concerned with the details of equipment design.

For a conventional modular, mechanical-draft type of dry cooling tower, the key design parameters would be the number of tower modules and the circulating water flow rate. For the combined dry cooling tower and thermal storage pond system the key design parameters would be the number of tower modules, the size of the pond, and the circulating water flow rate.

Knowledge of the specific conditions under which the heat rejection system will be operated is, of course, crucial to the correct design of the heat rejection system. For the combined tower and pond system, the meteorological parameters which are of major importance are the daily maximum ambient air dry bulb temperature and the diurnal range of the ambient air temperature. Since the pond water surface would be covered to suppress evaporation, other meteorological parameters such as humidity and insolation should not be of major importance.

The procedures for operating the solar plant and the combined tower and pond system will also have a significant impact on the performance of the system. Key operational parameters include the duration of the period of steam condensing and the rate of flow through the pond during both the discharge and regeneration modes of operation. Additionally, the performance of the combined system will be affected by the thermal response characteristics of the storage pond.

Section 3

THERMAL PERFORMANCE OF COMBINED POND AND TOWER DRY COOLING SYSTEMS

This section summarizes the results of an assessment of the potential thermal performance of the combined dry cooling tower and thermal storage pond waste heat rejection system concept as applied to a solar-thermal steam-electric power plant. Initially, a reference power plant and dry cooling tower system design is described. Subsequently, the general performance characteristics of a solar-thermal steam-electric plant employing the combined heat rejection system concept is assessed in relation to the performance of an identical plant employing conventional dry cooling. Finally, the significance and consequences of alternative combined system operating procedures and conditions are evaluated.

The objective of the analysis summarized in this section is the determination of:

1. the general performance characteristics of solar-thermal steam-electric power plants cooled using a combined dry cooling tower and thermal storage pond system,
2. the thermal performance advantage of the combined system over conventional dry cooling, and
3. the importance of the various system design and operational variables on the performance of the combined system concept.

This section does not address the question of the economics of the pond-tower concept. This is considered in Section 5.

3.1 The Power Plant and Heat Rejection System Model

An assessment of the thermal performance of the combined heat rejection system concept as applied to solar-thermal steam-electric plants has been achieved by the formulation and application of a computer model of the system. For each of the major system components (i.e., the turbine-generator, the steam condenser, the dry cooling tower, and the thermal storage pond), representative thermal performance models have been employed.

The heat rejection system modelled is a conventional surface steam-condensing type. As depicted in Figure 3.1, water would be circulated from the storage pond through a conventional surface condenser in order to condense the steam exhausted from the steam turbine. After being heated in the condenser the circulating water is passed through a conventional dry cooling tower and then passed into the pond. During the night when there is no steam condensing requirement, the water in the pond is circulated directly through the tower for cooling.

All the performance analyses presented in the following sections are based on the system component arrangement as shown in Figure 3.1. A possible alternative would be to have the condenser discharge directly into the pond and the pond subsequently discharge into the dry tower. This alternative, however, would lead to poorer plant performance and necessitate the storage of warmer water in the pond. Thus, the performance analysis of this alternative component arrangement has not been pursued.

3.1.1 The Solar Plant and Turbine-Generator Model

The central-receiver solar-thermal power plant concept as currently envisioned would employ a steam-turbine similar to that used at conventional fossil-fired steam-electric plants.

Figure 3.1

THE COMBINED DRY COOLING TOWER AND THERMAL STORAGE POND CONCEPT

In the evaluation of the thermal performance of various heat rejection systems, the key performance characteristic of the turbine generator is the variation of the turbine-generator heat rate (BTU/Kwhr) as a function of the condensing temperature. The relationship used in the computer model to express this characteristic has been deduced from the turbine performance curves given in Reference 1. To account for the effect of operating with lower-temperature steam from storage, the base heat rate of the turbine generator was increased from 8226 to 10109 BTU/Kwhr for those periods of operation with the lower-pressure steam.* In the analyses performed, no limitations have been placed on the operating condensing pressure, although it is recognized that most conventional steam-turbines are limited to operation with a condensing pressure of 5 in. Hg.

While operating with the higher pressure steam from the central receiver, the maximum output of the turbine generator is 100 MWe while operating with steam from storage the maximum power output is 70 MWe. Except where specified, the performance analyses have been conducted assuming steam flows to the turbine which are expected on or about June 21, the day of maximum plant operation. The maximum output of the potential solar-thermal plant on this day is 100 MWe between 8 a.m. and 8 p.m. The thermal storage is assumed to supplement the plant operation from 8 p.m. to 11 p.m. with a maximum output of 70 MWe.

*The thermal performance presented in this section is based on a Barstow type of plant design. The economic evaluations presented in Section 5 are based on a second-generation type of plant design with no degradation of the steam quality when operating from storage.

3.1.2 The Condenser Model

All the condensers modelled in the performance assessment of the combined system heat rejection concept are designed for a terminal temperature difference of 5°F at full thermal load. The terminal temperature difference (TTD) is the difference between the steam condensing temperature and the cooling water outlet temperature. This design approach is representative of current design practice for large surface condensers for steam-electric plants.

3.1.3 The Dry Cooling Tower Model

The dry cooling tower modelled in the performance assessment is assumed to be composed of a large bank of finned-tubes. Air is circulated over the finned-tubes using large diameter axial-flow fans. However, no specific tower geometry has been defined. Rather, in each case, the dry tower is specified by an overall heat transfer coefficient (UAF). The UAF is specified independent of the total water flow through the tower while the air flow through the dry tower is assumed to be proportional to the UAF. The constant of proportionality of $2.93 \frac{\text{lbm} \cdot ^\circ\text{F}}{\text{BTU}}$ was determined based on review of several dry tower design studies (References 7, 11, and 12).

The finned-tube type of dry cooling tower was chosen for the system analysis since it best represents commercially available dry cooling technology and it is the type of dry cooling system into which the thermal storage pond concept can be most easily integrated. This is because, as currently envisioned, the thermal storage pond would store directly

the water used in the condenser-tower cooling loop. Alternative dry cooling schemes (such as direct steam condensing or those concepts employing a boiling-condensing ammonia heat transfer loop between the steam condenser and dry tower) would not be as compatible with the thermal storage pond concept since an additional heat exchanger would be required.

3.1.4 Thermal Storage Pond Model

A plug-flow is assumed to occur in the thermal storage pond except for the one case where the performance affect of full-mixing is evaluated. The assumed pond depth is 20 feet and it is further assumed that heat transfer through the pond covering is negligible. The assumption of a depth of 20 feet does not affect the pond thermal response. It is used only as a reference to compute a pond surface area.*

3.1.5 Pumping and Fan Power

No accounting is made for water pumping power consumption since use of a thermal storage pond should not have a significant impact on the pumping requirements. However, a representative value of 1 megawatt of fan power per 100 megawatts of heat rejection by a dry cooling tower is assumed to evaluate the benefits of scheduled tower shutdowns.

3.1.6 Site Model

For the purposes of the performance assessment of various dry tower-pond combinations, the site meteorological conditions are adequately described by a daily maximum temperature and an associated daily temperature range. Assuming in all cases that the maximum ambient temperature and minimum ambient temperature occur at 3 p.m. and 6 a.m. respectively, and

*A stratified flow model which accounts for pond surface heat transfer to the environment has been used in the economic analyses presented in Section 5.

that the temperature varies sinusoidally an approximate temperature variation for each condition of interest could be generated. Much of the analysis which is performed is centered on the conditions of 105°F maximum temperature and 30°F daily temperature range. It is believed that this represents typical design or "hottest" day condition which would exist in the plant siting region of prime interest.

3.1.7 Operation Scheme

Unless otherwise specified, the tower-pond systems are operated such that during the period of the solar-plant operation:

1. the condenser discharges, at a constant flow rate, directly to the dry tower which is fully operation,
2. the tower in turn discharges its total flow directly into the thermal storage pond, and
3. the pond supplies the entire condenser cooling water supply..

During the period of no plant operation and pond cooling the pond volume is circulated through the dry tower at a flow rate such that the pond volume passes through the dry tower once.

3.2 Performance of a Solar-Thermal Steam-Electric Plant with Conventional Dry Cooling

3.2.1 Considerations in the Definition of a Reference Conventional System Design

The combined dry cooling tower and thermal storage pond heat rejection concept for solar-thermal steam-electric plants has been presented as a concept which has potential thermal and economic advantages over conventional dry cooling. Thus, in order to make a meaningful assessment of the actual advantage of the concept, it is necessary to define, for the purpose of establishing a reference point, the thermal performance of conventional dry cooling systems.

For the purposes of this study of thermal performance, no attempt has been made to define a reference optimum conventional dry cooling for the model solar-thermal plant.*

The approach taken in establishing a reference for comparison has been to compute the thermal performance of several different "size" dry cooling systems which represent the range of interest. The intent has thus been to evaluate the performance impact of integrating a thermal storage pond into each of these "size" dry cooling systems. This is a legitimate approach since, in general, it has become apparent that the thermal storage pond concept should have significant benefit both for high performance (i.e. close approach to ambient) and low performance (i.e., poor approach to ambient) dry cooling systems. The difference is that

1. the benefit for high performance systems is mainly in the reduction in the amount of dry cooling tower needed to achieve the desired level of performance, and
2. the benefit for low performance systems would be mainly in the improvement in the overall system thermal performance.

*Economics of conventional dry cooling system designs are presented in Section 5.

3.2.2 Reference Conventional Dry Cooling Systems

The thermal performance of three different sized dry cooling towers has been computed for the model solar-thermal plant. These three dry towers are characterized by the following overall heat transfer coefficients (UAF):

Tower Size 1: $UAF = 11.6 \times 10^6 \text{ BTU/hf-}^\circ\text{F}$
water flow = $9.7 \times 10^6 \text{ lbm/hr.}$

Tower Size 2: $UAF = 23.3 \times 10^6 \text{ BTU/hr-}^\circ\text{F}$
water flow = $19.4 \times 10^6 \text{ lbm/hr.}$

Tower Size 3: $UAF = 46.6 \times 10^6 \text{ BTU/hr-}^\circ\text{F}$
water flow = $38.8 \times 10^6 \text{ lbm/hr.}$

The thermal performance of these different size towers for the ambient temperature conditions of a 105°F daily maximum temperature and a 30°F daily temperature range is given in Figure 3.2. The tower size "2" shown above was selected as being representative of optimum dry tower for a 100 MWe baseload steam-electric plant. Tower size "3" is double this size and tower size "1" is one-half this size.

FIGURE 3.2

Comparative Performance of Three Different Simple Dry Cooling Systems

3.3 Performance of Solar-Thermal Plant with Combined Thermal Storage Pond-Dry Cooling Tower System

3.3.1 General System Performance Characteristic

The heat rejection performance of nine (9) different tower-pond combinations coupled to the model solar-thermal plant has been computed. In each case, the performance at a variety of ambient temperature conditions was determined. All analyses are based on the performance of the specific tower-pond combination for a 24-hour period. The results presented in each case represent the expected "steady-state" system behavior which would result after continuous operation through a number of 24-hour periods with the identical ambient temperature variations.

Effect of Various Sized Ponds Coupled to "Best Guess" Sized Dry Tower

The power production benefit of adding three different sized thermal storage ponds to a dry cooling system with a size "2" tower is presented in Figure 3.3. Recall that the size "2" tower represents a "best guess" as to what might represent a optimally-sized conventional dry cooling system for the solar-thermal plant. The various pond sizes are designated as 1 day, 1/2 day, and 1/4 day ponds. This designation refers to the fraction of the total flow volume passing through condenser during the period of steam condensing (15 hours) which the pond will store. Thus, for the size of "2" tower (with a flow rate of 19.4×10^6 lbm/hr), the equivalent pond volumes are:

Full day	-	5.36 acres by 20 feet deep
1/2 day	-	2.68 acres by 20 feet deep
1/4 day	-	1.34 acres by 20 feet deep

FIGURE 3.3

Plant Power Output with Size "2" Tower and Three Different Pond Sizes

In each case, the pond volume is circulated once through the dry cooling tower during the 9-hour night period of pond regeneration. Conclusions which can be drawn from examining the performance results presented in Figure 3.3 are:

1. All pond sizes significantly reduce (about 50%) the power loss due to dry cooling.
2. Only the "full day" size results in improved power output over the entire day.
3. The 1/2 day pond results in the maximum power at the maximum ambient temperature.
4. The 1/4 day pond system performs more poorly than the tower-only system during the last several hours of plant power production.

The specific condensing temperatures achieved by the combined systems in comparison to the towers-only system are presented in Figures 3.4, 3.5, and 3.6 for different maximum ambient temperatures. Examining these figures, it is observed that the relative impact of the pond is similar in each case, only the overall level of the temperatures changes. The increased energy production resulting from the addition of various ponds to the size "2" tower is outlined in Figure 3.7. Note that the results presented in Figure 3.7 indicate that, in terms of daily energy production, there is little incentive for using a "full-day" pond as opposed to a "1/2 day" pond. While use of a "1/2 day" pond results in significant electrical energy production increases over the "1/4 day" pond, there is little additional benefit to doubling the pond size again.

The magnitude of the daily range, of course, also has an important impact on the relative performance of the tower-pond system.

CONDENSING TEMPERATURE - °F

Tower Size - 2

○ - Tower Only

□ - Full Day Pond - 5.36 acre

△ - 1/2 Day Pond - 2.68 acre

◇ - 1/4 Day Pond - 1.34 acre

Max. Ambient Temp. - 65°F

Daily Range - 30°F

FIGURE 3.4

Effect of Thermal Storage Pond on Condensing Temperature
for 65°F Maximum Ambient Temperature

FIGURE 3.5

Effect of Thermal Storage Pond on Condensing Temperature
for 85°F Maximum Ambient Temperature

FIGURE 3.6

Effect of Thermal Storage Pond on Condensing Temperature
for 105°F Maximum Ambient Temperature

PLANT OUTPUT (MEGAWATT - HRS.)

FIGURE 3.7

Effect of Thermal Storage Pond on the Daily Total
Electrical Energy Production

In Figure 3.8 is presented the impact of the diurnal temperature range on the electrical energy production benefit of the pond for the three different pond sizes. Note that the benefit of the pond is significant in terms of power production when the diurnal range for a 105°F day is as little as 20°F. The expected temperature range in the arid regions of interest is generally in the range of 30°F and occasionally as high as 40°F.

Effect of Various Sized Ponds Coupled with A Smaller and A Larger Dry Tower

The performance of the model solar-thermal plant has been computed for three different ponds added to both the size "1" dry tower and the size "3" dry tower. In general, the result is that the relative benefit in terms of power output increases as the tower size decreases. The effect of coupling of a "full day", "1/2 day", and "1/4 day" pond to towers of size "1" and "3" are revealed in Figures 3.9 through 3.14. Note that since the water flow rate for the size "3" tower is four (4) times larger than that of the size "1" tower, the volume of the "full-day" pond for the size "3" case is four (4) times that of the size "1" case.

The specific electrical output of the plant for the size "1" and size "3" tower systems coupled to the different pond sizes is given in Figures 3.9 and 3.12, respectively. Comparing these power production profiles to those for the "best guess" tower size "2" presented in Figure 3.3, it can be observed that, generally

1. The smallest pond size (i.e., 1/4 day pond) results in a maximum plant capacity which is of only short duration.
2. The "1/2 day" results in the maximum plant capacity at the hottest time of the day.
3. The specific capacity increase attributable to the coupling of the pond with the tower decreases as the tower size increases.

FIGURE 3.8

Effect of Diurnal Temperature Range on the Electrical Energy Production Benefit of the Thermal Storage Pond

FIGURE 3.9

Plant Power Output with Thermal Storage Pond Coupled with Tower Size "1"

FIGURE 3.10

Steam Condensing Temperature for Operation with Dry Tower Size "1"

FIGURE 3.11
Impact of Thermal Storage Pond on Electrical Energy Production with Size "1" Tower

FIGURE 3.12

Plant Power Output with Thermal Storage Pond Coupled with Size "3" Tower

FIGURE 3.13
Steam Condensing Temperature for Operation with Dry Tower Size "3"

FIGURE 3.14

Impact of Thermal Storage Pond on Electrical Energy Production with Size "3" Tower

Figures 3.10 and 3.13 present the condensing temperature achieved with the size "1" and size "3" tower systems. From these figures it is seen that adding the "1/4" day pond to the size "1" tower system results in a dramatic 65-70°F reduction in the condensing temperature between about 9 a.m. and 11 a.m. Also, adding the "full-day" pond to the size "1" tower results in a nearly constant 160°F condensing temperature. Operation with only the size "1" tower results in condensing temperatures which vary between about 185°F and 225°F with average of about 205°F. The most interesting result shown in Figure 3.13 is that the condensing temperatures for the "1/2 day" pond at the time of the maximum temperature of 105°F is actually less than 105°F.

As is observed for the case with the "best guess" tower size, the larger pond sizes all result in greater net electrical energy production than is achieved with the smaller pond size. This is shown in Figures 3.11 and 3.14 for the size "1" tower and size "3" tower respectively. For the size "1" tower, the absolute increase in the daily electrical energy production of the plant is nearly constant over most of the ambient temperature range of interest and at the higher ambient temperature ranges, the output is about 7% higher than for the towers-only case.

3.3.2 The Thermal Storage Pond As A Means of Reducing The Dry Tower Size

The previous sections have shown the performance impact of adding a thermal storage pond to a dry cooling system. In this section an attempt is made to compare the performances of the combined pond-tower systems to that of a simple tower-only system by defining the pond-tower systems in terms of an equivalent tower-only system.

Equivalent Tower Size Based On Maximum Condensing Temperature

One way to define a combined pond-tower system in terms of an equivalent simple tower-only system is to determine the dry tower size which would result in the same maximum condensing temperature as does result from employing the specified combined pond-tower system. This is a particularly appropriate approach since the sizing of a dry tower cooling system can strongly depend on the maximum allowable condensing pressure (temperature).

Presented in Figure 3.15 is the daily maximum condensing pressure for a range of tower-only sizes for the ambient conditions of 105°F maximum and 30°F range. Also presented is the daily maximum condensing pressure experienced with three different sized ponds at each dry tower size. The data points at $UAF = 23.3 \times 10^6$ BTU/hr.°F correspond to the size "2" dry tower. Note that adding the "full-day" pond to a dry tower of this size reduces the maximum condensing pressure from about 9.5 in. Hg. to about 4.5 in. Hg. Additionally, it is important to note that a tower-only system with a UAF of about 56×10^6 BTU/hr. would be required to achieve this same level of performance. Thus, with respect to limiting the maximum condensing to 4.5 in. Hg., the combined system composed of the full-day pond and the size "2" tower ($UAF = 23.3 \times 10^6$ BTU/hr.) has the equivalent UAF of about 56×10^6 BTU/hr. In more simple terms, the benefit of the full-day pond is identical to more than doubling the size of the dry tower.

Examining Figure 3.15 further, it is observed that for combined systems with smaller towers (smaller UAFs) the impact of the pond in terms of equivalent UAF is not as large. For combined systems with a larger tower, the addition of the pond leads to maximum condensing temperatures which are physically unachievable using simple dry cooling. This is because, for the simple dry tower system, the maximum condensing temperature is limited to a close approach to the daily maximum ambient temperature while the combined pond-tower system is limited to a close approach to the daily minimum ambient temperatures.

FIGURE 3.15

Impact of Thermal Storage Pond on Maximum Condensing Pressure

Equivalent Tower Size Based On Daily Energy Production

An alternative method of equating a given pond-tower system to a specific simple towers-only system can be based on net energy production. This method involves the determination of the simple dry tower size which would be required to achieve the same net daily power production achieved by the combined pond-tower system.

This determination of the equivalent tower-only system based on net daily electrical energy production has been performed for the three pond sizes applied to each of the "1", "2", and "3" size dry cooling towers. The results of this computation are summarized in Table 3.1.

The general conclusion which can be drawn from the information presented in Table 3.1 is that the thermal storage pond can be applied to displace a significant amount of dry tower heat exchange capability and cost. The thermal storage pond should be most effective in displacing dry tower cooling capacity (i.e., UAF) when high performance (i.e., low condensing temperatures) is required.

3.3.3 Performance Impact of Alternative Operating Schemes and Conditions

The previous two sections of this report concerned the performance of combined pond-tower heat rejection systems of various size. In this section the performance impact of alternative methods of operating the pond-tower systems are examined. Also, the performance of pond-tower systems at "off-design" steam condensing loads is assessed.

System Water Flow Rate

For the size "2" tower system, the performance impact of an increased and a decreased circulating water flow rate has been computed for the conditions of 105°F maximum ambient temperature and 30°F range. Both a

Table 3.1

EQUIVALENT INCREASE IN TOWER SIZE BASED ON
DAILY TOTAL ELECTRICAL ENERGY PRODUCTION

<u>Effective Increase in UAF of Dry Tower</u>				
	Max. Ambient = <u>65°F</u>	<u>85°F</u>	<u>105°F</u>	(Range = 30°F)
A. Size "1" Tower				
with full-day pond	43%	45%	42%	
with 1/2-day pond	42%	42%	42%	
with 1/4-day pond	28%	29%	29%	
B. Size "2" Tower				
with full-day pond	52%	75%	69%	
with 1/2-day pond	52%	66%	63%	
with 1/4-day pond	30%	42%	44%	
C. Size "3" Tower				
with full-day pond	Infinite - use of pond results in performance levels which are unachievable using simple dry cooling.			
with 1/2-day pond				
with 1/4-day pond				

30% decrease and a 30% increase in the nominal flow of 19.4×10^6 lbm/hr. have been considered. In each case the size of the storage pond is changed in proportion to the water flow rate.

For the dry-tower-only system a 30% decrease in the flow results in a 0.7% decrease in the daily electrical energy production while a 30% increase results in a 0.2% increase in the plant electrical production.

For the "1/2 day pond" and tower system there is a somewhat greater change. The 30% flow increase results in a 0.35% increase in electrical energy production while the 30% decrease in flow results in a 1.1% decrease in the electrical output. Similar changes result for the other pond sizes coupled with the size "2" tower. Thus it appears that the performance of a solar-thermal plant with a pond-tower cooling system is not, within bounds, very sensitive to the variations of the circulating water flow rate. No substantial performance improvements above that outlined in previous sections would be expected as a result of the use of different water flow rates.

Mixing in Thermal Storage Pond

The performance results presented in Section 3.3.2 are based on the assumption that the pond operates in a plug flow manner. Depending on the design of the thermal storage pond, a significant amount of mixing can be expected to occur in the pond. The result of mixing would be to diminish temperature variations in the pond.

The performance of several pond-tower systems has been evaluated for the case of a fully-mixed thermal storage pond. A fully-mixed pond is simply one in which the temperature is uniform. The flow introduced to the fully-mixed pond during each time step is assumed to mix uniformly with the entire pond volume.

The affect of full-mixing is to reduce, in comparison to the plug-flow case, the power generation performance of a plant with a specific size pond-tower cooling system. For most of the cases examined, the electrical energy benefit attributable to the addition of the pond to the dry cooling system was reduced by about 20%. However, at the time of maximum ambient temperature, the fully-mixing was found to reduce the electrical generating capacity benefit of the pond by as much as 50%. Thus, to obtain the maximum benefit from a thermal storage pond of given volume, it appears necessary to avoid excessive mixing in the pond. It is expected, however, that relatively small amounts of mixing would be tolerable.

Multiple Cycling of Pond Volume During Regeneration

The performance results presented in Section 3.3.2 are based on the assumption that the pond volume is passed only once through the dry tower during the night-time regeneration mode of operation. An alternative approach would be to pump the pond water through the tower at a faster rate, thereby increasing the number of times the stored water volume is cycled through the tower.

The relative performance of pond-tower systems which are identical except for the night-time flow rate has been evaluated. The conclusion is that there is little additional benefit for multiple passes through the dry tower for "1/2 day" and "full-day" size ponds. This is especially true when one considers the extra pumping power required. However, for the "1/4-day" pond pumping twice through the dry-tower does appear to result in some performance improvement. In this case the extra pumping power would not be large since the flow rate through the dry-tower would still be less than that during the day.

Dry Cooling Tower Shutdown During Period of Peak Electrical Load

One of the potentially important advantages of the combined pond-tower concept is that it may be possible to shutdown the dry cooling tower fans during the afternoon and early evening period of peak electrical demand. The power consumption of the fans in a dry cooling system can be a significant fraction of the total electrical output of the power plant. The specific amount of power consumed would, of course, depend on the design of the dry tower. However, for most system designs considered to date for conventional steam-electric plants, the power consumption of a dry cooling system is in the range of 1.5 to 3% of the total plant electrical power capacity.

To assess the potential benefit of tower shutdown, an evaluation was made of the operation of several pond-tower systems which employed a 5-hour tower shutdown during the hottest part of the day. The results of this evaluation are summarized in Table 3.2 for various pond sizes coupled with the size "2" dry tower.

As indicated in Table 3.2, the shutdown of the dry tower results in a net increase in the plant generating output at 5 p.m., but results in a net loss in energy production over the entire day. Depending on the system, the increase ranges from 0.7 to 1.7 MWe. The full increase of 2 MWe of fan power is not realized since these results represent the "steady-state" system behavior which would result from continuous operation through a number of days with identical temperature variations. During the period of maximum ambient temperature, the operating tower in a pond-tower system would reject about 40% of the condenser heat load with only the remaining 60% being stored in the pond. Thus, the cooling not performed during the day because of fan shutoff would necessarily be made up during the night with the net effect being that the pond is not recooled to as low a temperature.

Table 3.2

PERFORMANCE OF POND-TOWER SYSTEMS WITH
PEAK LOAD PERIOD DRY TOWER SHUTDOWN

	<u>Full-Day Pond</u>		<u>1/2-Day Pond</u>	
	<u>105°F</u> <u>Max. Ambient</u>	<u>85°F</u>	<u>105°F</u>	<u>85°F</u>
1. Dry Tower Only Power at 5 p.m.	90.8MWe	94.3	90.8	94.3
2. Pond-Tower System Power at 5 p.m. No Shutdown	95.2 MWe	98.9	94.0	97.6
3. Pond-Tower System Power at 5 p.m. With Shutdown	95.9MWe	99.6	95.7	99.3
4. Percentage Change in Peak Power Due to Shutdown	+0.7	+0.7	+1.8	+1.8
5. Percentage Change in Daily Energy Production Due to Shutdown	-0.65	-0.64	-0.43	-0.31

Range = 30°F

UAF = 23.4×10^6 BTU/hr-ft²

Shutdown from 1 p.m. to 6 p.m.

Dry Tower Fan Power = 2MWe

Bypass of Thermal Storage Pond

All of the performance results presented in Section 3.3.2 are based on the assumption that the entire tower discharge is inlet to the pond and that the pond provides 100% of the condenser cooling water at all times. Alternative operating schemes would involve the selective bypass of the pond in an effort to more appropriately integrate the capacitive effect into the cooling circuit. The pond bypass flow circuit is shown in Figure 3.1. Such bypass operation might be used to cause the coolest part of the pond to be fed into the condenser at the hottest time of the day or at the specific time of day when the electrical load is highest.

The first case with pond bypass flow to be evaluated involved the use of a full-day pond with the size "2" tower at the ambient temperature conditions of 105°F maximum and 30°F range. For this case, the change in the plant power output at the afternoon period is small and a net loss of about 0.2% in the daily total electrical energy production (in comparison to the no-bypass situation).

A more interesting result was obtained for the "1/2-day" pond-tower system. For the same operating conditions of 105°F maximum ambient and 30°F range, the use of pond bypass was shown to be effective in shifting the peak power period toward the late afternoon. In Table 3.3, the detailed performance impact of two different bypass strategies is presented. For the situation where the pond is completely bypassed between 8 a.m. and 10 a.m. and 50% bypassed between 10 a.m. and noon and again from 8 p.m. to 11 p.m., the power increases during the late afternoon is about 2.0 MWe in comparison to the case with no pond bypass. Note, however, that bypass of the pond apparently always causes a reduction in the daily net electrical energy production.

Table 3.3

IMPACT OF POND BYPASS STRATEGIES

For: Tower Size "2"
"1/2 day" Pond

Max. Ambient = 1.05°F, Range = 30°F

Hours	Full Pond Operation	Bypass 100% at Hrs. 8, 9		Bypass 100% at Hrs. 8, 9 Bypass 50% at Hrs. 10, 11 Bypass 50% at Hrs. 20, 21, 22	
	Output	Output	% Change	Output	% Change
8	95.01 MWe	93.21	-1.9	93.2	-1.9
9	95.6	92.4	-3.3	92.4	-3.3
10	96.0	95.2	-0.7	94.1	-1.9
11	96.2	95.7	-0.5	94.0	-2.3
12	96.4	96.1	-0.4	96.0	-0.5
13	96.6	96.3	-0.3	96.3	-0.3
14	96.3	96.5	0.2	96.4	0.1
15	95.0	96.7	1.8	96.5	1.6
16	94.4	96.1	1.8	96.6	2.3
17	94.0	94.5	0.5	96.1	2.2
18	93.7	93.8	0.1	94.5	0.8
19	93.5	93.5	-0.0	93.3	-0.2
20	63.2	63.6	0.1	63.3	-0.5
21	63.3	63.5	0.2	63.4	0.1
22	63.0	63.5	0.5	63.6	0.6
Total	1333.4 MWhrs	1331.16	0.17	1330.3	-0.23

Bypass control strategies, as outlined in Table 3.3, can be employed in conjunction with tower shutdown to achieve very high power outputs during the afternoon period. Combining the extended bypass control scheme described in Table 3.3 with tower shutdown would yield a net electrical output of about 98.5 megawatts at the hottest time of the day (3 p.m.). The power level which would be achieved with a tower-only system with the same size dry tower would be only 89.8 megawatts. Thus, in terms of power output for this 105°F maximum temperature, the addition of the "1/2-day pond" results in a 10% increase in power and results in the recovery of about 85% of the power loss which would occur with simple dry cooling.

Performance with Reduced Steam Condensing Loads

The performance results presented in Section 3.3.2 are based on the assumption of steam condensing loads which would be expected on or about the summer solstice. High ambient temperatures may also occur about the time of the autumnal equinox when the daily total steam condensing load is less. The smaller daily steam condensing load would tend to benefit the pond-tower concept since more time would be available for the recooling of the pond and there should be a lower waste heat thermal storage requirement. Reducing the previously assumed period of steam condensing from 15 hours to 13 hours for various pond-tower systems, it was found that the power at the hottest time of the day would increase about 1% for the identical ambient temperature variations.

Section 4

DESIGN OF A WASTE HEAT THERMAL STORAGE POND

4.1 Introduction

In this section the many considerations relating to the design and construction of a thermally-effective and low-cost thermal storage pond are addressed. First, the general thermal performance requirements of a thermal storage pond are considered. Next, the current technology for the storage of large volumes of water is reviewed for application to the thermal storage pond concept. Subsequently, the comparative advantages and disadvantages of two different thermal storage pond design concepts are outlined.

4.2 Thermal Storage Pond Performance Requirements

4.2.1 Possible Thermal Behavior Characteristics

The benefit of a thermal storage pond in rejecting heat from a solar-thermal power plant will greatly depend on the thermal-hydraulic behavior of the pond. Both the flow rate and temperature of the water inlet to a thermal storage pond will vary with time depending on the mode of plant operation and ambient temperature. The manner in which the temperature of the pond outlet flow responds to this inlet flow rate and temperature variation will determine the heat rejection benefit of the pond.

Several general types of flow behavior could occur in a thermal-storage pond. Each of these different types of flow behavior would lead to a different thermal response characteristic. These general types of flow behavior are:

1. Plug flow,
2. Stratified flow,
3. Fully-mixed flow, and
4. Short circuited flow.

Plug-flow is the situation where the variation of the pond outlet temperature is identical to that of the inlet flow temperature, but out-of-phase with the inlet temperature by a constant time interval. This time interval is the plug-flow residence time of the pond (i.e., pond volume divided by flow-rate).

Stratified flow is the situation where there is a vertical temperature distribution in the pond. The outlet flow temperature depends on the temperature distribution in the pond at the level at which the water flows from the pond.

A fully-mixed pond is the situation in which the inlet flow is continually and completely mixed with the pond water inventory. At any given time the temperature in the pond is uniform and equal to the outlet water flow temperature.

Finally, a short-circuited flow is the situation where a portion of the pond is essentially stagnant and does not affect the outlet temperature. In the worst case, the outlet pond temperature would be equal to the inlet temperature and the pond would have no capacitative effect.

The actual thermal response in a real pond would undoubtedly exhibit some mixture of the specific characteristics. The thermal response of a specific pond will depend on the nature of the buoyant, pressure, inertial and frictional forces acting on the water in the pond. The factors which would affect the response would include:

1. The overall pond geometry,
2. the location and design of the inlet and outlet structure,
3. the time variation of the rate and temperature of the flow into the pond, and
4. the volume of the pond.

4.2.2 Impact of Pond Behavior on Pond Benefit

The results of the daily cycle performance studies reported in Section 3 have shown that plug-flow is preferred as compared to mixed

thermal-dynamic behavior. Complete mixing in the pond for the "design-day" conditions of ambient temperature and heat rejection was found to lead to about a 20% reduction in the power production benefit of the pond. This comparative analysis has also been extended to the cases of stratified flow and a partial short-circuit flow.

The results of the evaluation are presented in Figure 4.1. The performance of the cooling system for the pond behavior cases of ideal plug-flow, ideal stratified flow, fully-mixed flow, partially short-circuited flow, and completely short-circuited flow are presented. In the ideal plug-flow case, it is assumed that the water flows uniformly from one end of the pond to the other end with no mixing or density induced flows. In the ideal stratified flow pond, the effect of density induced flow and the potential for stable thermal stratification is used to advantage such that during the period of steam condensing the coolest water in the pond is used for condensing cooling (i.e., bottom withdrawal) and during the period of night-time pond cooling the hottest water is withdrawn for recooling in the dry tower (i.e., top withdrawal). In the partially short-circuited cases, it is assumed that a portion of the pond is stagnant and the remaining portion has a plug-flow residence time as indicated.

The results presented in Figure 4.1 demonstrate that the benefit of the pond is maximum with either a plug flow or a stratified flow pond. It is also apparent that a significant amount of mixing and some short-circuiting may be acceptable. Clearly, however, the pond must be designed to avoid rapid short-circuiting. Exactly what amount of short-circuiting and mixing is tolerable would depend on the economic penalty for the loss of power production and the cost of improving the thermal-hydraulic characteristics of the pond.

Figure 4.1
Impact of Pond Thermal Behavior On
Daily Power Production Benefit

4.3 Current Technology of the Storage of Large Volumes of Water

4.3.1 General

The problems involved in the construction of a thermal storage pond are similar to those faced by the water works industry. Economical designs and construction techniques for water storage facilities of the order of the size of the proposed thermal storage pond have been developed by this industry for preservation of valuable water supplies. The major differences between an impoundment constructed for water supply and thermal storage pond are that (1) the temperature of the stored water would be somewhat higher, and (2) the thermal storage impoundment may require an unconventional geometry in order to obtain the desired thermal behavior. The expected maximum temperature in a thermal storage pond would be about 120°F. This is not greatly in excess of the temperature of unheated bodies of water and does not appear to be a major constraint with regard to the application of conventional water storage technology to the thermal storage pond concept.

The volume of water to be impounded in a thermal storage pond would depend on many factors, including, of course, the size of the solar-thermal plant. An optimum pond size for a specific application can be arrived at only after a consideration of the tradeoffs among the costs and benefits of the pond to the heat rejection system. Nevertheless, for a 100 MWe solar-thermal plant, the optimum storage volume would be in the range of one to six million cubic feet.

Much of the information presented in this section regarding water storage technology has been adapted from Reference 13.

4.3.2 Available Storage Alternatives

The storage of large volumes of water may be accomplished by using tanks or so-called "cut-and-fill" reservoirs. Tanks can be classified as either wooden, steel, masonry, or concrete. Cut-and-fill reservoirs involves the excavation of a depression in the ground. In these reservoirs the earth removed in creating the depression is utilized to construct a fill or embankment around the top edge of the excavation to increase the effective depth of the depression and thus reduce the excavation costs. A number of materials have been applied to the cut-and-fill reservoirs to control both seepage and evaporation.

Wooden tanks, a common method of water storage until recently, do not appear to be suitable for the purpose of interest. Alternative types of tanks have significant economic and technological advantages over wooden storage tanks.

Steel tanks would provide a storage system with near zero water loss. The costs of steel storage tanks, however, are high in comparison to cut-and-fill types of impoundment. Additionally, steel tank storage systems can be adversely affected by corrosion, both internally and externally.

Masonry storage tanks are not, at present, commonly utilized for water storage. High labor costs and modern designs have nearly eliminated masonry tanks as an economic alternative except in special cases.

Both prestressed and reinforced concrete tanks could be applied for the purpose of interest. Again, however, the costs for these structures is seen to be large with respect to cut-and-fill reservoirs. Concrete tanks are subject to a variety of leakage problems which in many cases are solved through the use of a flexible membrane tank lining.

A cut-and-fill reservoir appears to offer the greatest potential for least cost. Cut-and-fill type reservoirs are presently widely used by both the water supply and pollution control industry for impounding volumes similar to those required by the thermal storage pond concept. In recent years significant improvements in the technology of lining and covering cut-and-fill impoundments have been achieved.

Critical to the design of a cut-and-fill reservoir is the design of the reservoir walls and the substrate drainage system. It is recommended that all walls have a slope of no less than 2 to ensure stability. In most cut-and-fill reservoirs, a liner of some type would be required to minimize seepage. A variety of lining materials have been applied and are currently available. To maintain the integrity of this liner a substrate reservoir underlining should be employed. The substrate underlining is required to provide uniform support for the lining and to allow for drainage of any leakage through the liner.

In addition to having a need for a good substrate material, it is also recommended that a cut-and-fill reservoir be constructed where the subsoil is stable whether wet or dry.

4.3.3 Lining for Cut-and-Fill Reservoirs

Lining for cut-and-fill reservoirs have been classified in three categories. These are flexible linings, rigid linings, and combination linings. The specific types of materials which have been applied as lining for reservoirs are grouped in Table 4.1 in these three categories.

A variety of factors would have to be considered in the selection of a lining system. These factors would include cost, seepage rate, durability, maintainability, reliability, and suitability to the particular application.

Table 4.1

LINING MATERIALS

I. Flexible Linings

Polyvinyl chloride and polyethylene (PVC and PE).
Chlorosulfonated polyethylene (Hypalon).
Chlorinated polyethylene (CPE).
Butyl rubber, ethylene propylene terpolymer (EPT), and
ethylene propylene diene monomer (EPDM).
Polychloroprene (neoprene).
Elasticized polyolefin.
Asphalt panels.
Compacted earth.

II. Rigid Linings

Concrete
Gunite
Asphalt Concrete
Steel
Soil Cement

III. Combination System and Other Systems

Bentonite
Chemical Treatment
Waterborne Dispersions
Sprayed Membrane
Concrete-Membrane Systems
Combined Flexible Membranes

For a thermal storage pond, near-zero seepage would be required if the heat rejection system is to be operated as a "dry" cooling system. This requirement necessitates the use of a continuous membrane lining. With the use of a membrane lining a major potential source of water loss is leakage at the seals at underwater penetrations such as might exist at intake and discharge structures, roof supports, and flow baffles. The flow velocities encountered in a thermal storage pond should not be sufficient to have any effect on the selection of the lining system.

For the purposes of this work, the cost of lining will be assumed to be \$0.75/ft². Also, costs for the preparation of an acceptable lining substrate and drainage system will be included in the total cost. The seepage losses from the thermal storage pond could be a significant part of the cost of operating the system. The seepage rate is difficult to estimate since for membrane linings the amount of seepage depends, in many cases, on the quality of the installation and not on the basic physical characteristics of the lining material.

4.3.4 Covers for Cut-and-Fill Reservoirs

The technology for covering public water supply reservoirs should be directly applicable to the proposed thermal storage pond. Covers are placed on storage reservoirs primarily to preserve the water quality. A TSP cover would have the dual benefit of suppressing evaporation and preserving water quality. Measures which would partially control evaporation and which would not preserve the quality of the stored water such as monolayer additives and floating rafts appear less attractive.

Chin⁽¹⁴⁾ has reviewed in detail many of the options available for constructed reservoir covering. Projected costs for steel and pre-cast concrete coverings supported on columns ranged from 2 to 3 times those of flexible floating covering. Past uncertainties concerning the working lifetime of flexible covers seems to have been resolved and the successful use

of this type of covering has been discussed by several authors^(16,17). Currently, vendors of the patented "Roofloat" system are anticipating 40 year lifetimes for conventional storage reservoir applications⁽¹⁵⁾. This cover, which is made from nylon reinforced Hypalon synthetic rubber appears to be satisfactory for use in covering a thermal storage pond.

A floating membrane cover cost of \$2.12 per square foot will be assumed for the purposes of estimating thermal storage pond construction costs. The drag on a floating cover which would result during the operation of a thermal storage pond has been evaluated and found to be insignificant.

4.4 Comparison of Horizontal and Vertical Flow Options for Pond Design

4.4.1 Horizontal and Vertical Flow Concepts

The effectiveness of a thermal storage pond can be maximized if the average residence of "holdup" time of a slug of water passing through the pond is maximized. Plug-flow and stratified-flow, which act to maximize the average residence time, are most beneficial types of flow behavior, while mixing and short-circuiting have the impact of effectively decreasing the "holdup" time and reducing the potential benefit of the stored volume of water.

Two pond design and operation concepts have been considered as options for achieving the desired plug or stratified flow. One option takes advantage of the stability of thermal stratification to achieve a controlled flow in the vertical direction. The other option is based on attempting to achieve plug flow in the horizontal direction. In this case, the intent to design the pond such that density-induced flow will not adversely effect the overall plug-flow behavior of the pond.

The considerations relevant to the design and operation of thermal storage ponds based on these two concepts are discussed in this section.

4.4.2 Horizontal-Flow Thermal Storage Pond

General Design Concepts

A horizontal-flow thermal storage pond would be one in which an attempt would be made to establish the required plug-flow behavior in the manner shown in Figure 4.2. Ideally, a uniform flow velocity would exist throughout the pond with each imaginary "plug" passing through the pond at the plug-flow velocity V_p .

$$V_p = Q/A$$

where Q = net flow through pond, and
 A = cross-sectional area of pond.

For conditions of uniform temperature in the pond, this is approximately what would occur since the flow would be turbulent. However, in the thermal storage pond the temperature variation, and hence the density variation of the individual plugs can cause significant deviations from this simple flow behavior.

In Figure 4.3 is presented the approximate temperature profiles which would exist in an ideal plug-flow pond for a typical design day. Each of the points plotted in Figure 4.3 represent the average temperature of a plug of flow passing through the pond. Each plug has an equivalent volume of one hour of circulating water flow. Note that the abscissa in Figure 4.3 is a non-dimensional distance parameter since, by definition, the ideal plug flow temperature profiles are independent of the specific geometry of the pond. Examining Figure 4.3, it can be seen that the maximum temperature difference from one end to the other is about 20°F, while the maximum difference between two idealized plugs is about 8°F.

These temperature differences could lead to density-induced flow which could significantly alter the overall flow field in the pond as depicted in Figure 4.4. The resulting effect may be the short-circuiting of the thermal capacitance of the pond. The potential magnitude of this effect can be evaluated by comparing the bulk plug-flow velocity in the

Figure 4.2
Ideal Horizontal Plug-Flow Thermal Storage Pond

"1/2 day" pond
Max. Ambient 105°F
Range 30°F

Figure 4.3
Pond Temperature Profiles with Ideal Plug-Flow

Figure 4.4
Possible Effect of Density Gradient in Horizontal Flow Pond

pond to the potential velocity of the density induced currents. The nominal plug-flow velocity would be only 0.012 ft/sec. in a 2,000,000 ft³, square-shaped storage pond which is 20 feet deep. The potential density-induced flow velocity in this pond can be estimated from

$$V_D = \frac{1}{2} \left(g \frac{\Delta \rho}{\rho} H \right)^{1/2}$$

where V_D = density-induced flow velocity

g = gravitational acceleration

$\Delta \rho$ = density difference

ρ = average density

H = depth of pond

This velocity equation is derived by applying an energy conservation equation to an initially vertical density interface. Assuming an 8°F temperature difference, the resulting density-induced flow velocity is 0.74 ft/sec or about 62 times that of the nominal plug-flow velocity. Thus, the density-induced currents would dominate the flow in this type of pond.

Two corrective actions are possible to produce an effective plug-flow in the horizontal direction. One is to design the pond such that the plug-flow velocity is greater than the density-induced velocity and the other is to somehow inhibit the action of the density-induced flow. The first approach is not practicable since to achieve a flow situation where the plug-flow velocity dominates the density-induced flow would require an excessively long and narrow thermal storage pond (channel). This would be an uneconomical design.

An inhibiting of the density-induced flow could be achieved by placing flow constrictions internally in the thermal storage pond, as depicted in Figure 4.5. The intent in placing flow constrictions in the path of the channel-type flow is to provide for the repeated remixing of any temperature strata which might develop in the pond. The repeated remixing of the flow should act to diminish the proagation of density currents in the storage pond. This potential application of this type of horizontal-flow pond design--a channel with flow constrictions--is examined in detail in the next section.

Figure 4.5
Horizontal-Flow Pond with Mixing Flow Constrictions

Horizontal Flow Channel with Flow Constrictions

The key to the performance of a horizontal flow thermal storage pond as shown in Figure 4.5 is the flow behavior at the flow constrictions. Because of the low plug-flow velocity in the pond segments between the constrictions, the density-induced flow would likely dominate and a vertical temperature distribution would develop. For effective mixing to occur at the constriction, the flow into each constriction would necessarily have to be drawn uniformly over the depth of the pond and, subsequently, mixed in the jet discharges into the downstream pond segment. Specific considerations relevant to the design of an effective flow constriction would be:

- 1) The characteristic dimension of the constriction should be of the same order of size or the depth of the pond. This is necessary to allow the production of eddies in the jet discharge large enough to promote top-to-bottom mixing in the discharge region.
- 2) The flow velocity through the constriction should be as high as possible. The constriction will produce the desired effect only if the hydrostatic pressure difference across the constriction dominates any local buoyancy forces. This is necessary to ensure near uniform withdrawal from the upstream segment in the presence of a thermocline. A high velocity jet discharge would also ensure that the discharge is unstable (i.e., not stratified).
- 3) The constriction(s) should be uniformly spaced along the flow cross-section in order to avoid "dead spaces" or stagnant regions.
- 4) The head loss across the constriction should be small. The pond would necessarily have to be designed to hold the volume of water in both the flow and no-flow situations.

- 5) The constriction should result in good mixing in both the advancing cold front and hot front situations.
- 6) The overall geometry of the flow constriction and pond segments must be compatible with cut-and-fill reservoir constriction practice.

A variety of different pond and constriction designs have been evaluated for a nominal 100 MWe solar plant in light of these objectives and constraints. The design parameters employed in this evaluation are

1. Flow rate = $86 \text{ ft}^3/\text{sec}$,
2. Temperature differences = 8°F ,
3. Total volume = $3 \times 10^6 \text{ ft}^3$.

The design presented in Figure 4.6 is postulated to be a reasonable compromise based on a qualitative and, where possible, quantitative arguments. The postulated design involves a channel nominally 100 ft. wide and 30 ft. deep with sloping sidewalls. The length and number of constrictions on the pond (channel) would depend on the desired storage volume. At each point of flow constriction, there would be two orifices approximately eight ft. in diameter.

This specific orifice size has been selected by first determining the size which would be required such that the pressure loss through the orifices was five times the potential buoyancy force due to the 8°F temperature difference over a 20 ft. depth. Two orifices are seen as a minimum in attempting to avoid stagnant regions in the pond segments. Next, available criteria with respect to the stability of selective withdrawal and the stability of thermal jets for two-layer stratified systems have been applied.

The first criterion applied relates to the performance of the orifice constriction in not selectively withdrawing water from the upstream segment. The situations which would appear to maximize the tendency for

Figure 4.6

Proposed Flow Constriction Design for Horizontal-Flow Pond

selective withdrawal for the proposed design is depicted in Figures 4.7a and 4.7b for a two-layer situation. For the postulated constriction design, the criterion given by Harleman et al⁽²¹⁾ for the selective withdrawal with a skimmer wall geometry indicates that the withdrawal of both layers would occur for the condition shown in Figure 4.7a. The alternative situation, which would also tend to maximize the potential for selective withdrawal, is when there is a shallow cold layer upstream of the orifice as depicted in Figure 7b. There appears to be no selective withdrawal flow criterion which has been developed for this particular situation. Related criteria, such as that suggested by Craya⁽¹⁹⁾, indicates that some significant volume of cold water would remain trapped below the level of the orifice.

The discharge of the orifice must not result in a thermally stable two-layer flow in the downstream segment. To evaluate the potential for this to occur, the criteria of Jirka⁽²⁰⁾ has been applied for the situation depicted in Figure 4.7c. The result is that, for the postulated orifice design and for the assumed "worst-case" 8°F temperature difference between the jet and the ambient fluid, the jet should not be stable and a mixed flow field should develop downstream.

Guyer⁽¹⁰⁾ has shown that flow constricting barriers similar to those described above can be effective in promoting a horizontal plug-type flow for conditions similar to those which exist in the proposed horizontal-flow thermal storage pond. However, in this situation, vertical slot constrictions were employed which enhanced the non-selective withdrawal from the upstream segment. However, use of vertical slot constrictions do not appear to be feasible in the present case since to obtain the required flow velocity the slot constriction would need to be excessively narrow. A slot constriction with a small aspect ratio (width/height) would not be effective in promoting the vertical mixing in the discharge into the downstream segment.

Figure 4.7a

Two-Layer Design Condition for Maximum Selective
Withdrawal-Drawdown Condition

Figure 4.7b

Two-Layer Design Condition for Maximum Selective
Withdrawal-Drawdown Condition

Figure 4.7c

Two-Layer Design Condition for Maximum Jet Stability

4.4.3 Vertical-Flow Thermal Storage Pond

General Design Concept

A vertical-flow thermal storage pond would be one in which the required thermal response would be achieved by taking advantage of the stability of a thermally stratified reservoir. The vertical-flow concept is illustrated in Figure 4.8. Ideally this pond would function such that there is little or no mixing at the flow inlet and at the outlet water would be selectively withdrawn from the pond. The thermal response of an ideal stratified pond would be somewhat different from that of an ideal plug-flow pond. However, as discussed in Section 4.1, the benefit of a stratified flow pond would at least equal that of a plug-flow pond.

The key in the design of the vertical-flow or stratified-flow pond is the design of the inlet flow header and flow withdrawal structure. To minimize cost, it is of interest to use a single structure at the top of the pond and a single structure at the bottom of the pond for both flow discharge and withdrawal. In the following, the requirements for the design of the flow headers are evaluated assuming a two-layer thermally stratified pond.

Stratified Flow Withdrawal Criteria

If line sinks are used for water withdrawal from a stratified pond, fairly simple criteria can be used to describe the layer dimensions as shown in Figure 4.9.

For either withdrawal from the top or bottom, only the desired layer will be withdrawn so long as⁽²¹⁾

$$U_{\infty} < C \sqrt{g' z}$$

where U_{∞} = velocity in stratified layer at infinity

$g' = g \cdot \Delta\rho/\rho$

g = acceleration due to gravity

ρ = density of fluid

Condenser Cooling Operating Mode

Regeneration Operating Mode

Figure 4.8
Schematic Cross-Section of
Stratified Vertical Flow Pond

a. Surface Withdrawal

b. Bottom Withdrawal

Figure 4.9
Selective Withdrawal Flow Criteria for Two-Layer Systems (Ref. 21)

$\Delta\rho$ = density difference between layers
 z = height of layer being withdrawn
 C = constant = 1.52, withdrawal from top
 = 1.66, withdrawal from bottom

A 10°F temperature difference for the two-layer thermal structure shown in Figure 4.9 has been employed to evaluate the ability to achieve selective withdrawal. This is representative of the bulk temperature difference between the "hot" and "cold" volume of water which would exist at any time in the pond. For purposes of evaluation of withdrawal concepts, a hypothetical pond 350 by 350 feet by 20 feet deep has been evaluated. The assumed rate of withdrawal is 80 ft³/sec.

The feasibility of selective withdrawal has been evaluated by determining the maximum layer velocity which can be sustained during withdrawal without drawdown (or drawup) of the interface as depicted in Figure 4.9. The approximate criterion for selective withdrawal is

$$U_{\infty} < 1.5 \sqrt{g' z}$$

while $Q = U_{\infty} \cdot A$ = withdrawal flow

where A = flow area = 350 ft. x z

Combining the above yields

$$Z_{\min} = \left(\frac{Q}{1.5 \sqrt{g'} \times 350} \right)^{2/3}$$

where $Z_{\min}$ is the minimum depth of the layer which can be withdrawn.

Computing $Z_{\min}$ for the proposed design yields

$$Z_{\min} = 0.67 \text{ feet.}$$

Clearly, this illustrates that selective withdrawal to very shallow depths is feasible and should not be a major obstacle in the design of a stratified flow thermal storage pond. This holds for both the top and bottom withdrawal situations since the equation for Z_{min} is a good approximation for both situations.

Flow Discharge in Stratified Flow Pond

Two questions arise with regard to the discharge of flow into the stratified thermal storage pond. First is whether or not the flow jet affects the thermal strata existing in the pond. Second, it is of interest to know whether or not the flow will distribute laterally over the pond area at a velocity which is large in comparison to the bulk vertical velocity of the pond.

The impact of the inflow will be minimized by having a horizontal discharge at a low velocity. The vertical temperature distribution existing in the pond will tend to inhibit mixing in the vertical direction which might be induced by the discharge into the pond. Specific requirements for the design of the discharge have not been developed. However, approximate analysis indicates that use of headers with small discharge ports (about 3" dia.) should have very localized effects in comparison to the total depth of the pond.

The equation presented by Guyer⁽¹⁰⁾ for the propagation of a density front can be used to estimate the rate of spreading of the discharge into the pond. Assuming the spreading front is 1 foot in depth and the temperature difference is 10°F between the spreading water and the ambient water leads to the calculation of a flow velocity which is at least two orders of magnitude in excess of the bulk vertical flow velocity. Thus, for the purpose of predicting the thermal response of the pond, it should be adequate to assume that the thermal strata in the pond are established essentially instantaneously once the flow has been discharged into the pond.

Decay of the Thermocline

The potential effect of heat transfer between the thermal strata existing in the pond has been evaluated and has been found to be minimal. Simple conduction is the only important mode of heat transfer since the thermocline existing in the pond will strongly suppress any vertical convection. Also, heat transfer to the walls of the pond has been evaluated and should have a minimal effect on the general thermal-hydraulic behavior.

Impact of Time-Varying Discharge Temperature

Generally, the stratified thermal storage pond would work most effectively when, during the heat storage mode of operation, the temperature of the discharge into the top of the pond rises continually. In this situation, each slug of water is more buoyant than the preceding slug and thus spreads across the pond at the top surface. In the pond regeneration mode when the flow is discharged in the bottom of the pond, the opposite situation would generally exist--the inlet flow would be continually decreased in temperature and thus continually spread across the bottom of the pond.

In certain situations this may not be the case. Near the end of the heat storage mode, the inlet flow temperature may decrease due to the declining ambient temperature. Conversely, the temperature of the inlet flow to the pond during the latter part of regeneration period may rise as the ambient temperature begins to rise. Either situation could result in the discharge flow into the pond sinking (or rising) to some intermediate level in the pond. The situation could potentially have an adverse impact on the thermal response characteristic of the pond by disrupting the thermal strata. However, consideration of the specific situations in which the temperature reversal might occur does not indicate this to have a potentially large impact. The reasons for this are as follows:

- 1) During the heat storage mode of operation, if the temperature of the discharge into the pond began to decrease because of the declining ambient temperature, the temperature of the discharge would rapidly begin to approach the still increasing pond outlet temperature. This is shown in Figure 4.10. It is reasonable to assume that, during any period of power plant operation, the pond would be utilized only when it functions as a heat sink and not a heat source. Operation of the pond would, in most cases, cease shortly after the discharge temperature begins to decline. Thus, as a practical matter, it is not expected that the pond would operate with a decreasing inlet temperature during the heat storage mode (at least not for any extended length of time).
- 2) During the regeneration mode, a rising inlet temperature would not be encountered except in unusual conditions when the pond is abnormally cold. This is because normally the temperature of the dry tower outlet (pond inlet) would not closely approach the ambient temperature as shown in Figure 4.10. Thus, the increasing ambient temperature in the morning before the plant startup would not greatly affect the pond cooling process.

4.4.4 Comparative Cost of Horizontal-Flow and Vertical-Flow Thermal Storage Ponds

To evaluate the horizontal-flow concept in comparison to the vertical-flow concept, both the thermal response and the cost per unit of storage volume need to be considered. In order to evaluate the comparative cost of the two concepts, total costs have been estimated based on the schematic designs presented in Figure 4.5 (Horizontal-Flow) and Figure 4.11 (Vertical-Flow). These costs are presented in Tables 4.2 and 4.3. Most of the unit construction costs presented in Tables 4.2 and 4.3 were obtained from Reference 22.

Figure 4.10

Overall Temperature Difference in Stratified Thermal
Storage Pond Compared to Ambient Temperatures

Figure 4.11
Proposed Preliminary Design of Stratified Vertical Flow Thermal Storage Pond

Table 4.2.

COST OF VERTICAL-FLOW POND

<u>Item</u>	<u>Quantity</u>	<u>Unit Cost</u>	<u>Cost</u>
Excavation	2,400,000 ft ³	\$2/cubic yard	\$180,000
Side Wall	16,500 ft ³	\$145/cubic yard	89,000
First Grade	89,000 ft ²	\$1.45/square yard	14,000
Substrate	178,000 ft ³	\$5/cubic yard	33,000
Second Grade	89,000 ft ²	\$1.45/square yard	14,000
Lining	102,600 ft ²	\$0.75/ft ²	77,000
Covering	80,000 ft ²	\$2.12/ft ²	170,000
Piping			
Cold	360 ft	\$42/ft	15,000
Hot	1,500 ft	\$28/ft	42,000
			Total \$634,000

For 2,400,000 ft³ storage pond = "1/2 day" pond for 100 MWe plant.

Table 4.3.

COST OF HORIZONTAL-FLOW POND

<u>Item</u>	<u>Quantity</u>	<u>Unit Cost</u>	<u>Cost</u>
Excavation	2,400,000 ft ³	\$2/cubic yard	\$180,000
Side Wall	18,000 ft ³	\$145/cubic yard	97,000
Baffles	20,000 ft ³	\$145/cubic yard	108,000
First Grade	145,000 ft ²	\$1.45/square yard	23,000
Substrate	290,000 ft ³	\$5/cubic yard	53,000
Second Grade	145,000 ft ²	\$1.45/square yard	23,000
Lining	157,000 ft ²	\$0.75/ft ²	117,600
Covering	130,000 ft ²	\$2.12/ft ²	275,000
Piping	200 ft	\$60/ft	12,000
			Total \$888,000

For 2,400,000 ft³ storage pond = "1/2 day" pond for 100 MWe plant.

The cost of the vertical-flow stratified pond is estimated to be significantly less. This is because a greater depth can be used to minimize the expensive lining and covering and because less concrete is required for the structure.

4.5 Preliminary Recommendations for Thermal Storage Pond Design and Construction

Alternative approaches to the design and construction of a thermal storage pond for use with a dry cooling heat rejection system at a solar-thermal electric plant have been considered. Both thermal performance and cost factors have been considered.

The general result is that a stratified, vertical-flow storage reservoir appears to offer the best potential for lowest cost and maximum thermal performance. The difficulties associated with inhibiting density-induced flow in a horizontal-flow type of pond limits the thermal performance potential of the alternative. Also, the construction cost per unit of stored volume for the horizontal-flow pond is estimated to be higher than that of a stratified pond. Correct design of the flow inlet and outlet devices at the top and bottom of the vertical-flow pond should result in a good approximation to ideal stratified flow thermal behavior.

Conventional techniques for the construction of "cut-and-fill" water storage reservoirs can be utilized for the construction of a thermal storage pond. Currently available membrane lining and covering materials may be applied to minimize seepage and evaporative water losses.

Section 5

ECONOMIC EVALUATION OF THE COMBINED DRY COOLING TOWER AND THERMAL STORAGE POND WASTE HEAT REJECTION SYSTEM

5.1 Introduction

In this section the economics of the combined dry cooling tower and thermal storage pond system are evaluated in comparison to conventional dry cooling systems. The cost of the system is evaluated for second-generation steam-electric solar-thermal plants sited in Barstow and Inyokern California. All costs are presented in 1979 dollars. The economic analyses presented in this section focus on the cost of heat rejection for 100 MWe solar plant with 6 hours of thermal energy storage sited at Inyokern

In the economic analysis, particular attention is paid to the effect of the turbine operating characteristics and operating limits on the economics of the combined pond-tower dry cooling system.

5.1.1 Approach to Economic Evaluation

The cost of each of the cooling systems considered in the economic evaluation of the combined pond-tower dry cooling systems is computed as a lifetime cost of owning and operating the heat rejection system. The general cost equation is

$$C = C_c + C_o + \Lambda_c + \Lambda_o$$

where C = present-worth cost of owning and operating the heat rejection system over its lifetime

C_c = present-worth of capital investment associated costs

C_o = present-worth of cost of operating the cooling system (pumping power and maintenance)

Λ_c = present-worth cost of the loss of plant net operating capacity attributable to the cooling system (in reference to the plant maximum output)

Λ_o = present-worth cost of the loss of electrical energy production attributable to the cooling system (in reference to the plant maximum output)

The capital investment cost, C_c , is computed from

$$C_c = Pwf_i^m (afcr) C_I$$

where Pwf_i^m = present-worth factor for a series of uniform annual payments at interest rate i for m years
 $afcr$ = annual fixed charge rate
 C_I = total capital cost

The heat rejection system capital costs include all costs associated with the acquisition and construction of all plant equipment "downstream" of the turbine exhaust flange.

The operating cost, C_o , is determined using the equation

$$C_o = Pwf_i^m (F \cdot \lambda_p \cdot E_o + C_m)$$

where F = the availability of the plant
 λ_p = fuel cost for the electrical power to operate the pumps and fans
 E_o = annual energy required to operate the heat rejection system (assuming no outages)
 C_m = annual maintenance cost

The loss of capacity penalty is computed from

$$\Lambda_c = \Lambda_{c,m} + \Lambda_{c,s}$$

where $\Lambda_{c,m}$ = capacity penalty due to meteorologically-induced power losses, and
 $\Lambda_{c,s}$ = capacity penalty for power requirement for operating the cooling system

The former term is computed as

$$\Lambda_{c,m} = Pwf_i^m (afcr) K_p \delta_m$$

where K_p = the incremental cost of peaking type of electrical generating units (\$/KW), and
 δ_m = the loss of capacity due to condenser pressure variations (KW)

The latter term is computed as

$$\Lambda_{c,s} = Pwf_1^m (afcr) K_b \delta_s$$

where K_b = the incremental cost of baseload type of generation capacity (\$/KW)
 δ_s = power required to operate the heat rejection system

The two capacity penalties are evaluated differently to reflect the real costs in replacing the lost capacity. The loss of capacity due to the high ambient temperature is of only short duration while the power requirement of the cooling system is nearly constant. For the present purpose, the term δ_m is assumed to be the annual maximum loss of capacity.

The loss of energy production penalty, Λ_o , is determined by

$$\Lambda_o = Pwf_1^m \cdot F \cdot \lambda_r \cdot E_m$$

where λ_r = average incremental fuel cost for the utility system
 E_m = annual loss of energy production resulting from condenser pressure variations

Having determined the total present worth cost, C , the incremental cost of power attributable to the cooling system is

$$e_i = \frac{C \cdot 1000}{Pwf_1^m \cdot G}$$

where e_i = incremental cost of power production attributable to the heat rejection system in excess of a zero cost cooling system with ideal performance, and
 G = net annual power generation of an ideally cooled plant

To correctly evaluate the cost of a particular type of heat rejection system it is necessary to determine the particular design which yields the minimum total cost of heat rejection. For a simple dry cooling system the major system design variables which need to be optimized are the general size of the dry cooling tower and the circulating water flow rate. For combined pond-tower systems, the volume of water stored in the pond is an additional system design variable.

5.1.2 Plant and Cooling System Thermal Performance Models

Power Plant and Operating Conditions

The power plant modelled in the case studies presented in this section is a nominal 100 MWe second-generation solar-thermal steam-electric plant. The plant is assumed to have the same base conversion efficiency when operating from either the receiver or the thermal energy storage supply. Two cases of thermal energy storage are considered - zero and six hours of capacity. The operating characteristic of the turbine with respect to condenser pressure (temperature) variations is assumed to that of a conventional tandem-compound 3600 RPM, 2400 psig/1000°F/1000°F steam turbine. The operating range of the turbine is assumed to extend to 8" Hg back pressure for operation with a dry cooling system as shown in Figure 5.1. Cases in which the turbine operating back pressure is limited to the conventional limit of 5" Hg are also considered. For the particular turbine chosen, operation at 5" Hg back pressure results in about a 5% power loss while operation at 8" Hg results in about an 11% power loss.

The turbine performance characteristic can be of major importance in determining the cost of heat rejection. An alternative choice to the turbine presented in Figure 5.1 would be a so-called "high-pressure" design which has a smaller exhaust annulus area in the low pressure sections. As illustrated earlier in Figure 2.1, this turbine would have less performance deterioration at high exhaust pressures but is considerably less efficient at normal operating condensing pressures. It is because of this lower efficiency at normal

Figure 5.1 Heat Rate Curve for Turbine-Generator

operating pressures (which would be encountered most of the time) that this type of turbine design has not been considered. The high capital cost of the solar energy collection system mandates maximum conversion efficiency.

To correctly model the performance of a cooling system for a solar-thermal plant is necessary to represent the relationship between the ambient temperature and the steam-condensing load. To account for this relationship, a simulation program for solar thermal power plants was used to produce a times-series of steam condensing loads (23). The steam condensing loads are based on simulation of the plant operation during a year represented by a time-series of meteorological data (one hour time steps). The steam condensing data along with the ambient temperature data accurately represent the conditions under which the cooling must operate. Thus consistent data sets of ambient temperature and steam-condensing load for both sites (for the two cases of zero and six hours thermal energy storage) were used to model the operating requirements and environment of the heat rejection system.

Heat Rejection System and Component Designs

The type of dry cooling system modelled in these case studies is a simple water-loop system employing a conventional surface condenser and a mechanical-draft finned-tube type of dry cooling tower. The pond is located at the tower discharge and may be bypassed as shown in Figure 3.1. The condenser may also be bypassed when there is no steam condensing load. The circulating water pumps are assumed to be located just upstream of the condenser.

The dry tower is assumed to be composed of a number of identical cooling tower modules, each having its own fan. The size of the dry cooling tower is thus specified by the number of dry tower modules. All the dry tower modules are in a parallel flow arrangement with equal flow through each module. The flow arrangement of the air and water flows in a module is crossflow. The finned-tube bank in the module is 61 feet long and composed of one inch O.D. tubes arranged in a two-pass flow configuration. Each dry

tower has an overall heat transfer coefficient of 1,150,000 BTU/hr · °F and an air flow rate of 8,050,000 lbs/hr. The heat transfer effectiveness of the dry tower is assumed to be independent of temperature.

The condenser is assumed to be constructed with tubes 20 feet long with a one-inch O.D. It is also assumed that the condenser is designed to yield a 5°F approach to the steam condensing temperature.

The thermal storage pond is assumed to be a stratified vertical flow type with a membrane covering and lining as described in section 4. The pond is assumed to exhibit ideal stratified flow behavior and there is no mixing of the flow inlet to the pond at different time-steps. The pond outlet temperature for a specific time-step is the average temperature of flow discharged from the pond during that time-step. The pond cross-sectional dimensions are those given in Figure 4.11. The volume of the pond thus depends only on the length. Heat transfer from the pond surface is computed using the equations recommended in Reference 24.

Cost Models

The capital cost algorithms used in the study have been formulated to be consistent with those given in References 12 and 25. The capital costs are computed in 1979 dollars and are given as functions of the number of dry tower modules, ND; the circulating water flow rate and the volume of the thermal storage pond. The specific cost relationships are

- 1.) Dry tower modules and foundations

$$C_1 = \$246,000 \times ND$$

- 2.) Condenser

$$C_2 = \$1,347,000$$

- 3.) Electrical Equipment

$$C_3 = \$40,000 \times ND$$

- 4.) Pipelines

$$C_4 = \$14 \times GPM$$

- 5.) Circulating Water Pumps and Motors

$$C_5 = \$3.6 \times \text{GPM}$$

- 6.) Circulating Water Pump Structure

$$C_6 = \$0.362 \times C_5$$

- 7.) Thermal Storage Pond

$$C_7 = \$0.22 \times \text{Volume} + \$100,000$$

- 8.) Indirect Cost₇

$$C_8 = 0.25 \sum_{i=1}^7 C_i$$

The operating costs and penalties are evaluated somewhat differently for the simple dry cooling system and the combined pond-tower dry cooling systems. The specific assumptions are presented in Table 1.

The penalty rates of \$1,350/KW and zero fuel cost are used for the fan and pump power for the simple dry system and for the pump power for the combined system since these power requirements are constant and non-interruptable. These particular rates reflect the capital and fuel costs of incremental solar plant capacity and energy costs. No capacity penalty is charged for the fan power for the combined pond-tower systems since this is an interruptable power requirement (see section 4). However, for the fan power for the combined system the energy penalty rate of 31.5 mills/KWhr (40 year levelized value) is assigned since there is a fuel cost associated with the energy consumed in the operation of the fans. The rate of 31.5 mills/KWhr reflects a time-average incremental fuel cost of a typical utility system.

The capacity charge rate of \$200/KW is charged for the power losses resulting from the variation in the condenser pressure. This incremental capacity charge rate is representative of the incremental capital cost of the type of generating equipment many utilities install to meet intermittent peaking loads of relatively short duration. Along with this capacity charge, a fuel cost rate of 77.6 mills/KWhr (40 year levelized value) is used. This reflects the fuel cost and efficiency of typical peaking types of generating units.

Table 5.1

Cost Assumptions for Performance Penalties

	<u>Simple Dry Cooling</u>	<u>Combined System</u>
Capacity Penalty for		
Fan Power	\$1350/KW	0
Pump Power	\$1350/KW	\$1350/KW
Condenser Pressure Variation	\$200/KW	\$200/KW
Energy Penalty for		
Fan Power	0.00	31.5 mills/KW hr
Pump Power	0.00	0.00
Condenser Pressure Variation	77.6 mills/KW hr	77.6 mills/KW hr

The assumption which has been employed regarding the general economics factors are as follows:

- 1.) annual fixed charge rate = 18%
- 2.) real increase in the cost of fuel for the utility system = 4%
- 3.) annual interest rate = 9%
- 4.) plant lifetime 40 years

Additionally it has been assumed that the plant is shutdown 15 days per year for maintenance and that the forced outage rate is 10%.

The annual cost of maintenance for the cooling system is assumed to be 1% of the capital cost. Also, for the combined pond-tower systems no cost is assigned to the water needed to fill the storage pond. This is a reasonable assumption since, even for very high water costs in the order of \$1,000 per acre-foot, the total cost of the water would be small in relation to the other capital costs.

Simulation Procedure and Operating Rules

To model the annual performance of a specific cooling system, the performance of the plant is computed for each hour of the year for the existing steam condensing requirement and ambient temperature. The performance simulation covers one calendar year and, for those cases in which a thermal storage pond is a part of the cooling system, the pond is assumed to have an initial temperature of 50°F.

For the cooling systems with no thermal storage pond, the system operating rules are simple. The circulating water pump and the dry tower fans operate only when there is a steam condensing load. For the combined pond-tower systems the assumed operating rules are

- 1.) if there is a steam condensing load and if the thermal storage pond can provide cooler water than the dry tower, the circulating water pump and dry tower fans operate and there is no pond bypass or condenser bypass.

- 2.) if there is a steam condensing load and if the thermal storage pond cannot provide cooler water than the dry tower, the circulating water pumps and dry tower fans operate and the pond is bypassed.
- 3.) if there is no steam condensing load and the tower can cool the pond, the circulating water pumps and the dry tower fans operate and the condenser is bypassed.
- 4.) if there is no steam condensing load and the tower cannot cool the pond then the cooling system does not operate.

The System Simulation and Cost Analyses Computer Program

The FORTRAN computer program which has been formulated for the economic evaluation of the combined pond-tower water heat system is based on the simulation of the system thermal performance for a one-year period with one-hour time-steps. The program was formulated to model both simple and combined pond-tower dry cooling systems and has as its input

- 1.) the time-variation in the steam condensing load and the ambient temperature
- 2.) the size of the dry tower, the circulating water flow and the storage volume of the pond, and
- 3.) the major economic parameters.

Internal to the program are the performance characteristics of the turbine-generator, the physical characteristics of the heat rejection equipment and the capital cost of the heat rejection equipment. The output includes both the thermal and economic performance of the input system design.

A simplified flowchart of the computer program is presented in Figure 5.2. The state-state operating condition of the system is computed for each hour of the year using the control logic, the current steam condensing load and ambient temperature, the current temperature distribution in the pond, and the input-output characteristics of the condensor and dry tower. The performance is recorded for each hour as the loss of power with respect to the maximum power.

Figure 5.2

Computer Program Flow Chart

5.2 Economics of Conventional Dry Cooling at Inyokern

The cost of conventional dry cooling with solar-thermal plants has been determined in order to define a baseline for the economic evaluation of the combined pond-tower dry cooling system. In Table 5.2 is presented a cost and performance summary of an optimum simple dry cooling system for the Inyokern plant with 6 hours of thermal energy storage. Table 5.3 summarizes the specific capital costs for this system. The basic result presented in Table 5.2 is that the optimum number of dry cooling modules is fifteen (15) and the incremental power production cost is 8.24 mills/KWhr. This size dry tower results in a condensing pressure of 8" Hg (152°F) and a power loss of about 11 MWe during the hottest hour of the year (102°F).

The optimum design for the case of a steam turbine with a 5" Hg condensing pressure limit was found to be 30 dry tower modules. The incremental power production cost for this system is 10.5 mills/KWhr.

Also, the optimum design for the case of no thermal energy storage has also been determined. The size of optimum dry tower for the 8" Hg limit case is identical to the 6 hour thermal energy storage case. The incremental power production cost is higher (9.35 mills/KWhr) because the total annual amount of electrical energy produced with no thermal energy storage is considerably less.

5.3 Economics of Combined Pond-Tower Dry Cooling at Inyokern Turbine with 8" Hg Back Pressure Limit

The first combined pond-tower designs to be considered are those in which the amount of the dry cooling capability is identical to that which is optimum with no thermal storage pond. As shown in Table 5.2 this size is 15 dry tower modules. The sizes of the ponds added to this dry tower and resulting incremental power production cost are:

Inyokern Site	1 acre pond	= 7.01 mills/KWhr
with	2 acre pond	= 6.64 mills/KWhr
6 hours TES	4 acre pond	= 6.48 mills/KWhr

Table 5.2
Optimum Simple Dry Cooling System

Site - Inyokern	
Solar plant	100 MWE, 6 hours TES
Design parameters	8" Hg condenser pressure limit
1) number of dry cooling modules	15
2) condenser flow rate	87.4 cfs
3) regeneration flow rate	0 cfs
4) pond area	0 acres
Operating performance	
1) fan power requirement	1.68 MWe
2) pump power requirement	.62 MWe
3) annual energy use by auxiliaries (pump and fan)	11,200 MWhrs
4) maximum meteorology-induced capacity reduction	10.9 MWe
5) annual lost electrical energy due to meteorology-induced capacity reduction	15,500 MWhrs
Operating economics	
1) annual cost of power for operating cooling system	\$ 0
2) annual loss of energy penalty for meteorology-induced power reductions	\$ 850,000
3) capacity penalty for auxiliaries	\$ 3,100,000
4) capacity penalty for meteorology-induced power reduction	\$ 2,100,000
Present-worth costs	
1) loss of capacity	\$10,200,000
2) loss of energy production	\$13,500,000
3) operation of cooling system	\$ 770,000
4) capital investment	\$15,500,000
5) total	\$40,000,000
Incremental power production cost	8.24 mills/KWhr

Table 5.3

Capital Cost of Optimum Simple Dry Cooling System

Site - Inyokern

Solar Plant - 100 MWe, 6-hour TES, 8" Hg condenser pressure limit

Design parameters

1)	number of dry cooling modules	15
2)	condenser flow rate	87.4 cfs
3)	regeneration flow rate	0 cfs
4)	pond area	0 acres

Capital Costs

1)	dry tower module and foundations	\$3,690,000
2)	thermal storage pond	\$ 0
3)	condenser	\$1,370,000
4)	electrical equipment	\$ 610,000
5)	pipeline	\$ 553,000
6)	circulating water pump structure	\$ 52,000
7)	water pumps and motor	\$ 143,000
8)	indirect capital cost	\$1,600,000
9)	total capital cost	\$8,020,000

The cost of the simple dry cooling system with no pond is 8.24 mills/KWhr. Thus, the addition of the 1 acre pond results in a 15% reduction in the cost of dry cooling while the addition of the 4 acre pond results in a 21% reduction in the cost of dry cooling. The total capital cost for these systems is, of course, greater. However, as these reduced total costs indicate, this capital cost increase is more than offset by the greatly improved plant performance. The 4-acre pond reduces the total annual lost energy production from 15,500 MWhrs to 6,430 MWhrs while the maximum loss of capacity is dramatically decreased from 10.9 MWe to 2.06 MWe (taking into account the ability to shut off 1.68 MWe of fan power). The cost and performance description of the 15 dry tower module and 1 and 4 acre pond systems is given in Table 5.4. The capital cost of the 1 and 4 acre thermal storage ponds are \$340,000 and \$1,060,000, respectively. The capital cost for the other components of the cooling system are identical to those in Table 5.3.

Although the system composed of 15 dry tower modules and a 4-acre pond is the least costly of the above alternatives, this may not be the optimum combination of dry tower modules and pond storage volume. By examining different alternative designs it was determined that a cooling system composed of 6 dry tower modules and a 6 acre storage pond yielded the low incremental cost of power reduction of 6.06 mills/KWhr and provided the same cooling performance as the simple system composed of 15 modules. This represents a 26% reduction in the cost associated with dry cooling. The performance and cost for this system is given in Tables 5.5 and 5.6.

Steam Turbine with 5" Hg Back Pressure Limit

Operating with a steam turbine with a 5" Hg back pressure limit requires that the condenser temperatures not exceed 133°F at any time. This is in comparison to the 8" Hg limit of 153°F. As pointed out earlier, 30 dry tower modules are required to maintain this 133°F limit at a cost of 10.5 mills/KWhr for a 100 MWe solar plant with 6 hours of thermal energy storage.

A number of different combined pond-tower system designs were evaluated for the case of the 5" Hg condenser pressure limit. The best

Table 5.4

Cost and Performance of Combined Pond-Tower Systems

Site - Inyokern

Solar Plant

100 MWe, 6 hour TES - 8" Hg Condenser
Pressure Limit

Design parameters

1) number of dry cooling modules	15	15
2) condenser flow rate	87.2	87.2 cfs
3) regeneration flow rate	36.6	87.2 cfs
4) pond area	1.0	4.0 acres

Operating performance

1) fan power requirement	1.68	1.68 MWe
2) pump power requirement	.62	.62
3) annual energy use by auxiliaries (pump and fan)	18,500	18,500 MWhrs
4) maximum meteorology-induced capacity reduction	8.72	3.74 MWhrs
5) annual lost electrical energy due to meteorology-induced capacity reduction	9,900	6,400 MWhrs

Operating economics

1) annual cost of power for operating cooling system	\$ 280,000	\$ 281,000
2) annual loss of energy penalty for meteorology-induced power reductions	\$ 468,000	\$ 304,000
3) capacity penalty for auxiliaries	\$ 835,000	\$ 835,000
4) capacity penalty for meteorology-induced power reduction	\$ 1,740,000	\$ 747,000

Present-worth costs

1) loss of capacity	\$ 4,900,000	\$ 3,064,000
2) loss of energy production	\$ 7,420,000	\$ 4,820,000
3) operation of cooling system	\$ 5,260,000	\$ 5,372,000
4) capital investment	\$16,300,000	\$18,130,000
5) total	\$34,000,000	\$31,300,000

Incremental power production cost

7.04 mills/KWhr 6.48 mills/KWhr

Table 5.5
System Cost and Performance with 5" Hg Condenser Pressure Limit

Site - Inyokern			
Solar Plant			
		100 MWe, 6-hour TES	5" Hg Condenser Pressure Limit
Design parameters			
1)	number of dry cooling modules	30	6
2)	condenser flow rate	124	62.3 cfs
3)	regeneration flow rate	0	62.3 cfs
4)	pond area	0	10 acres
Operating performance			
1)	fan power requirement	3.36	0.67 MWe
2)	pump power requirement	0.84	0.57 MWe
3)	annual energy use by auxiliaries	20,500	10,800 MW hrs
4)	maximum meteorology-induced capacity reduction	5.57	5.13 MWe
5)	annual lost electrical energy due to meteorology-induced capacity reduction	8,030	8,100 MW hrs
Operating economics			
1)	annual cost of power for operating cooling system	\$ 0	\$ 113,000
2)	annual loss of energy penalty for meteorology-induced power reductions	\$ 380,000	\$ 384,000
3)	capacity penalty for auxiliaries	\$ 5,660,000	\$ 760,000
4)	capacity penalty for meteorology-induced power reduction	\$ 1,110,000	\$ 1,020,000
Present-worth costs			
1)	loss of capacity	\$13,100,000	\$ 3,400,000
2)	loss of energy production	\$ 6,010,000	\$ 6,090,000
3)	operation of cooling system	\$ 1,330,000	\$ 2,530,000
4)	capital investment	\$26,600,000	\$14,900,000
5)	total	\$47,200,000	\$27,000,000
Incremental power production cost		10.5	5.6 mills/KWhr

Table 5.6

Capital Cost Summary for Case of 5" Hg Condenser Pressure Limit

Site - Inyokern

Solar Plant

100 MWe, 6-hour TES 5" Hg condenser
Pressure Limit

Design Parameters

1) number of dry cooling modules	30	6 modules
2) condenser flow rate	124	62.3 cfs
3) regeneration flow rate	0	62.3 cfs
4) pond area	0	10 acres

Capital Costs

1) dry tower modules and foundations	\$ 7,370,000	\$1,470,000
2) thermal storage pond	\$ 0	\$2,510,000
3) condenser	\$ 1,370,000	\$1,370,000
4) electrical equipment	\$ 1,220,000	\$ 240,000
5) pipeline	\$ 790,000	\$ 411,000
6) circulating water pump structure	\$ 74,000	\$ 38,700
7) water pumps and motors	\$ 205,000	\$ 106,000
8) indirect capital cost	\$ 2,760,000	\$1,540,000
9) total capital cost	\$13,800,000	\$7,710,000

design was found to be a system composed of 8 dry tower modules and a 10 acre thermal storage pond. The incremental power production cost for this design was determined to be 5.6 mills/KWhr. This represents a dramatic reduction of 46% in the cost of heat rejection. This cost reduction is directly attributable to the fact that the capital cost of the system is reduced from \$13.8 million to \$8.4 million while the capacity supply and energy production performance of the plant remains essentially unchanged.

The performance and costs of these simple dry tower and the combined pond-tower systems are presented in Tables 5.5 and 5.6

Solar-Thermal Plant with Minimal Thermal Energy Storage

It is possible that some solar-thermal electric plants may be designed with only a very limited amount of thermal energy storage. For these cases, the benefit of the combined pond-tower dry cooling systems is greater than that for the previously discussed case of 6 hours of thermal energy storage. The cost of a simple dry cooling system to maintain the 8" Hg condenser pressure limit is 9.35 mills/KWhr. This system consists of 15 dry tower modules. A combined pond-tower system consisting of 5 dry tower modules and a 4 acre pond results in an incremental cost of power production of 5.35 mills/KWhr or a 42% reduction in total cost.

5.4 Combined Pond-Tower System Economics at Barstow Site

The use of dry cooling at the Barstow site is generally more costly than at the Inyokern site because of the higher ambient temperature. The maximum ambient temperature at Barstow is about 112°F while at Inyokern the maximum is about 102°F. This is a large difference with respect to maintenance of a 133°F (5" Hg) or 153°F (8" Hg) condenser pressure limit.

The size and cost of the simple dry tower system to maintain the different condenser pressure limits at Barstow are:

8" Hg	-	24 dry tower modules	-	10.3 mills/KWhr
5" Hg	-	48 dry tower modules	-	16.1 mills/KWhr

These costs are for a 100 MWe plant with 6 hours of thermal energy storage. Two pond-tower systems which maintain equivalent levels of plant performance are:

8" Hg Condenser Pressure Limit	-	7 dry tower modules 6 acre pond	-	6.72 mills/KWhr
5" Hg Condenser Pressure Limit	-	15 dry tower modules 6 acre pond	-	7.60 mills/KWhr

Thus, the combined pond-tower cooling system has a total cost which is between 48% and 66% of the cost of the simple dry tower system depending on the condenser pressure limit.

5.5 Thermal Behavior of Storage Pond

As indicated in the previous section, the optimum pond sizes in a variety of instances was in the range of 4 to as much as 10 acres. These size ponds represent "full-day" or larger ponds as discussed in section 3. To put the storage capacity in perspective, a 6 acre pond has the capacity to store the full waste heat load of the plant for a typical summer day with a temperature rise of only 18°F. Thermal storage ponds of this size not only allow taking advantage of the normal night-to-day temperature variations but also functions to isolate the condenser from extreme day-to-day temperature fluctuations.

In Figure 5.3 is plotted the 3:00 pm ambient temperature and the pond bottom (i.e., condenser inlet temperature) for one year for a system of 6 dry tower modules and 6 acre pond at the Inyokern site. The temperatures for every third day are plotted. Note that the temperature of the water supplied to condenser by the pond during the fall, winter and spring is often below the ambient temperature at 3:00 pm, usually the hottest time of the day. During the summer, the condenser inlet temperature exceeds the ambient temperature by only a few degrees on the hottest days.

With these large sized ponds, the amount of heat transferred from the pond surface can be significant. For the same 6 acre pond system, the total amount of heat transferred to the atmosphere from the pond

Figure 5.3

Thermal Behavior of Storage Pond

surface during the year is 3.83×10^{11} BTU. This is 3.4% of the total heat which is stored in the pond over the year and represents an average rate of pond cooling of about 2.5°F per day.

5.6 Summary of Economic Analysis and Impact of Economic Assumptions

In Figure 5.4 is depicted the comparative costs of the combined pond-tower dry cooling systems and the conventional alternatives. Note that an estimate of the cost of a conventional evaporative cooling system is presented along with the previously presented costs of dry cooling systems. The stated cost, 3.8 mills/KWhr, for the evaporative cooling is based on the assumption of the use of conventional wet cooling towers with no appreciable cost of water. This cost has been estimated from the cooling system costs presented in references 12 and 25 .

The most significant result presented in Figure 5.4 is that the cost of heat rejection using the combined pond-tower dry cooling system approaches the cost of heat rejection using evaporative cooling with low cost water ($\approx$ \$100/acre foot). Most importantly, the overall plant power generation performance with the combined pond-tower system is very similar to that achieved with an evaporative cooling system. It is not necessary to trade-off (i.e., reduce) plant performance to achieve an acceptable cost of heat rejection with the combined pond-tower system. Because conventional evaporative and combined pond-tower dry cooling systems can be compared on an equal performance basis, the relative cost of the systems is not sensitive to the assumed penalties for the meteorology-induced power losses.

For the simple dry cooling systems this is not the case. As shown in Figure 5.4, requiring high cooling performance (i.e., maintenance of 5" Hg condenser pressure) results in very high cost of heat rejection. To achieve lower costs, the designer needs to consider the benefit/penalty trade-off of reduced plant capacity supply and power generation performance. However, as demonstrated for both the Inyokern and Barstow, the increase in the acceptable condenser pressure from 5" Hg to 8" Hg (i.e., increase in power loss from 5% to 11%) still does not yield a lower cost than that

Figure 5.4

achieved with the high performance combined pond-tower system. It is significant that this result is obtained even for the conservative (i.e., low) assumed capacity penalty rate of \$200/KWhr.

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